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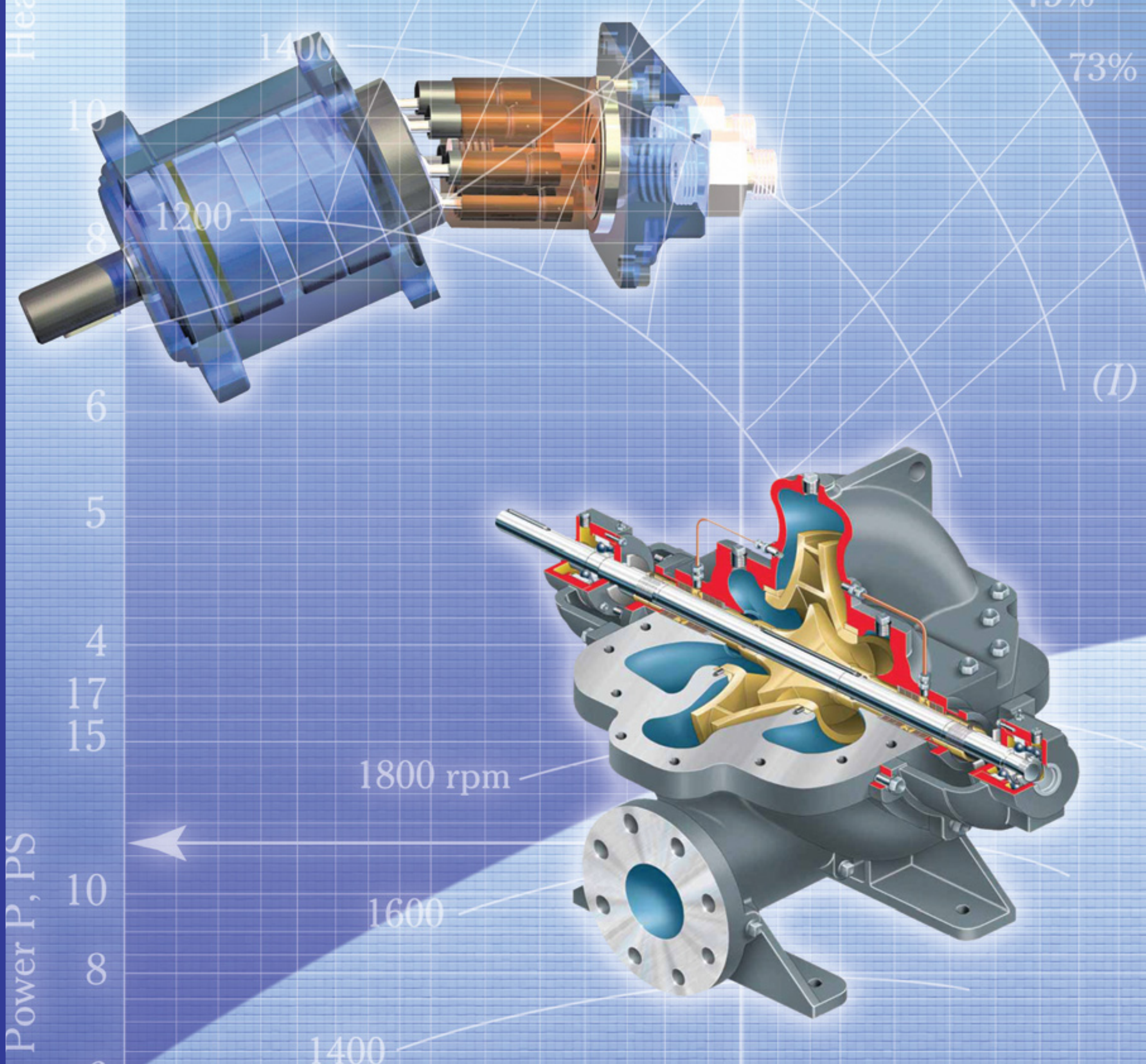
GOLD MEDAL AWARD OF
THE ACADEMY OF ATHENS

MERCHANT MARINE ACADEMIES
EDUCATIONAL TEXTBOOK

PUMPS

IOANNIS K. DAGKINIS
ALEXANDROS I. GLYKAS



ATHENS 2017

EUGENIDES FOUNDATION

GOLD MEDAL AWARD OF THE ACADEMY OF ATHENS

MERCHANT MARINE ACADEMIES

EDUCATIONAL TEXTBOOK

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PROLOGUE OF THE EUGENIDES FOUNDATION

In 1952, Eugene Eugenides (1882-1954) ordered with his testament the establishment of the Eugenides Foundation, the only purpose of which would be "to contribute to the education of young Greeks in the fields of science and technology". The founder and sponsor of the Eugenides Foundation rightly predicted that a necessary factor for the progress of Greece was the sound training of Greek skilled workmen in compliance with the vocational education of other European countries.

On 23 February 1956, the establishment of the public benefit "Eugenides Foundation" was approved, the administration of which was undertaken by the founder's sister, Marianthi Simou, according to Eugenides' explicit wish. Thus, Marianthi Simou and her scientific advisors embarked on the implementation of the Foundation's purpose and the fulfillment of one of the most pressing needs of our national life. Marianthi Simou's work was continued from 1981 onwards by Eugenides' valuable associate and successor, Nikolaos Vernikos-Eugenides (1920-2000). Since 2000, the work of the Foundation is carried on by Mr Leonidas Dimitriadis-Eugenides.

One of the very first activities of the Eugenides Foundation, immediately after its founding, was the writing and publishing of suitable textbooks for students of Technical Schools, as it was found that it was urgently needed to equip students with series of books, which would lay the foundations for their education and would, at the same time, constitute an invaluable library for every skilled workman. The final result of this activity was The Skilled Workman's Library (1957-1975), which includes 32 titles, the Technician's Library (1962-1975), which consists of 50 titles, the Technical Library (1969-1980) with 11 titles and the Technical Assistant Chemist's Library (1971-1973) with 3 titles. Furthermore, from 1977 to the present, 171 titles have been published for the students of Technical and Vocational Schools and 16 for the students of Intermediate Technical and Vocational Schools.

Yet another book series of the Eugenides Foundation is the Seafarer's Library (1967-today), which resulted from the cooperation between the Eugenides Foundation and the Directorate of Maritime Education of the Ministry of Shipping and Island Policy. The writing and publishing of educational textbooks for the students of Merchant Marine Academies was assigned to the Eugenides Foundation with the No. 61288/5031/9.8.1966 decision of the Ministry of Mercantile Marine, when the responsible Publications Committee also functioned, which had been established in 1958. The collaboration of the Eugenides Foundation with the Ministry of Shipping and Island Policy was renewed with the ministerial decision No. M2111.1/2/99, as it was amended by the ministerial decision No. M3611.2/05/05/16-12-2005, with which the Ministry of Shipping and Island Policy assigned to the Eugenides Foundation the writing of educational textbooks for the Merchant Marine Academies.

The Seafarer's Library includes 134 titles so far: 27 titles for the State Merchant Marine Schools, 42 titles for the Higher State Merchant Marine Schools, 37 titles for Merchant Marine Academies, 13 On Board Training Record Books and 15 Maritime Textbook Translations.

All the books of the Seafarer's Library, apart from conforming to the requirements of the curricula of the Academies and satisfying students' needs, are generally useful to all Merchant Marine officers, who follow a seafarer's career or get a promotion to higher ranks. Furthermore, the authors and the Publications Committee use their best efforts so that the books are scientifically sound and adapted to the needs and the abilities of the students.

During 2012-2013, and according to the No. M3616/01/2012/26-09-2012 document, the

Ministry of Shipping and Island Policy entrusted the Publications Committee of the Eugenides Foundation with the establishment of an experts' special task group to update the curricula of the Merchant Marine Academies, the Seafarers Post Training Centres and the Special Schools for Deck and Engine Officers, thus applying the new requirements for the training and certification for seafarers, according to the International Convention STCW '78 (Standards for Training, Certification and Watchkeeping for seafarers – Manila amendments 2010). On the basis of the new curricula for Merchant Marine Academies that were implemented for the first time during the academic year 2013-2014, the update of all educational textbooks began in 2014, in order to be compliant with the new international requirements.

Offering its publications to instructors, students of Merchant Marine Academies and to all merchant marine officers, the Eugenides Foundation continues to contribute to the technical education in Greece, fulfilling for more than 60 years the vision of its founder, late benefactor Eugene Eugenides.

The scientific advisor of the Greek edition of the book *Pumps* (*Αντλίες*, Eugenides Foundation 2016) is **Nikolaos Tsitsos**, Naval Architect, Mechanical Engineer of NTUA and Instructor in the Merchant Marine Academy of Aspropyrgos.
The translation of the book *Αντλίες (Pumps)* in English was made by **Ioannis K. Dagkinis**, Chief Engineer, STLP University of the Aegean, Dpt. of Shipping Trade & Transport.
The text was edited by Dr **Paraskevi Papaleonida** (PhD in English Language and Literature, Aristotle University of Thessaloniki), Maritime English Instructor, Merchant Marine Academy of Macedonia.

EUGENIDES FOUNDATION

THE SEAFARER'S LIBRARY

PUMPS

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AUTHORS' PREFACE

The book Pumps aims to provide the students of Merchant Marine Academies (Engineers), as well as every pump user, with the necessary knowledge related to the types of pumps and their applications on ships, as well as information on prevention, determination of cause, and designation of solutions on problems that arise during their use.

The book conforms to the existing curriculum of Merchant Marine Academies (Engineers) and is divided into the following three thematic sections:

The ***first section*** *(chapters 1 – 3) contains the general classification, the types and the operational details that differentiate pumps.*

The ***second section*** *(chapters 4 – 6) focuses on the characteristic variables of pumping systems, the flow and performance characteristic curves of dynamic pumps, and discusses how the fluids handled by pumps affect their operation.*

The ***third section*** *(chapters 7 – 10) presents the pumps commonly found on ships, the pumps that are used depending on the ship's propulsion system, as well as their sealing, bearings, start-up and operation. Finally, the basic principles of the study and the design of pumping systems are concisely presented.*

Thus, the content of the book addresses not only the needs of students of Merchant Marine Academies, but those of Marine Engineer Officers as well.

From the 10 chapters of this book, chapters 1, 2, 3, 4, 7, 8 and 9 have been written exclusively by I. Dagkinis, while chapters 5 and 6 by I. Dagkinis and A. Glykas. Throughout the writing process, the book Μηχανική Ρευστών ("Fluid Mechanics") by N. Pantzalis, Eugenides Foundation, 2008, has been a significant aid. Chapter 10 of this book was integrated in Pumps after being revised by I. Dagkinis.

We would like to thank the Publications Committee and the Eugenides Foundation for commissioning us to write this book, Mr. N. Tsitsos, instructor in the Merchant Marine Academy of Aspropyrgos, who contributed to the integrity of the book with his substantial interventions in its content, as well as the staff of the Eugenides Foundation Publications Department for their cooperation and contribution to the complete and carefully executed presentation of the book.

Considering that this book can be improved as a result of creative and well-intended criticism, we would like to thank in advance readers that will send their comments and recommendations, which will contribute to its improvement in a future edition.

The authors

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TABLE OF SYMBOLS

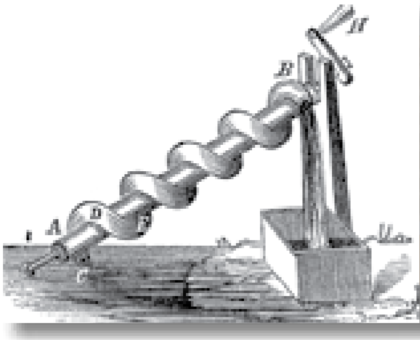
A	area, surface (m^2)
a	static coefficient
B	buoyancy (N)
BEP	best efficiency point
b	dynamic coefficient
C	constant
D	diameter; unless otherwise subscripted
D	discharge
d	discharge side piping system
D_V	displacement
F	force (N or Kg_f or Kp)
Fr	Froude number
G	weight (N)
g	acceleration due to gravity [9.80665 m/s^2 (32.174 ft/sec^2) at earth sea level]
H	head of liquid column (m or ft)
H_t, H	total discharge head (m)
H_d	discharge head (m)
H_{st}	static head (m)
H_{stp}	pump static head (m)
H_{sts}	suction static head (m)
H_s	suction head (m)
H_v	velocity head (m)
h	hydrostatic height (m) of column H_2O or liquid
h_f	height of energy losses due to friction (m)
h_{fS}	height energy losses in suction, (m) of column H_2O or liquid
I_o	moment of inertia ($kg{\cdot}m^2$)
K	coefficient of local losses
L	l length (m)
(L_1, L_2)	angular momentum ($kg{\cdot}m^2/s$)
LL	geometric similarity (Length Length)
M	torque (N·m or lbf·ft
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s or lbf·sec/ft)
$\dot{m}_s$	sea mass flow rate (kg/s or lbf·sec/ft)
$\dot{m}_{st}$	steam mass flow rate (kg/s or lbf·sec/ft)
N_s, $N_{s(US)}$	specific speed (in rpm, gpm, ft units
N_{ss}	suction specific speed (in rpm, gpm, ft units)
NPSH	net positive suction head (m or ft)
NPSHa	available NPSH (m water column or of liquid)
NPSHr	required NPSH to prevent significant loss of pump or to protect the pump against cavitation damage, whichever is greater (m water column or of liquid)
n	rotational speed of the impeller (rpm)
OP	operating point
P	power (W)
P_l	delivered, assigned, real or effective power (W)
P_{ls}	power losses (W)
P_m	motor power (W)
P_n	normal power (W)
P_s	shaft power (W)
P_{th}	theoretical power (W)
p	total pressure (Pa or bar)
p	pressure (Pa or bar)
p_{atm}	atmospheric pressure (Pa or bar)
p_v	vapor pressure (Pa or bar)
Re	Reynolds number
r	radial distance from axis of rotation (m or ft)
S	slip
SG	specific gravity, namely, the ratio of liquid density to that of water at 15.6°C (60°F), so $SG = \rho/\rho_{Ref}$ with $\rho_{Ref} = 1000\ kg/m^3$
T	temperature (°C or °K)
t	time (sec or s)
u	tangential velocity component
V	volume (m^3 or ft^3)
$\dot{V}$	volume flow rate or, more conveniently, “flow rate” or “capacity”, m^3/s (ft^3/sec)
$\dot{V}_i$	inner flow rate (m^3/s)
$\dot{V}_n$	normal volume flow rate (m^3/s)
v	velocity ($m{\cdot}s^{-1}$)
v_r	radial velocity components
W	work (J)
w	normal velocity component of fluid relative to rotating impeller, m/s (ft/sec)
z	height (m)
∞	infinity symbol, for an infinite number of blades that also produce zero blockage of the flow

Greek symbols

α	angle, of the absolute velocity vector (°)
β	angle, of the relative velocity vector or impeller blade in the plane of the velocity diagram from the circumferential (tangential) direction (°)
γ	fluid weight density [N/m^3 (lbf/ft^3) $=\rho g$ ($1N = 1\ kg{\cdot}m/s^2$)]
Δp	pressure difference
ε	roughness (absolute) height (m or ft)
η	pump efficiency
η_h	hydraulic efficiency
η_V	volumetric efficiency
η_m	mechanical efficiency
η_{pm}	pump motor efficiency
μ	absolute viscosity ($Pa{\cdot}s$ or $N{\cdot}s/m^2$, or $lbf\text{-}s/ft^2$)
ν	kinematic viscosity ($=\mu/\rho$) (m^2/s or ft^2/s)
ρ	fluid mass density (kg/m^3 or $lbf{\cdot}s^2/ft^4$) $= \gamma/g$
σ	cavitation coefficient
ΣP_m	mechanical power losses (W)
Σh_{pf}	the height due to flow losses (hydraulic loss) inside the pump (m)
Ω_s	universal specific speed (unitless)
ω	angular velocity (rad / s)

Subscripts

D	discharge flange or exit (ex) of the pump
S	suction flange or inlet (in) of the pump
d	discharge, discharge side of blade or passage or piping system
ex	external
s	suction, suction side of blade or passage or piping system
m	"mechanical"
V	volumetric
I	input to fluid
L	losses
n	normal
out	pump outlet flange or port
S	shaft
st	static
in (or S)	pump inlet flange or port
f	friction
l	liquid
hyd	hydrostatic
h	hydraulic
mp	pump motor
D	drag due to disk friction
t	total
vap	vapor
i	inner
th	theoretical
env	environment



CHAPTER ONE
Introduction

1.1 Historical background.

Since ancient times, ensuring the availability of water has been one of the most important selection criteria for choosing a place as a permanent living residence. Water supply has been a substantial prereq-uisite for the development of human activities. The physicist, philosopher Thales the Miletus (640 – 546 BC) had characterized water as «ἀρχήν τῶν ὄντων ἀπεφήναντο το ὕδωρ...ἐξ ὕδατος δε φησί πάντα εἶναι και εἰς ὕδωρ πάντα ἀναλύεσθαι...»[1], and the great lyric poet Pindar (552 – 443 BC) had emphasized the value of water by indicating that «ἐκ τῶν στοιχείων τοῦ κόσμου τό ὕδωρ εἶναι τό ἄριστον...»[2]. Therefore, the supply of water was considered one of the main infrastructures of any organized community, and the conservation of water quantities for daily needs is reflected in numerous excavation finds. When the ancient settlements began to be organized into cities, it was no longer sufficient for water to simply be in close proximity. The need to create transmission and distribution networks of water became imperative, in order to ensure the development of organized cities. The creation of pipe networks was supplemented by the development of mechanisms which exploit the properties of the fluids. These mechanisms were used for the transfer of water.

The ***Archimedes screw*** (or hydraulic interminable screw) was such a mechanism. Archimedes of Syracuse (287 – 212 BC) was a great Greek mathematician, physicist, engineer, inventor, and astronomer of the Hellenistic period. The Archimedes screw (fig. 1.1) consisted of a wooden shaft, which brought thin and flexible windings of osier or willow twigs, linked to each other to create an interminable screw.

The screw was encased by a wooden cylinder and was tangent to the interior of the cylinder. The machine was installed in the water with a 30° inclination. Due to the rotations of the interminable screw, the amount of water entrapped within the screw was raised and flowed out of the wooden cylinder through an orifice. It was a water-pumping mechanism, with big flow but small elevation difference and was used in many areas of the ancient world. The operating principle of Archimedes' screw is applied even today in technological applications that are used to transfer liquids or pulverized materials.

Fig. 1.1
The Archimedes' screw.

Another mechanism was the ***piston force pump of Ctesibius*** (285–222 BC), mathematician, engineer, inventor of the Hellenistic period and founder of the Alexandrian School. Ctesibius' pump consists of two identical cylinders with pistons inside them which, with the aid of a hand lever, move in reciprocating motion (fig. 1.2). The cylinders are submerged in water.

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[1] The archetypal element, a primary ingredient of all things is water.
[2] From all the world's elements, water is the optimum one.

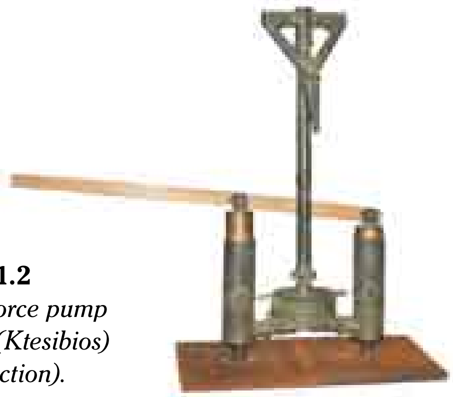


Fig. 1.2
The piston force pump of Ctesibius (Ktesibios) (reproduction).

As the pistons move inside the cylinders, vacuum is created, which, in turn, causes the suction of the water. Then each piston, alternatively, forces the water to flow through the tube and be conveyed out of the space where the pump is submerged.

Also, Filon the Byzantius (c. 250 BC) invented a pump with chains, a winch and buckets (paternoster chain pump)[3], as well as a type of air pump (blower). The Heron ***fire pump***, (Heron was a disciple of Ctesibius) (c. 1st century AD.), was a twin suction force piston pump with continuous water flow. It consisted of two pistons that reciprocated opposite each other, with the help of a common articulated lever, inside two vertical cylindrical containers submerged in the water tank which was probably wheeled (fig. 1.3).

Fig. 1.3
The fire pump of Heron (reproduction).

This pump is considered as the first discharge piston pump of humanity. One can reasonably argue that similar or simpler machines were developed in other ancient cultures (e.g. Babylonians, Assyrians, etc.), but are unknown due to lack of written evidence.

The required energy for the operation of all these premature water-handling machines, which were mainly used for potable water and irrigation water, was provided by human muscle power. Later, wind energy, as well as the potential energy of flowing water, were exploited – to a limited extend – for pumping water for rivers.

Generally, the most widely–used pump before the industrial revolution was the pump which consisted of a wheel, chain and customized containers or fins (chain pumps). We encounter this pump, in the Roman Empire, in Byzantium, in Europe, and also in China during the 1st century AD (fig. 1.4). Improved water pumping systems appear in the Arab world (Al-Jazari, 1200, fig. 1.5).

Fig. 1.4
Typical square pallet chain pumps in China (reproduction).

There was a peak in the requirements for fluid handling during the industrial revolution, and these requirements were no longer limited to water. Along with technological progress, came the development of more complex machinery that accommodated such requirements.

Thus, in the early 18th century the first centrifugal

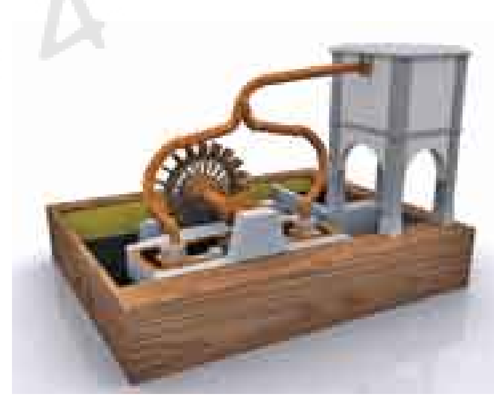


Fig. 1.5
Arabian improved pumping system (reproduction).

[3] Mechanism for pumping water from the well.

pump and the air pump were manufactured by Denis Papin (1647–1712), and later, in the 19^{th} century, the steam reciprocating pump was constructed by Henry R. Worthington (1860), the fan pump by Osborne Reynolds (1875), etc.

1.2 Fluid transfer and fluid handling machinery.

There are certain energy requirements that should be satisfied in order to achieve the flow of a fluid. For example, a water reservoir (natural or artificial) located at a higher level flows to a reservoir which is placed at a lower level due to the difference of dynamic energy (namely we have flow due to gravity 2→1, power production) (fig. 1.6).

But the inverse flow, viz., from the lower to the higher level, could be obtained only if the necessary dynamic energy is given to the water by an external source (1→ 2, power absorption). The same happens with pressure energy. This means that, if a high pressure container is connected to a low pressure container, the fluid from the high pressure container (or space) flows to the low pressure container (or space). Hence, to cause inverse flow (i.e. from a space where low pressure prevails to a space with high pressure) energy should be added to the fluid.

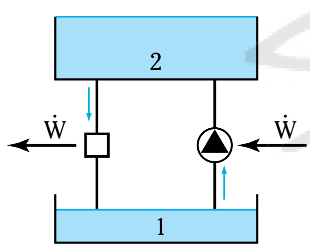


Fig. 1.6
Liquid flow between tanks with altitude difference.

But beyond the energy difference between the two spaces of fluid transfer, there are also energy losses due to friction, which are manifested in the fluids during their flow and must be addressed.

There is also the need for increasing the flow rate and the corresponding speed in the various flow systems (networks). If we add this need to the previously mentioned energy requirements that should be satisfied, then it becomes obvious that additional energy needs to be transferred to the fluids for their handling.

Special machinery is used to transmit the additional power, for the purpose of fluid transfer. The design and operation of this machinery depend on the nature of the fluid. Therefore this machinery is classified into two major categories:

1) The ***machinery handling uncompressed fluids (liquids)***, called ***pumps*** and

2) the ***machinery handling compressed fluids (gases)***, namely ***fans***, ***blowers*** and ***compressors***. Also, vacuum pumps or air pumps are appliances which are used for handling gases. These appliances pump air or other gases (e.g. refrigerant gases) from a space in order to achieve very low pressure (vacuum).

In both cases, energy from the machinery (namely mechanical work) is transferred for the movement of the fluid. The result of this transfer is ultimately either to increase the supply, and hence the speed (namely kinetic speed), increase the pressure (pressure energy), or (in the case of liquids) increase the level of the fluid (dynamic energy).

This classification cannot be absolute, since there are certain types of pumps (ejectors or Giffard), that handle both fluids and gases.

It must be noted that the mechanical work which is transferred from this machinery to the fluid is not actually produced by them. Actually, they are energy intermediaries, who transmit (in an appropriate form) the received energy that is produced by a heat engine or an electric motor to the fluids. The efficiency (η) depends on the mechanical characteristics of the ***"mediator"***. The necessary energy, in the simplest cases, is provided by muscle power, e.g. hand liquid pumps or hand air pumps.

As in all machinery there are energy losses, because part of the energy cannot be transmitted to the fluid, due to frictions. Therefore, from the energy point of view, the quality of the machinery is determined by the efficiency degree, i.e. the effective supplied energy percentage, which, in this case, is the energy that reaches the transferred fluid. For example, the apparatus (or device) shown in figure 1.7, receives the fluid in condition (1), and discharges

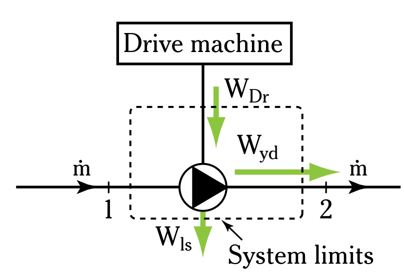


Fig. 1.7
Typical energy transmission scheme during the movement of the liquid.

it in condition (2), which is characterized by higher energy content.

That is, during its course through the machine the fluid receives energy. The apparatus (fig. 1.7) acts as a mediator where: W_{Dr} is the received work from the driving machine and W_{yd} is the work that has been yielded to the fluid. The difference $W_{yd} = W_{Dr} - W_{ls}$ is the friction losses. The efficiency degree of the machine is given as:

$$\eta = \frac{W_{yd}}{W_{Dr}}.$$

There are two general methods for the transmission of the energy to the fluid: The ***positive displacement method*** (by thrust), and the ***method of dynamic change*** (or, more commonly, the principle of increasing the fluid kinetic energy). So, all fluid handling machines can be divided into two major categories, respectively:

1) The ***positive displacement machines***, where pressure is exerted directly to the fluid. The pressure is exercised either by reciprocating machines (reciprocating pumps and compressors), or rotary machines (pumps with lobes, screws, fans, etc.).

2) The ***dynamic machines***, which transmit energy (by applying a force) to the fluid by changing its position or its state. These machines essentially increase the fluid kinetic energy (as it happens in centrifugal pumps, in rotary compressors, blowers, etc). The dynamic devices of centrifugal action are rotary machines (with few exceptions) and are characterized by high rotational speeds.

Nowadays, fluid handling appliances are very popular, both in daily life as well as in the production process. These machines can be met in water supply systems, irrigation, firefighting, heating systems (circulation pumps, fuel pumps), in air conditioning units, in refrigeration installations, in ventilation systems, in cars, in service stations etc., and they also have a highly significant role in the industry section.

Hundreds of these fluid handling machines operate onboard ships and they play a decisive role in the functionality of various ship systems, for safety, comfort, loading, etc. ***All the piping networks established on a ship or at shore contain a principal component made up of a corresponding machine which adds (transmits) to the fluid the energy needed in order to circulate. Otherwise, the piping network cannot function.***

1.3 Basic terms for pumps and pumping systems.

The machinery that performs the transfer of liquid from one place to another through an appropriate piping network, by consuming mechanical work, is named ***pumps***. The process is called ***pumping***, and the system developed is called pumping system. For the pumping procedure to be complete, ***suction*** should be performed from one location and ***discharge*** should be performed in another location, while energy must be imparted to the liquid that is moved. Without a pump in a piping line, effective fluid flow is impossible. As it is shown in figure 1.8(a), in a two-tank system linked with a pipe (communicating vessels system), where their free surfaces are on the same plane and at the same pressure, no flow is possible. The system is in a state of static equilibrium.

In order to achieve a flow from tank A to tank B, a pump should be fitted that will suction the fluid from tank A and discharge it in tank B, adding to it the necessary energy to counterbalance friction losses as well as increasing its dynamic pressure, as the level of tank B is rising.

It makes sense that, the need for a pump, and also the energy amount provided by the pump to the fluid, is higher in two cases. The first case is where the free surface of tank B has been placed higher than the surface of tank A, and the other case is when the free surface of tank B has a higher pressure than the one exerted on the free surface of tank A.

If the surface of tank B is lower than the surface of tank A (or has a lower pressure), the liquid will flow from tank A to tank B, even if there is no pump, due

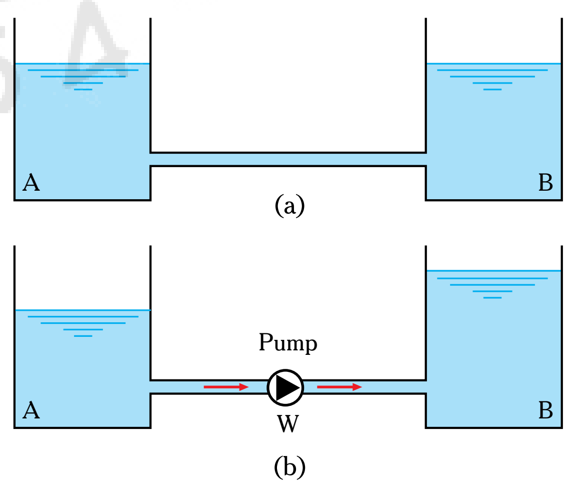


Fig. 1.8
(a) State of static equilibrium
(b) liquid flow by means of a pump.

to the height that causes a difference to the potential energy[4]. But in this case, the flow rate may be lower than the required one. Therefore, the movement of the liquid could be increased when it flows through a pump in order to reach the desired flow rate levels.

In general, the fluid flow from an area with high energy to an area with lower potential energy could take place without the need to insert an external energy supply unit, due to gravity. But, if the energy difference is small and losses are big (because the diameter of the pipe is small or the length of the pipe is large), the flow rate which is ensured by the natural flow is low, and the use of pumps is usually required.

Hence, the use of pumps becomes essential when the desired effect is:

1) ***To achieve fluid flow from a low to a high*** (or equal) ***energy level*** and

2) ***to increase an existing flow rate.***

Thus, the pump is interposed in the pipe line, between the two tanks, and creates suction of the fluid from one side while discharging it to the other side (fig. 1.9). Pumping systems could be open or closed.

A pumping system is an ***open*** system, when the liquid, after discharging (by the pump) and its passage through the equipments in use, is discarded in the environment. There are, however, some pumping systems wherein the liquid circulates in a closed circuit, such as the Main Engine jacket cooling system (M/E).

A pumping system is ***closed*** when the liquid, after the discharge and after passing through the equipments of use, is recirculated in the suction pipe. In these systems, the pumps have to tackle the energy losses due to friction.

A pumping system is composed of three segments (fig. 1.9):

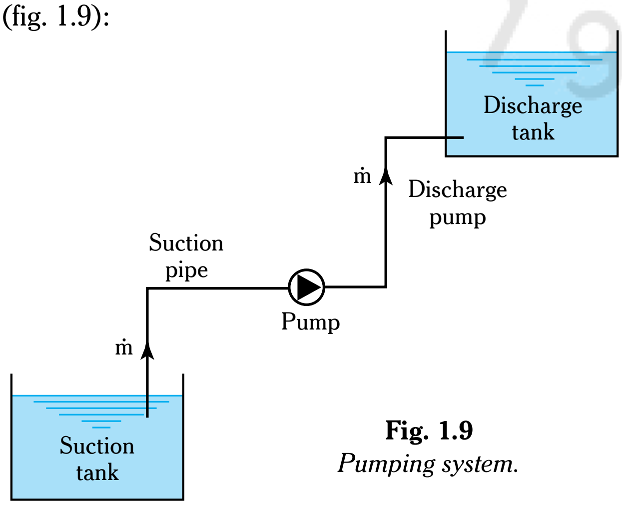


Fig. 1.9
Pumping system.

1) the ***suction segment*** (suction pipe), whereby the liquid is transfered to the inlet of the pump (called pump suction).

2) The ***pump*** or the pump group, i.e. a set of pumps which cooperate for pumping fluid, and

3) the ***discharge*** segment (discharge pipe), where the fluid is directed after being handled by the pump, and through which the liquid continues to flow, after the fluid has acquired the energy added to it by the pump. It is commonly called pump discharge.

1.4 Classification of pumps.

There are various criteria that could be used for the classification of the numerous and ostensibly completely different pumps, which are found in hundreds of applications.

Pumps, for instance, may be classified on the basis of the ***liquid they handle***. Under this classification, they may be divided into viscous liquid handling pumps (handling high viscosity fluids), medium and low viscosity liquid handling pumps, corrosive liquid handling pumps, water pumps, sewage pumps, etc.

On the basis of their ***orientation in space***, they may be divided into vertical, horizontal and (rarely) inclined pumps. On the basis of their ***mode of operation***, they may be classified into reciprocating and rotary pumps. There are additional criteria that could be used, such as the applications they serve, the materials they are constructed from, the power they yield, etc.

But the most important criterion that can serve as a basis for a more comprehensive system of classification and study of pumps is the ***method by which the mechanical work is transmitted to the liquid*** (energy transmission in the form of mechanical work). In fact, the transmission method of mechanical work to the liquid constitutes the operating principle of pumps.

Therefore, their further classification is based on the particular manner through which the transmission of the energy is achieved and on the system geometries employed.

Under this system, namely, their operating principle, pumps may be classified into two major categories (page 6):

1) ***Positive displacement pumps*** (or static type) and

2) ***dynamic pumps*** (or kinetic type).

[4] The energy height of a fluid is defined by the potential energy of the fluid, because of its position related to height.

Classification based on pumps operating principle

1
Positive displacement or static type

- **A) Reciprocating**
  - **Piston/plunger**
    - **Suction single acting**
    - **Discharge single acting**
    - **Double or duplex acting**
  - **Diaphragm**
    - **Fluid operated**
    - **Mechanical operated**
- **B) Rotary**
  - **Single rotor with:**
    - **Vane**
    - **Screw**
    - **Peristaltic**
    - **Rotor or cavity**
    - **Liquid piston vacuum**
    - **Piston** with:
      - a) Axial piston variable displacement
      - b) Radial piston variable displacement
  - **Multiple rotor**
    - **Gears type**
    - **Lobes type**
    - **Screws type**
    - **Circumferential piston**

2
Dynamic or kinetic type

- **A) Rotodynamic**
  - **Centrifugal or radial flow**
    - **Single-stage** which are **Single** or **Double suction***
    - **Multi-stage** which are **Single** or **Double suction***
  - **Mixed flow**
    - **Single-stage** which are **Single** or **Double suction***
    - **Multi-stage** which are **Single** or **Double suction***
  - **Axial flow**
    - **Single-stage:**
      - a) Fixed pitch
      - b) Variable pitch
    - **Multi-stage**
      - a) with Open impeller:
        - – Fixed pitch
        - – Variable pitch
      - b) with Closed impeller
  - **Peripheral flow or vortex or regenerative**
    - **Single-stage**
    - **Multi-stage** with **Open** or **Closed impeller**
- **B) Special effect**
  - **Jet, educator or ejector**
  - **Pressurised air or gas lift**
  - **Hydraulic ram**
  - **Electromagnetic**

* Subdivided into three types: Open impeller, Semi-open impeller, Closed impeller.

CHAPTER TWO
Positive displacement pumps

General introduction.

Positive displacement pumps or static type pumps is the machinery where the transmission of mechanical work from the pump to the liquid is realized by applying a force which forces the liquid to be moved.

If on a liquid elementary volume (fig. 2.1) force F is exerted, and under the influence of this force, the volume is moved at a distance ds, then the work that is given to the liquid is:

$$dW = F \cdot ds$$

The work dW is converted into pressure energy, since the force, as exerted on fluid surface A, exerts pressure equal to:

$$p = \frac{F}{A}$$

It is noted that the operating principle of positive displacement pumps closely approximates the working principle of the first simple pumping systems. When a container, for example, is submerged in a well and then pulled out with the containing water, the work that is transferred to the water that is pulled increases its dynamic energy. The same energy transformation (the work is transformed into dynamic energy) is shown in the Archimedes screw, and in water wheel pumps with containers or with vanes and chains.

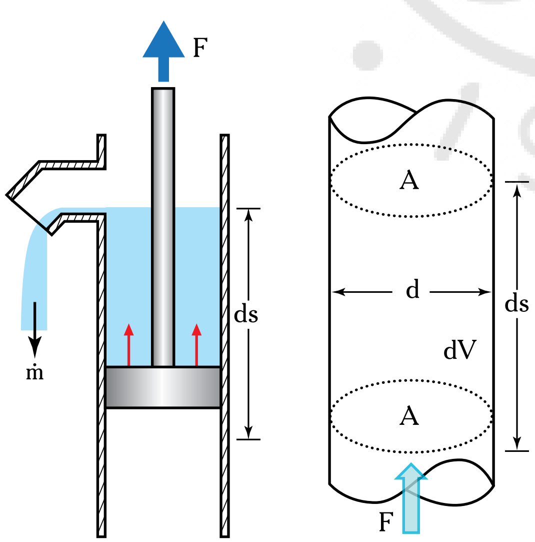


Fig. 2.1
(a) Schematic and (b) linear illustration of the operating principle of positive displacement pumps.

Modern positive displacement pumps differ significantly. They are complex machines, with continuous or periodic mode of operation, which are adapted in piping systems where the energy conversion is different. This means that the mechanical work received by the drive motor, that is transmitted to the liquid, is initially converted into pressure energy. Then, after the liquid passes from the pump, the pressure energy is converted into the forms of energy required by the pumping system.

2.1 Reciprocating pumps.

Pumps can be classified, as presented on page 6, based on the criterion of the transfer method of mechanical work to the liquid and its conversion into pressure energy, and, more generally, their own mode of operation. Furthermore, positive displacement pumps, in accordance to the type of movement of the parts that create the pressure, are divided into two main categories. These are the ***reciprocating pumps*** (or ***piston pumps*** generally) and the rotary pumps. In more detail:

1) ***Reciprocating*** are the pumps which displace the fluid that is passing through them, by the iterative movement of a mechanical element within a chamber with fixed volume. This element creates two new distinct categories of reciprocating pumps (which are detailed in the following paragraphs) and may be:

a) A ***plunger*** or ***piston***, thus creating the category of plunger or piston pumps.

b) A ***diaphragm***, where the reciprocating movement is obtained directly or indirectly by a piston, thus creating the category of diaphragm pumps.

2) ***Rotary pumps*** are the pumps in which the liquid is pushed and moved into the cylinder or the pump casing by suitably shaped rotating lobes, pistons, vanes, etc.

The pumping ability of ***piston pumps*** (reciprocating pumps) is based on the gradual reduction and increase of the volume inside the chamber where the liquid passes from. The pumping ability does not depend on the change of fluid velocity to develop the pressure, because the operation speed is very small compared to that of dynamic pumps. Also, piston pumps do not try to increase the head height but they try to increase the discharge pressure (which replaces the difference of head height, because it is not affected by the suction pressure). Therefore, piston pumps must develop adequate pressure to the discharge side, in order to force the liquid to flow to the pipe line. In general, the liquid in piston pumps is pressed from the suction to the discharge, and this can be achieved at any speed. Also, the developed pressure tends to be very large compared to the physical dimensions of the pumps.

The operation of the piston pump is based on the principle according to which a body submerged either fully or partially in a liquid displaces a volume of liquid equal to the submerged body volume. Thus, if a solid cylinder (e.g., a piston) is submerged in a container A which contains liquid, the volume of displaced liquid in the lower container B is equal to the portion of the cylindrical solid which is submerged in the liquid of the container A (fig. 2.2). Correspondingly, in a piston pump, at each stroke, a certain volume of liquid is drawn in and discharged from each cylinder, almost independently of the pressure difference that prevails into the pipelines at the inlet and the fluid outlet. This volume depends on the volume created by the displacement of the piston within the swept volume of the cylinder, and the size of the cylinder.

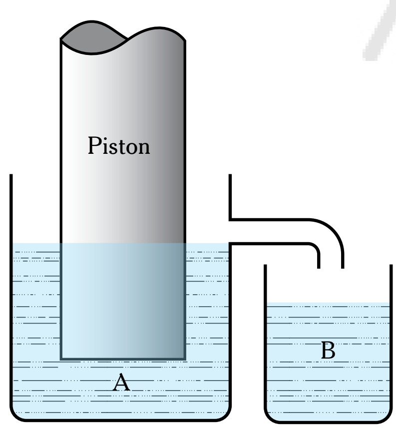


Fig. 2.2
Containers where the transfer of liquid volume is shown, the volume of liquid displaced by a solid equals the volume of the solid.

In the case where the cylinder is connected to a piping system, the reciprocating motion of the piston (displacement pumping action) causes the liquid to be drawn in (suction) into the cylinder when the inner space of the cylinder increases, and to be extruded (discharged) in the reverse motion of the piston. The liquid passage from the pump cylinder is realized through the non-return valves, which are respectively called ***suction*** and ***discharge valves***, and they are installed on the cover (or head) of the cylinder.

Piston pumps are divided into:

1) ***Suction*** or ***discharge pumps***, depending on the manner of pumping and discharge of the liquid.

2) ***Single*** or ***double-acting pumps*** (this distinction relates only to discharge pumps). Pumps are called ***single acting*** when the liquid comes into contact with one side of the piston surface. Otherwise, when the liquid is admitted to either side of the piston where it is alternately drawn in and discharged, they are called ***double-acting***.

3) Depending on the number of cylinders, pumps are divided into ***single-cylinder***, ***twin cylinder, three cylinder*** pumps, etc., or, generally, ***multi-cylinder*** pumps when many cylinders are available.

4) Depending on their arrangement in space, they are classified into ***horizontal*** or ***vertical***.

5) Depending on the method power is produced for their functioning, they are divided into ***manual motivated***, ***steam powered***, ***diesel engine powered***, ***electric powered***, ***hydraulic*** and ***magnetic driven.*** These are called (just) pumps (due to their independent motivation), while, when motivated by a mobile part of the engine or the machine, they are called ***driven pumps*** (or attached pumps).

6) Depending on the means of power transmission for the pump movement, pumps are divided into ***direct transmission*** and ***crank drive piston pumps***. In direct transmission pumps, the piston is driven straight by the drive machine, and in crank drive piston pumps, the piston is moved through a system of crank – connecting rod – yoke and piston rod.

Piston pumps are also divided into ***simplex*** (single devices) and ***duplex*** (coupled devices). This distinction is related to the steam distribution valve[1] for steam powered pumps.

Pumps are characterized as simplex when the distribution steam valve is driven by the piston rod where it provides the steam, and are designated as duplex when each of the steam distribution steam valve is moved by the piston rod of the other complex. The term ***complex*** is used to define the set of piston rod –piston– cylinder of the pump. For example, a pump having two complexes comprises of two steam cylinders with pistons and respectively two cylinders with pistons for the movement of liquid.

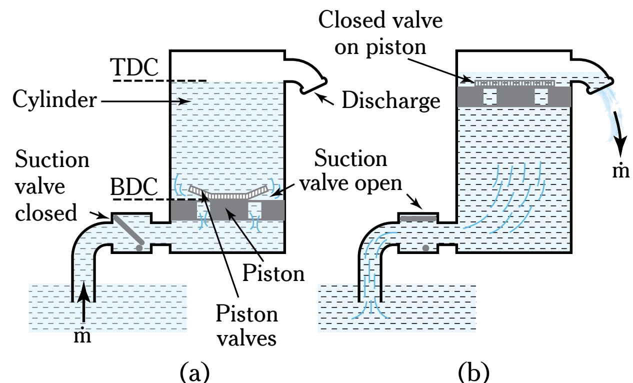


Fig. 2.3
Suction pump operation.

2.1.1 Suction pump.

In one operation cycle, i.e., from top dead center (TDC) to BDC (bottom dead center) and again to TDC, a ***suction pump*** (or common pump) sucks the liquid through a valve, then, by the piston thrust, elevates the liquid, which then flows alone. Therefore, these pumps are also called ***lift pumps***.

A discharge pump (see par. 2.1.2) is an extension of the lift pump because the liquid is sucked through a valve, then is lifted by the piston and discharged through a valve by overcoming an external resistance.

The suction pump consists of the cylinder, where the suction pipe is connected, the suction valve and the piston, where there are valves which allow the fluid passage from one piston side to the other.

The liquid discharge occurs through the opening on the top of the cylinder to the environment, where the pressure is equal to atmospheric pressure.

As shown in figure 2.3(a) the piston is at the lower position of its stroke (BDC), where the suction valve is closed, the valves in the piston surface are open, and the cylinder is full of liquid. As the piston moves up to TDC, the pressure of the liquid closes the valves on the piston, while, simultaneously, creating vacuum in the space underneath the piston, inside the cylinder. The vacuum created means that there are pressure conditions lower than atmospheric pressure or lower than the pressure applied on the free liquid surface, when the liquid is in a closed tank. Then, the suction valve opens and the space between the underside of the piston and the cylinder is fulfilled, as the liquid enters. The existing liquid on the upper side of the piston maintains the valves (on piston surface) in closed position, so that the liquid quantity which reaches (by piston movement) the outlet port can flow out of the cylinder [fig. 2.3(b)]. When the upward stroke is completed and the piston begins to descend to BDC, the valves on the piston surface open and the suction valve closes, almost simultaneously, due to the pressure generated within the cylinder from the liquid that is present. The upper side of piston, within the cylinder through the valves on the piston surface, is filled again with liquid, as the piston reaches BDC. This cycle is repeated and at each stroke the lifted liquid quantity equals the free volume within the cylinder space.

2.1.2 Discharge pump.

A discharge pump is a single-acting pump (fig. 2.4). It consists of the cylinder, the piston, the suc-

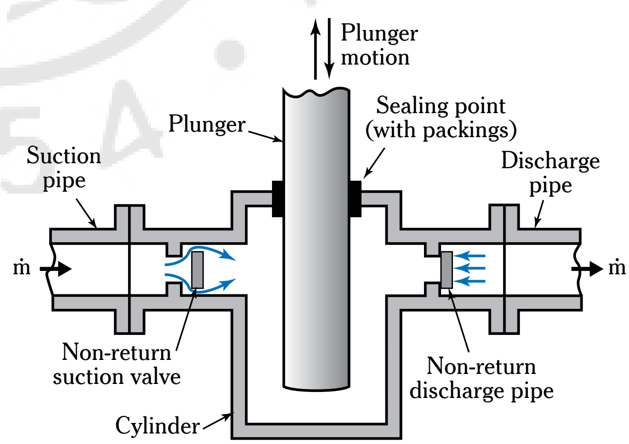


Fig. 2.4
Schematic illustration of a Single-Acting Discharge plunger pump.

[1] A ***steam distribution valve*** is a device with a piston (or slide) valve and ports, in which, driven by the movement of the piston rod, it either reveals the ports and the steam is passed to the steam cylinder, or conceals the ports and stops the passing of steam.

tion and discharge piping system, wherein respectively are positioned the suction and discharge check valves.

As the piston moves toward TDC, the discharge valve remains closed, so that, by the vacuum created into the cylinder, liquid introduction is achieved through the suction valve. Suction will be continued until the end of the stroke to TDC. Subsequently, the piston begins to move to BDC, and, as a result, the liquid which exists inside the cylinder, due to the received pressure from the plunger, presses the suction valve to close. Simultaneously, the discharge valve opens, because the fluids are incompressible, and, under the pressure energy of piston movement, the fluid is passed into the discharge pipe, by overcoming the resistance of the total discharge height.

The discharge is continued till BDC wherein the piston stroke is reversed, and, by the vacuum created inside the cylinder, the discharge check valve closes and the suction check valve opens, so as a new quantity of liquid to be drawn into the cylinder and a new operating cycle to be repeated.

The same operating cycle takes place in double acting discharge pumps. The differences (fig. 2.5)

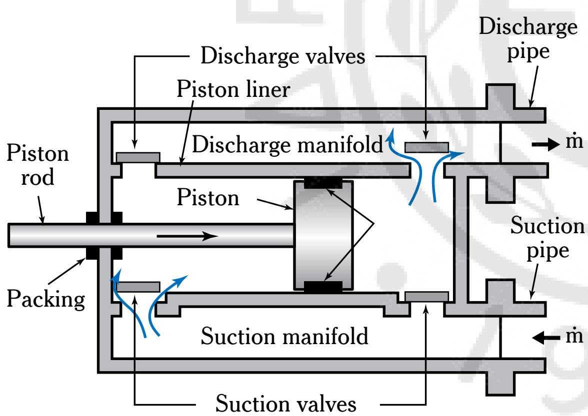


Fig. 2.5
Schematic illustration of a double-acting piston pump.

are identified in the fact that the liquid comes into contact with both sides of the piston, and the arrangement of the suction and discharge valves is placed in pairs on each side of the cylinder, allowing the suction and discharge at TDC and BDC of the piston stroke.

The valves are positioned in such way that, as the piston moves to each direction, the liquid is sucked from one side and simultaneously discharged from the other, and vice versa.

The discharge pressure is not constant due to the variations of the pressure by the reciprocation of the piston. For this reason, a special device may be placed in the discharge pipe line which smoothens these fluctuations and is called air chamber (cushion) or air accumulator tank (see par. 2.3).

2.2 The basic components of piston pumps.

A piston pump consists of a ***cylinder block***, and each cylinder has a ***cylinder wear liner*** (or just liner), a ***cover*** (cylinder head), a plunger or piston, and ***valves***, through which the displacement of the liquid is achieved. The body of the pump constitutes the cylinder liner wherein the piston is reciprocated without requiring a particular liner (sleeve); this is not rare, and typically occurs in small pumps. The power for the pump operation is provided by a driving machine. Depending on the manner by which the movement transmission is obtained, the pump has a piston rod that is directly connected to the drive machine, e.g. a steam cylinder (fig. 2.6), or a connected rod with bearings and crankshaft when the piston is driven via the crank (fig. 2.7). The piston pumps, depending on their characteristics, are rarely constructed for a specific application. Usually they are constructed to be able to be used in a wide range of applications, for example, a pump

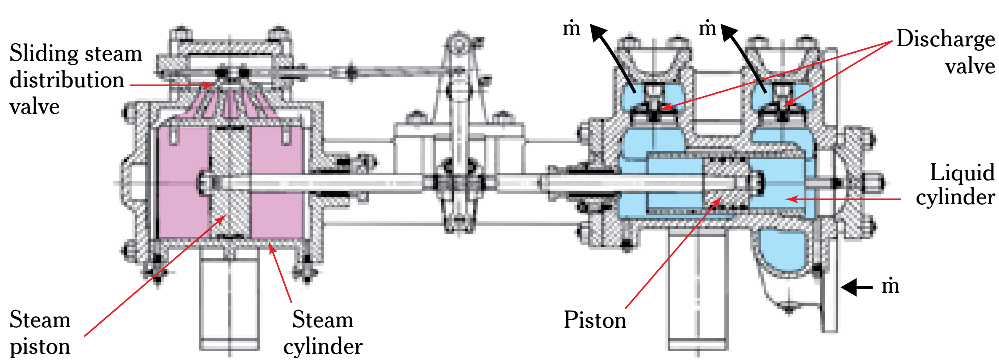


Fig. 2.6
Cross section of auxiliary pump in horizontal arrangement which is driven by the steam.

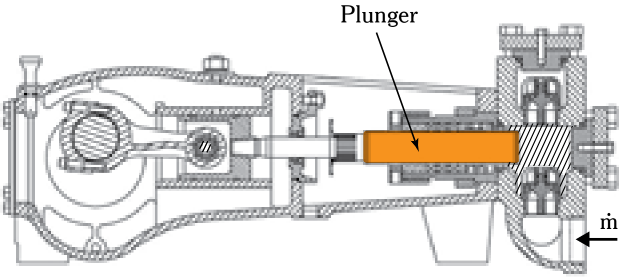


Fig. 2.7
Crank piston-plunger pump.

which is used for oil transfer may, in another pipe line system, be used for transferring other kinds of liquid, or for servicing (as feed pump) a machine.

The main parts of piston pumps are:

1) The ***cylinder with the cylinder liner*** or ***the block*** (or body) of the cylinder, where the increasing of discharge pressure is performed, resulting in operation of constant stress conditions. Usually, in horizontal arrangement pumps, there are suction and discharge ports on the cylinder block, while, in the vertical arrangement pumps, the suction and discharge valves are located on the cylinder head.

The cylinder block is usually made of cast iron or brass, and cast steel when the pumps are to be used at high pressures. For the pumps with more than one piston, the cylinder block is constructed with several ports, equal to the number of pistons. Where liners are used, they are mounted internally in the cylinder block, and have a slightly larger length than the piston stroke, to achieve the smooth operation of the pump. The cylinder wear liner is usually of Ni-resist material, or other corrosion resistant materials.

2) The ***pistons***, which, depending on their design or their mode of operation, can be divided into:

a) Submerging pistons (or plunger) that have a length much longer than their diameter, and their construction is solid or hollow. The sealing of these pistons, because of their great length, is ensured by a suitable gland (see par. 9.1.1) located on the cylinder at the inlet side of the piston. Their construction material is brass or stainless steel.

b) Disk-shaped pistons which have shorter length than their diameter and they owe their name to their shape. They are designed in various types, of which the three most common are the following:

– The ***body-and-follower type*** of piston [fig. 2.8(a)]. For their operation and sealing, soft fibrous packing or hard-formed composition rings (e.g. ebonite) are used.
– The ***solid piston type*** or a body and follower with rings of cast iron, brass or other materials [fig. 2.8(b)]. This type is commonly used in pumps handling oil, lube oil or other hydrocarbons. Their piston rings natural tension keeps them in contact with the cylinder liner, assisted by fluid pressure under the ring.
– The ***cup piston type***, which consists of a body-and-follower type of piston with molded cups [fig. 2.8(c)]. The ring construction material is rubber reinforced with fabric or other synthetic material.

3) The ***valves***, which control the flow of the liquid handled by the pump, and are divided into ***suction*** and ***discharge valves***. Valves are irreversible (check or non-return), i.e. they allow the liquid flow in one direction and close when the liquid flows

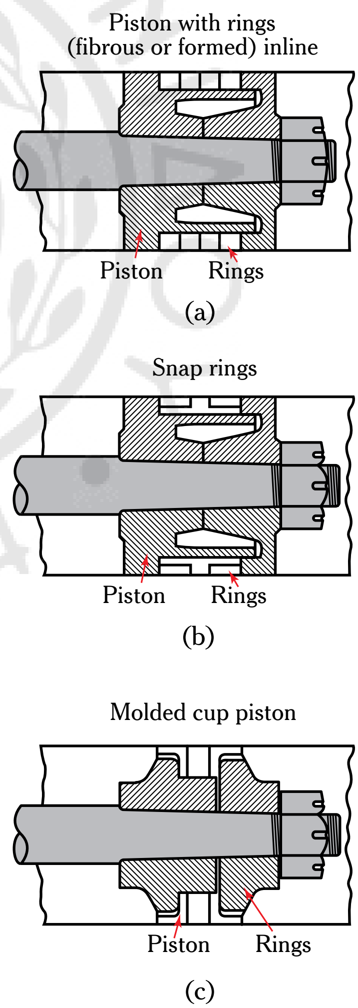


Fig. 2.8
Disk shaped piston types.

in the opposite direction. Keeping the valve in the closed position is achieved either by the liquid pressure presented in the piping system, or by the spring tension which is applied to the valve surface and remains constant by a suitable arrangement. In their simplest form, i.e., when they open and close with liquid pressure, valves operate:

a) By vacuum creation, which opens the suction valve.

b) By increasing the pressure of the liquid, as it is compressed from the piston and opens the discharge valve.

c) Both valves are kept closed when there is no pressure.

As shown in figure 2.9, the valve is closed (and be in equilibrium) when the forces acting above and below the valve are in balance. The balance is expressed by the relation:

$$p_1 \cdot A_1 = p_2 \cdot A_2 + S_f + M \qquad (2.1)$$

where: p_1 the pressure below the valve, A_1 the area below the valve exposed to p_1, p_2 the pressure above the valve, A_2 the area above the valve exposed to p_2, S_f the force of the valve spring (if any) and M the mass of the valve and half the mass of the spring.

The side surface of the cylinder formed by the

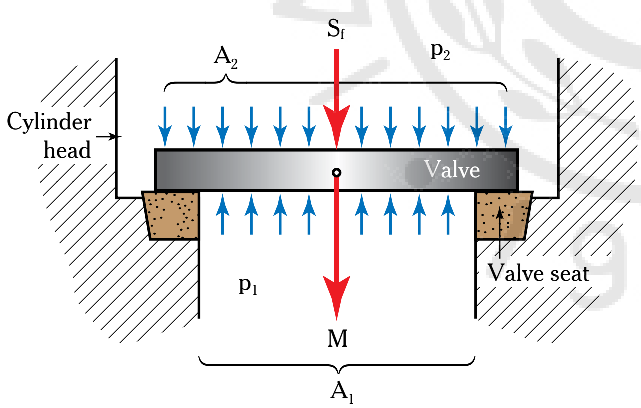


Fig. 2.9
Force balance of a typical valve.

elevation of the valve is called ***spill area*** and is the area through which the liquid flows. It is defined by the manufacturer and depends on the elevation (h) of the valve. Theoretically it is equal to ¼ D, where D is the diameter of the valve seat, so for:

$$E_{ssv} = \pi \cdot D \cdot h \qquad (2.2)$$

$$E_{ss} = \pi \cdot \frac{D^2}{4} \qquad (2.2a)$$

$$\text{from (2.2)(2.2a)} \Rightarrow \pi \cdot D \cdot h = \pi \frac{D^2}{4} \Rightarrow h = \frac{D}{4} \qquad (2.2b)$$

where: E_{ssv} the side surface of the valve and E_{ss} the surface of the seat.

The spill area is different among the various valve types, and given in figure 2.10.

In order to reduce the losses due to friction and the swirling present in the liquid by diverting the flow vein, as it passes from the spill area, the elevation of the valve is calculated larger to facilitate the flow of the liquid, and, depending on the valve type, is equal to $0.36 \cdot D$ to $0.40 \cdot D$, instead of $0.25 \cdot D$ of the relation (2.2b).

Piston pumps, depending on their application and the type of liquid that they handle, may be provided with different types of valves (fig. 2.10). Detailed types of such valves are:

a) The ***plate*** (or disk) ***valves*** [fig. 2.10(a)], which are for general use, and intended for the transferring of clean liquids. These valves can be operated at low Net Positive Inlet Pressure NPIPr,[2] and are suitable for piston drive shaft rotation speeds that are more than 300 rpm. The operating pressure is limited to 350 barg and they are available at mass production[3]. The horizontal pumps with unified cylinder cover usually have plate valves for the passage of liquid.

b) Valves with ***wing guide*** [fig. 2.10(b)], which are designed for light or heavy duty and used in respective applications. These valves can be operated with low NPIPr and are suitable for any rotation

[2] The ***Net Positive Inlet Pressure*** refers to the conditions that prevail in the suction of positive displacement pumps. It is equivalent to the meaning of Net Positive Suction Head, which is used for dynamic pumps, but in this case the pump manometric head is not used because the efficiency as well as all the parameters of the positive displacement pumps are expressed in pressure. Correspondingly to the required and available Net Positive Suction Head, there is also the required and available Net Positive Inlet Pressure, denoted as NPIPr and NPIPa respectively.

[3] The ***Barg*** (Bar gage) is used as an indication of the reading pressure relative to the current atmospheric pressure. The index g means the pressure indicated on to the pressure gauge. Applies $p_{abs} = p_g + p_{atm}$ or $p_g = p_{abs} - p_{atm}$, where p_{abs} is absolute pressure relative to absolute vacuum. The measurements of absolute pressure are defined as bara.

speeds of the pump shaft[4]. Their operating pressure may be up to 550 barg, and, with some of these, better seal can be achieved when used in handling fluid which contains suspended solid particles in an amount of 5% by volume, since they have elastomeric material at the contact points with the valve seat.

c) The ***ball*** (or semispherical) type valves [fig. 2.10(c)], which are suitable for fluids with particles, clean liquids and for pressures that reach up to 2 000 barg. Their operation is for pumps with shaft rotational speed below 300 rpm.

d) The ***plug*** (or piston) valves [fig. 2.10(d)], which are heavy duty valves for chemicals and pure liquids, and are placed in pumps with medium characteristics of NPIPr. Usually they are used for pumps with shaft rotational speed below 350 rpm, and in applications where the pressure is up to 550 barg.

e) The ***valves for handling viscous liquids and slurry mortars*** [fig. 2.10(e)], which are specifically used in pumps for the transfer of sludge, sewage and oil. They can operate at high NPIPr, and should operate in pumps with shaft rotational speed below 200 rpm and pressures up to 150 barg.

In the upper surface for some types of valves, there is a fixed protrusion, which is called ***inhibitor*** (fig. 2.11). The purpose of the inhibitor is to adjust the maximum opening of the valve at the desired limits, and this is achieved by an added piece and an adjusting screw suitably installed on the external side of the cylinder head. In the lower side of the

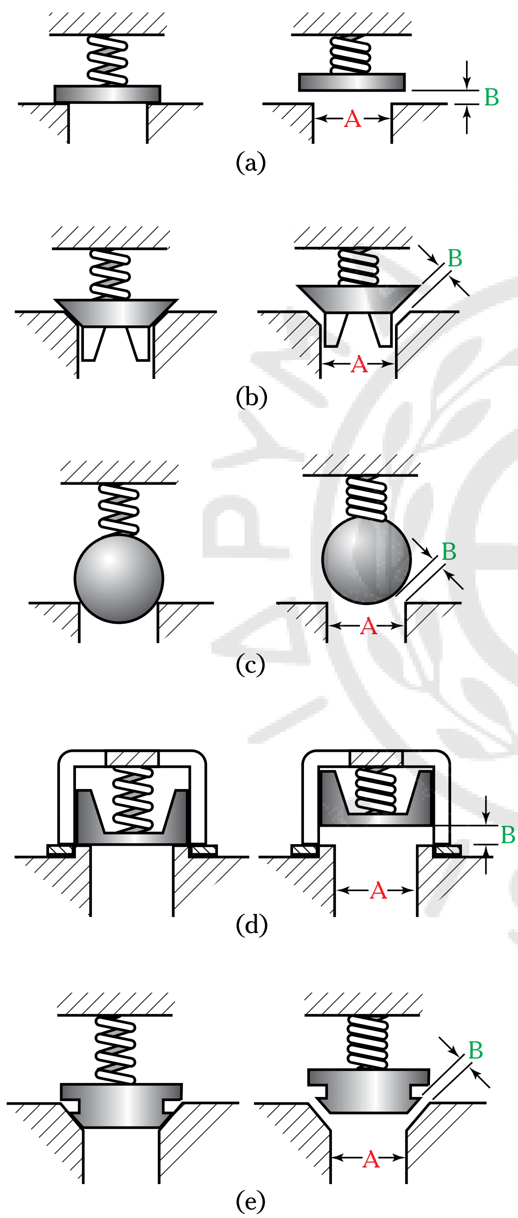


Fig. 2.10
Valves of piston pumps. Spill area for each valve type. (Where A = cross section of valve seat, B = spill area).

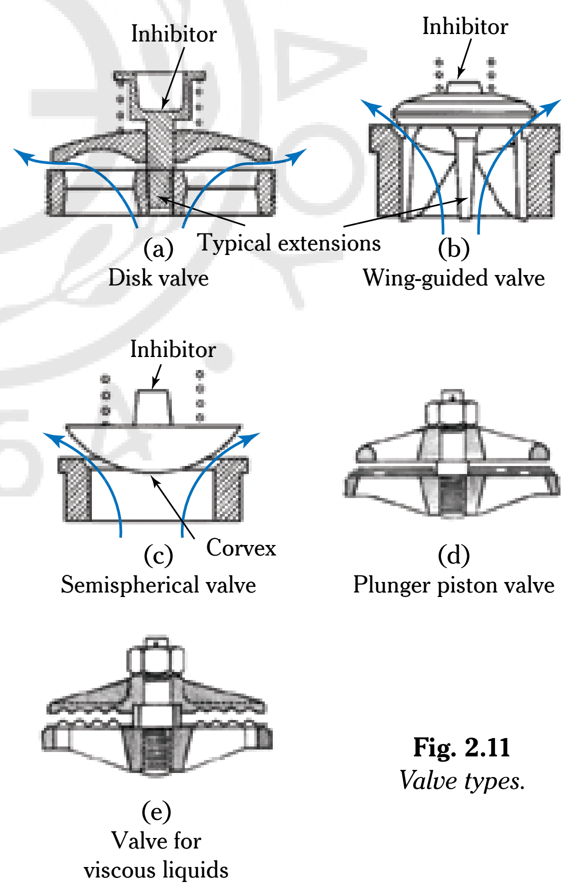


Fig. 2.11
Valve types.

[4] They are not available in mass production but are produced by the manufacturers of the specific pumps.

valve, there may be a guide which either has the form of a central axis and moves within the cylindrical housing, or has the form of wings disposed circumferentially on its lower side, or has a convex surface to maintain the valve in the desired position, and to ensure the seal of the valve when it is closed (fig. 2.11).

The construction material of these valves is brass, while the plate valve may be made of rubber or leather in various forms.

2.3 Accumulators.

The purpose of smoothing the fluctuations in flow rate which are created in a piping system, when piston pumps are used to transfer liquid, can be reached by the installation of specific devices named accumulators, air chambers or side branch accumulators. These devices allow the liquid flow into the pipes of a hydraulic system when the energy of the piston of the pump, which acts and pushes the liquid to move, stops. The interruptions to the energy of the piston of the pump as well as of the liquid flow into the piping system result from the piston move as it moves, for example, from TDC to BDC on a single acting piston pump. However, when an accumulator is installed in the piping system, the sudden pressure reduction (by the sudden interruption of flow) is averted, while, simultaneously, the losses of the part of the energy that is given to the liquid in every effective piston stroke are avoided. An effective piston stroke is defined as any piston stroke where there is liquid discharge to the piping system.

The accumulators consist of a metal chamber with sufficient volume and contain air under controlled pressure. They can be located either between the discharge valve and the connection pipe flange of the pump piping system, or between the connection pipe flange to the piping system and the suction valve (when the liquid to the suction side is under pressure), or they can be placed on both sides. The volume of liquid that exists inside the accumulator can increase or decrease as it enters or exits from the openings on the bottom of the chamber, which is connected to the piping.

By the discharge of the pump, during its operation, the liquid supplied to the pipe system meets some resistance that is owed either to the filling of pipes from the liquid (because of the finite cross section of the pipes or the volume of the liquid they may contain), or to the discharge height which can be reached, or the pressure that may exist in the discharge side. So, due to the resistances which the liquid meets as it is discharged to the system (by frictions or pipe fittings), the result is that a quantity is not passing to the pipes and as a consequence increases in accordance to the discharge pressure. This causes an amount of the liquid, when the discharge pressure exceeds the pressure of the air in the accumulator, to be provided within the accumulator and not to the pipes, because the discharge pressure prevails over the air pressure. The increase of air pressure inside the accumulator becomes similar in value with the pressure in the discharge valve (fig. 2.12). The air volume change as a function to the pressure change is given by the equation:

$$V = \frac{m \cdot R \cdot T}{p} \tag{2.3}$$

where: V the volume of air, p the pressure of the air, m the mass of air, R is the gas constant[5] (air), and T the absolute temperature of the air.

When the piston has reached the end of the effective stroke, after the flow is interrupted by the pump, and it starts moving in the opposed direction, the air under pressure is expanded and the concentrated liquid from the air accumulator returns to the piping system, achieving continuous flow. This flow exists until the pressure inside the accumulator is equal-

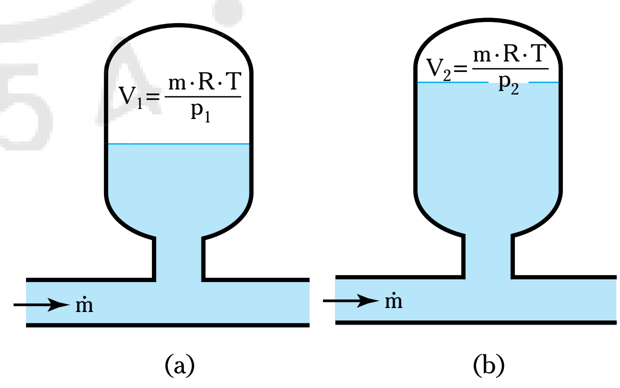


Fig. 2.12
Change of the liquid level inside accumulators and the change of the gas volume in relation to the change in pressure where $p_1<p_2$.

[5] The ***gas constant*** (or ideal gas constant, R) is a physical constant which is featured in many fundamental equations such as the ideal gas law.

ized with the pressure of the height of the liquid column on the discharge system. In an open system, flow continues until the pressure becomes equal to the pressure of the free surface of the tank which is under atmospheric pressure or, in a closed network, it becomes equal to the pressure of the system where the liquid is discharged.

In air accumulators where the operating pressure is regulated by compressed air supply to the chamber of the device, the air pressure should not exceed the pressure of the liquid column of the system, because under these conditions air enters to the system, and this results in decrease in the pump efficiency.

By adding the air accumulator to the pump discharge, a smooth and continuous flow is provided, protecting the piping system against the hydraulic hammer (see par. 4.4), which could be caused by sudden fluctuations in fluid pressure. On the other hand, placing the device on the suction side achieves the smooth introduction of the liquid in the pump cylinder, preventing the valve knocks[6], and the strain on the cylinder head.

The liquid level in large accumulators is monitored by a tubular type level gage. This indicator is made of glass or reinforced synthetic transparent tube, and is mounted vertically on the side of the chamber. One end of the tube is connected to the upper side of the chamber at a point where air is entering the tube, and the other to a trough from which water enters so as to form a close loop causing the tank liquid to seek its level in the gauge. Thus, inside the tube one can read directly the corresponding liquid level of the chamber [fig. 2.13(a)]. Additionally, to measure the pressure and to maintain the desired pressure, a pressure gauge and a pressure relieve valve are used accordingly. The pressure relieve valve, by means of a spring with adjustable tension, will open when the pressure inside the chamber is increased beyond the desired one. It is noted that the setting of the pressure relief valve shall not exceed the discharge pressure of the pump, because excessive pressure increases can cause damages to the pump and the piping system.

The total volume of the accumulators is 2-4 times greater than the volume of the pump cylinder, or may depend on the length of the piping system in which it is installed. The chamber is typically 1.2 -1.5 times the volume that is displaced from the piston when it moves inside the cylinder. When centrifugal

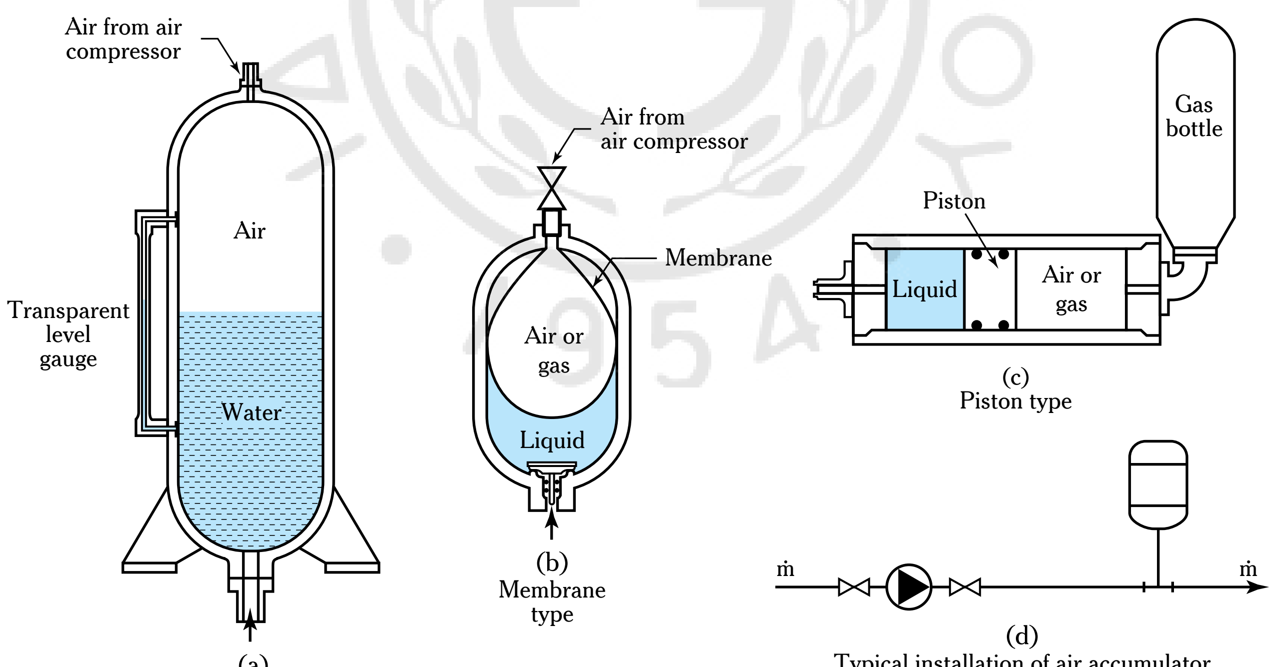


Fig. 2.13
Types of accumulators.

[6] ***Valve knock*** is the phenomenon which is caused by the sudden pulsations on fluid pressure as fluid flows through valves and results in the valve closing forcefully against the seat, so as to produce an audible sound.

pumps are installed in the piping system, the size of the devices depends on the system size. ***Accumulators are used to maintain pressure inside the piping system without the need for the pump to be in operation.*** By these devices we achieve saving of energy because the continuous operation of the pump is avoided and furthermore the wear is reduced. Accumulators can also be used to install pumps with less power in piping systems in order to provide liquid with pressure (fig. 2.13).

2.4 Piston pumps flow curves.

Piston pumps are used in systems where high discharge pressures are required, or for liquid transfer at high altitude while simultaneously providing a wide flow range from very small to very big heights.

The pulsatile flow rate of the liquid to the discharge of the pump and into the piping system is produced by the reciprocating motion that characterizes piston pumps, and is significantly affected by the piston number or the utilized pump type, e.g., direct transmission or power pumps.

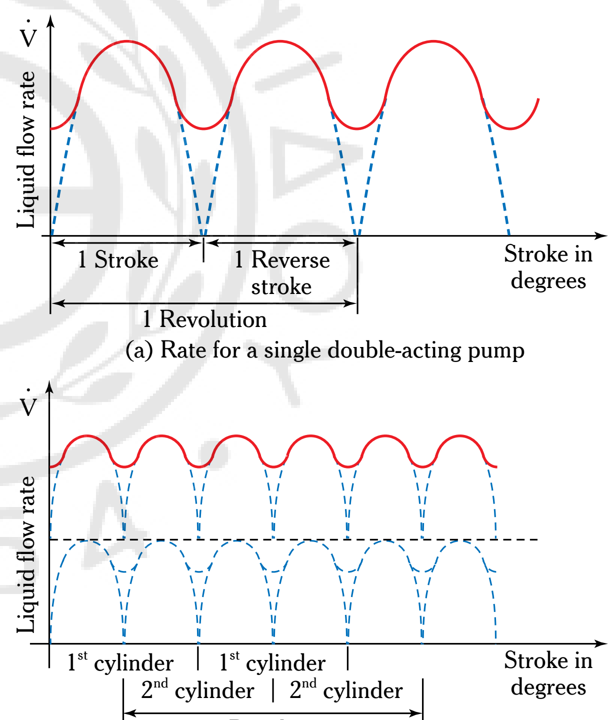
The pulsations or fluctuations in the discharge flow are due to the high energy potential that is produced by the pump at normal speed conditions, when the resistance on the system where the liquid is provided reacts with the flow, as the pressure is created. But given that the magnitude of the discharge pulsation is mostly affected by the number of cylinders, increasing the number of cylinders will reduce the flow pulsations[7].

Moreover, reduction in the pulsation or fluctuation intensity is achieved by adding an air chamber to the discharge pipeline of the pump. With the employment of an air chamber, the drop of the flow, which will theoretically stop upon completion of the discharge due to the reversal of the direction of the piston, is smoothed.

In direct acting steam piston pumps with single double-acting piston (***simplex*** pump), at normal operating speed conditions, the liquid flow rate is constant until nearly the end of the piston stroke, where the piston stops and reverses its direction. As liquid is flowing to the discharge without the air chamber, the flow rate will have the form of the dashed (blue) line in figure 2.14(a). The pulsation on the flow will be continuing until zeroed, and then it increases, due to the liquid discharging from the other side of the piston. However, by the existence of an air chamber on the discharge side, the flow rate follows the red solid line, as illustrated in figure 2.14(a), where the pulsations are considerably less.

On coupled configuration steam pumps (***duplex*** pump) with double-acting pistons, due to the phase difference (by half a stroke) of the liquid discharge from the cylinders, we will have the form which is illustrated in figure 2.14(b). The characteristic curves on discharge flow rate, for pumps with two cylinders and double-acting pistons, are the sum of the height of the curves. So, the flow rate will follow the solid line of the graph, whilst the fluctuations can be further reduced by applying the air chamber (accumulator).

Comparing the graphs in figure 2.14, it is shown that while the characteristic curves between the



(a) Rate for a single double-acting pump

(b) Discharge rate in a duplex pump with double-acting piston

Fig. 2.14
Discharge flow rate in simplex and duplex steam pumps, and frequency of pulses per revolution of the crankshaft.

[7] For example, when one piston is used, the fluctuations are high due to the interruptions of the discharged liquid as the plunger moves toward BDC, with two pistons the fluctuations are reduced because as the discharge of one piston is interrupted the discharge of the other piston begins, with three pistons it will be reduced further, and so on.

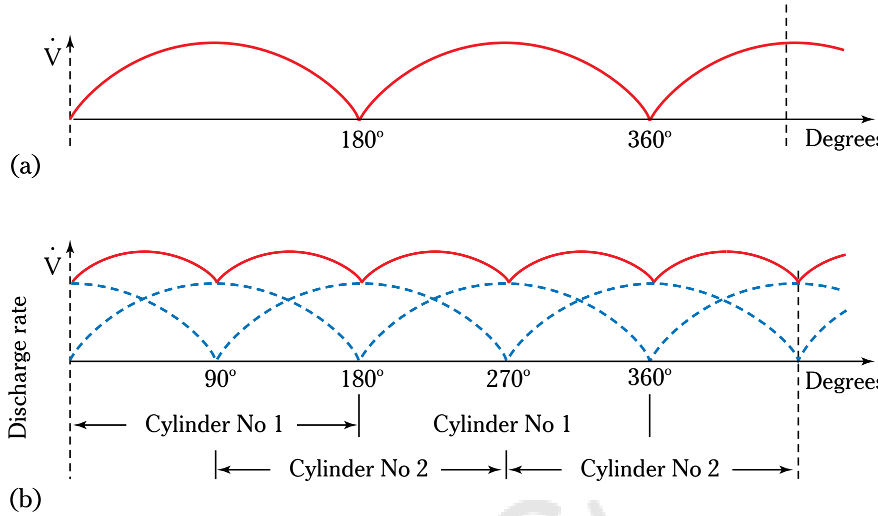


Fig. 2.15
Flow rate on power pumps.

pumps with two pistons and one piston have doubled the fluctuations, the minimum point of each fluctuation for the two piston pump is always higher than the minimum point of the pump with one piston.

In power pumps, the flow rate fluctuations are caused by the movement that is transmitted from the rotating crankshaft to the piston, through the piston rod. Considering the instantaneous piston movements, it is observed that they are not proportional to the instantaneous crank rotation angles. Then, as the movement of the crank (which is driven at constant speed) is converted to piston linear motion, it always creates a variable flow rate, since the movement of the piston is accelerated and decelerated.

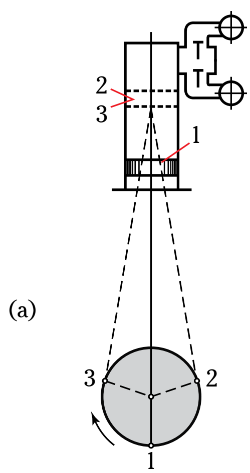
The principle movement of the crankshaft drives the piston to begin its motion at low speed, then to accelerate until reaching the maximum speed in the middle of each half stroke, and slow down to zero at the top and bottom dead centers of each stroke.

Furthermore, the acceleration and the speed of the piston determines the flow rate of the fluid that is discharged from the cylinder.

Thus, the amount of liquid-mass flow rate that is discharged by a ***simplex double-acting pump*** (reciprocating piston pump) has the form of figure 2.15(a). For single-cylinder single-acting pumps, the flow diagram of discharge rate is similar, except that after each pump stroke, where the suction stroke is realized, the flow is zero, and it will coincide to the horizontal line of the diagram.

Accordingly, in a ***duplex double-acting pump*** (piston pump with two double acting pistons), due to the difference in the liquid discharge phase, the liquid discharge flow rate has the form of a continuous line which is the sum of the depression curve of the two cylinders in figure 2.15(b).

For a triplex single-acting pump, with ***three single-acting pistons*** which are installed in three parallel cylinders and their connecting rods are connected in a common shaft with phase difference 120° degrees [fig. 2.16(a)], the discharge rate has the form of the continuous line illustrated in figure 2.16(b).



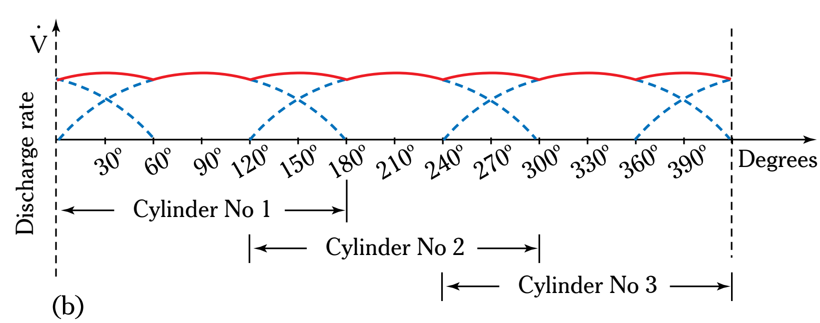


Fig. 2.16
(a) Piston positions during the rotation of the crankshaft.
(b) Discharge rate in a pump with three single-acting pistons.

The discharge from each movement of the piston, in each stroke, is represented by the sinusoidal curves. But due to the difference in timing of the cylinders (due to the difference of 120° degrees), the total discharge flow of the pump shows slight fluctuations. A similar form is shown in the diagram of figure 2.17, which refers to the fluctuations of the discharge rate of a pump, when the pistons are double-acting, as well as in the diagram of figure 2.16, where a pump operates with three single-acting pistons. Hence, it is established that the curve of liquid discharge flow rate of a pump with more cylinders is smoother, so the necessity of the air chamber installation is eliminated.

2.5 Theoretical and actual flow of piston pumps.

The ***theoretical flow rate*** of a piston pump depends on the type and number of the pistons. So:

1) The theoretical flow rate $\dot{V}_{th}$ for a single-acting pump is given by:

$$\dot{V}_{th} = \frac{\pi}{4} \cdot D^2 \cdot s \cdot n \tag{2.4}$$

where: D is the internal diameter of the cylinder, s the piston stroke and n the number of reciprocations of the piston. The $\pi / 4 \cdot D^2 = A$, represents the effective surface area of the piston, which in relation (2.4) can be written as A, and the flow is given as:

$$\dot{V}_{th} = A \cdot s \cdot n \tag{2.5}$$

In piston pumps that are driven by a crankshaft, the s in relation (2.5) is equal to 2R and the relation (2.4) is given as:

$$\dot{V}_{th} = \frac{\pi}{4} \cdot D^2 \cdot 2\,R \cdot n \quad \text{or}$$

$$\dot{V}_{th} = \frac{\pi}{4} \cdot D^2 \cdot R \cdot n \tag{2.6}$$

2) For double-acting piston pumps the volume of displaced liquid in each stroke is:

$$V_1 = \frac{\pi}{4} \cdot D^2 \cdot s \tag{2.6a}$$

for one piston move from BDC to TDC, while in the opposite direction because of the piston rod surface area, the volume of displaced liquid is given by:

$$V_2 = \frac{\pi}{4} \cdot (D^2 - d^2) \cdot s \tag{2.6b}$$

where d is the diameter of the piston rod, so that the active surface area of the piston is given as:

$$A = \frac{\pi}{4} \cdot (2D^2 - d^2) \cdot s \tag{2.6c}$$

and the theoretical flow is given by the relationship:

$$\dot{V}_{th} = \frac{\pi}{4} \cdot (2D^2 - d^2) \cdot s \cdot n \tag{2.7}$$

or
$$\dot{V}_{th} = (2A - a) \cdot s \cdot n. \tag{2.7a}$$

where a is the cross-sectional area of the piston rod.

3) The total flow rate of the pumps with more cylinders, where the pistons are driven by the same crank shaft, is calculated by multiplying the flow relation of a cylinder with the number of pistons.

The theoretical flow rate arises from the geometrical characteristics of the pump, without counting the various losses which are caused by leakages, because valves where the liquid passes through are actually not completely sealed, and leakages also occur in the cylinders where the piston reciprocates, the glands, etc. Therefore, because of the leakage, the actual flow rate ($\dot{V}$) is lower than the theoretical ($\dot{V}_{th}$).

These losses are measured by the volumetric degree of pump efficiency, which is the ratio of actual to theoretical flow rate [relation (2.8)]. Thus, taking into account the volumetric efficiency η_V, ***the ac-***

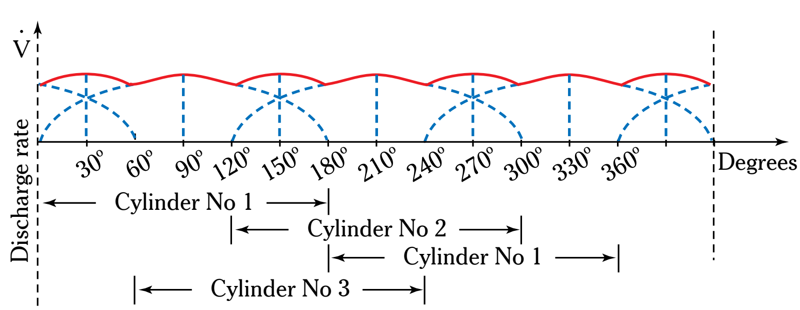


Fig. 2.17
Discharge rate in a pump with three double-acting pistons.

tual flow rate of the pump is expressed as:

$$\eta_V = \frac{\dot{V}}{\dot{V}_{th}}$$

or $$\dot{V} = \eta_V \cdot \dot{V}_{th} \tag{2.8}$$

The volumetric efficiency ranges between 0.7 and 0.97, depending on the size and type of pump.

2.6 Power transmission in piston pumps.

Typically, piston pumps that are used on ships are divided according to the available energy consumed (and transmitted) by the driving machine into: steam-driven direct acting piston-type pumps; crank piston or motor-driven plunger-type reciprocating pumps with a system of crankshaft –connecting rod– yoke - piston rod; and diaphragm pumps powered mechanically or pneumatically.

2.6.1 Steam-driven direct acting piston-type pumps – Operation.

The required energy for the operation of direct acting (or direct transmission) piston pumps is provided by the steam entering the cylinder where the corresponding steam piston reciprocates (fig. 2.6). Thus, the steam cylinder forms the drive (motor mechanism) of the pump. The steam piston, through the piston rod, is connected with the liquid piston of the pump and, in turn, reciprocates in the corresponding cylinder, where by suction and discharge the transfer of the liquid is achieved. The parts that direct acting piston pumps are composed of:

1) The ***steam cylinder***, in horizontal or vertical arrangement, on the body of which piping connections are properly fitted for steam inlet and outlet, for the piston movement. However, when more steam cylinders are used for driving the pump, they have parallel arrangement to each other, and can be installed horizontally or vertically in space. Also, each cylinder has the corresponding steam inlet and discharge connections.

2) The ***steam piston***, with the piston rings which are placed circumferentially of the piston in order for the cylinder compression space to be sealed.

3) The ***piston rod*** (or direct transmission rod). The steam piston of the drive cylinder is connected on the one end of the rod. The compression piston (or liquid piston) of the pump cylinder is connected on the other end of the rod.

4) The ***inlet and outlet valves*** of operating steam.

5) The ***control unit*** for the supply of operating steam, where, by steam valve movement, at the correct order, the reciprocating movement is produced.

6) The ***cylinder of liquid transfer*** of the pump, which could be in a horizontal or vertical layout. When two or more steam pistons are used, two or more pistons will be used for the liquid transfer respectively. They also are installed parallel in horizontal or vertical arrangement.

7) The ***liquid pressure piston***, with the respective piston rings.

8) The ***suction and discharge valves*** of the pump.

9) The ***stuffing boxes*** (glands) with ***packings***, for the sealing of the piston rod at the entry points to the steam cylinder and the liquid pressure cylinder.

During the operation of a steam pump with double-acting piston, the movement of the piston is achieved by providing adequate steam pressure into the cylinder on one side of the piston, while steam exits simultaneously from the other side of the cylinder. The steam expansion, which could affect the pump performance, is too small as the steam enters into the cylinder space, since it is provided at a constant rate throughout the operating cycle.

The collision (or striking) of the moving parts onto the stable parts, which could occur due to the lack of mechanical terminal end on each piston stroke (i.e. the steam piston or the liquid piston and their piston rod collide on the stationary portions of the cylinder or the cylinder head), is avoided by regulating the steam distribution in the cylinder. In this way, a reduction in the intensity of the piston movement is achieved. This is performed by the piston, which, as it approaches the end of its stroke, covers the steam outlet ports so that a quantity of the steam is trapped into the space between the piston and the cylinder body.

Thus, on the steam which has been trapped, the pressure is increased, and, as a result, it acts as a brake on the movement of the piston until it stops. After the short pause of the piston movement at the end of its stroke, a new quantity of steam is supplied through the steam inlet valve, which begins to push the plunger in the opposite direction.

The supply to the steam valves is controlled by the ***steam distribution mechanism*** (fig. 2.18), which governs the reciprocating movement. The operation of the steam distribution mechanism differs among various types of pumps, and depends on the number of cylinders. These mechanisms are:

1) The ***simplex-type*** valve or steam distribution mechanism in simplex pumps. In this distribution mechanism, the main steam valve that supplies steam to the cylinders is controlled by the shaft of the pilot valve of the distribution mechanism, which is connected on the piston rod of the same steam cylinder where the piston reciprocates. Consequently, the control of the main piston steam valve is not performed directly by the movement of the piston rod, but through the pilot valve which is operated by a coupled connection to the piston rod. So, the movement of the pilot valve, at regular intervals, regulates the flow of the steam and provides the steam alternately at both ends of the piston of the ***main steam valve***. The arrangement illustrated in figure 2.18 is one of the mechanisms available to produce this operation.

With the pilot valve in the right position, as shown in figure 2.18, the operating steam from the piping system flows through the pilot valve and steam port A into the space at the left, and shifts the slide valve piston in the right position, while

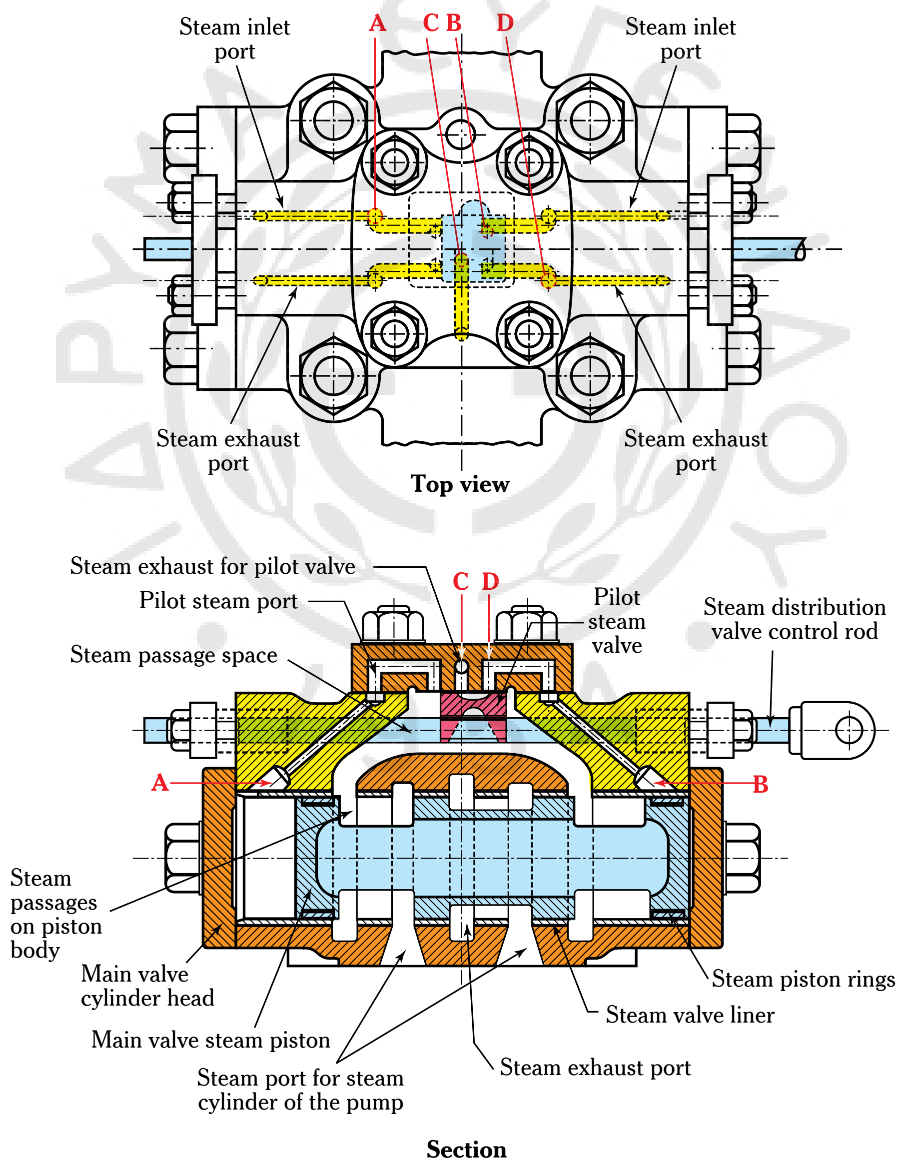


Fig. 2.18
Steam distribution mechanism.

the pilot valve interrupts the extraction of steam from port C. By moving the main valve piston to the right, communication is achieved between the steam chambers on the valve body (or chest), and so, the steam passing through the chambers enters into the steam cylinder from the left port of the distribution mechanism.

Simultaneously, the pilot valve interrupts the flow of steam from port B, and the D section of the pilot valve connects the steam space at the right-hand end of the main valve with the exhaust port, thereby releasing the trapped steam. The piston of the main valve has moved completely across to the right end of the chest.

With the piston of the main valve in this position, steam is allowed to flow from the chest to the left steam cylinder port, for the move of the steam cylinder piston. Also, in this position, the main valve piston, due to its shape, creates passage chambers that connect the lower right port with the steam exhaust port, through which the steam presented into the cylinder is released. The steam piston of the pump now moves to the right and, through the coupling mechanism to the piston rod, the pilot valve moves to the left.

In this position, the cycle previously described now takes place at the opposite end of the steam chest (fig. 2.18). There, the steam flows to the right end of the main valve and the operating steam passing through the ports moves the piston of the steam cylinder of the pump to the opposite direction. Because the main valve is steam operated, it can only be in two positions, either at the left side or at the right side end of the chest, without it being possible to have an intermediate dead center. Hence, steam can always flow either to one side or to the other of the steam piston, regardless of the position of the steam piston. The adjustments of the valve are outside the steam chest, through (regulatory) nuts that are screwed onto the thread of the pilot valve control rod. So, the possibility is given to adjust the valve while the pump is in operation.

2) The ***duplex*** steam valve distribution (or actuating) mechanism. It relates to pumps having two cylinders, arranged in parallel, which operate together as a single unit.

In these pumps, the piston rod of one cylinder, during the operation cycle, affects the steam valve which controls the inlet or outlet of the steam to the other cylinder (fig. 2.19).

The ***Y-shaped*** lever (wishbone-shaped) is used to transmit the motion of the piston rod to the tappet (or control rod) of the distribution mechanism. The Y lever of the piston rod of the one cylinder is connected to the control tappet of the steam supply valve of the other cylinder and, through another

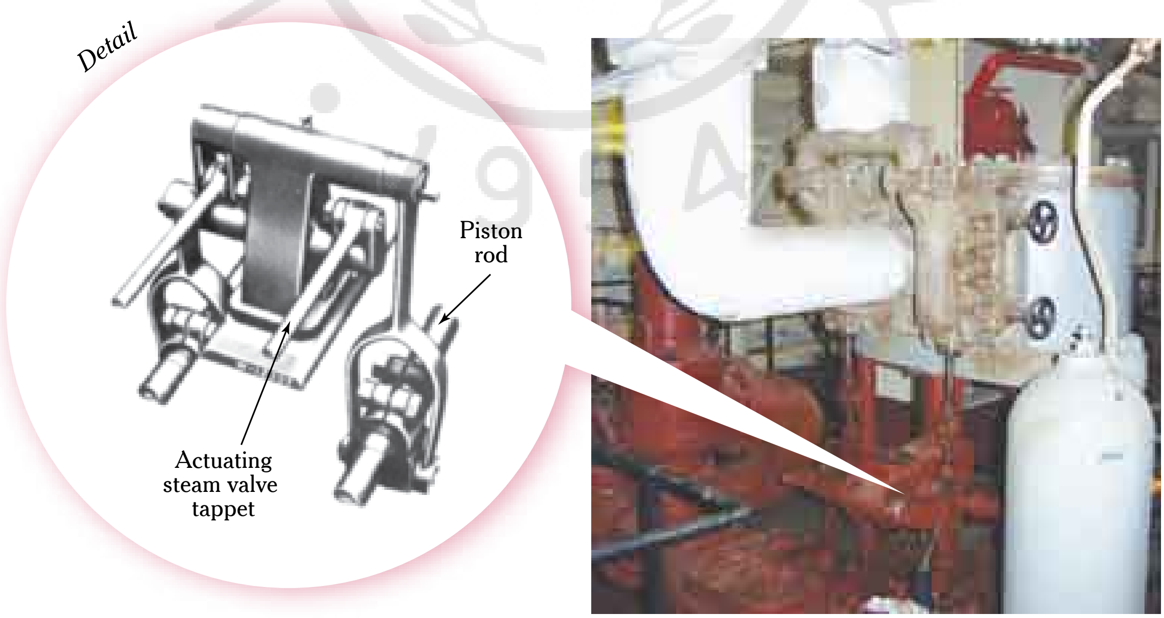


Fig. 2.19
Coupling at piston rods. Steam valve actuating (distribution) mechanism or a duplex pump.

Y lever, a corresponding connection is performed on the piston rod of the first cylinder. Thus, when the first piston has completed its stroke, it must be stopped until the steam valve is actuated from the movement of the rod piston of the second piston, and thereby controls the admission or exhaust of steam in the second pump. Because one of the two steam cylinder ports in the valves is always open, there is no dead center condition, and the pump is always ready to start when steam is supplied to the steam chest. The movements of both pistons are synchronized to provide a well-regulated flow of liquid without excessive fluctuations and interruptions.

3) The mechanism of ***flat slide steam valves***. In these valves, the steam enters the pump through the two outer ports at the top of the steam cylinder and is extracted from the central ports (fig. 2.20). Valves of this type are used in most reciprocating pumps. The D-type steam valve of the steam ports slides on the flat part (mirror) on the cylinder of the steam distribution valve mechanism. In the chest of the distributing valve are the steam ports, and so the yoke[8] of the cylinder by the piston rod of the pump plunger can move the slide valve of the other cylinder (fig. 2.19). The steam pressure acting upon on entire top area of the valve maintains the D-type valve in contact with the mirror of the steam ports; this is called an unbalanced valve.

Flat slide valves are used in operating steam pressure up to 17 bar, and their setting is achieved by a rod with threaded and adjusting nuts (fig. 2.20). Due to friction on the surfaces, the duration of their efficient operation can be extended by lubricating the abutting surfaces. On large pumps, the force required to move an unbalanced valve is considerable, so it is anticipated that the force required to initiate this type of valve will rise accordingly, causing instability in their operation. Therefore, on large pumps other types of distribution mechanisms, i.e. balanced piston steam valves that have greater stability, are used.

The balanced piston steam valve (fig. 2.21) is

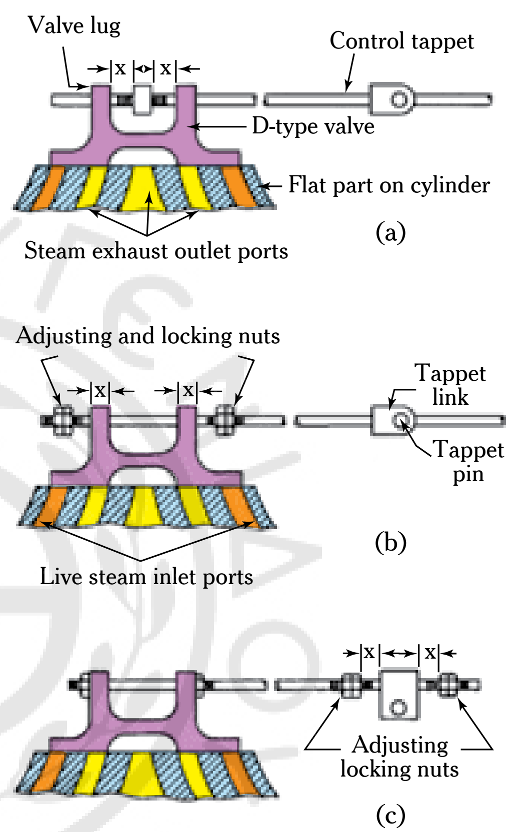


Fig. 2.20
Duplex pump sliding steam valve, and motion arrangement.

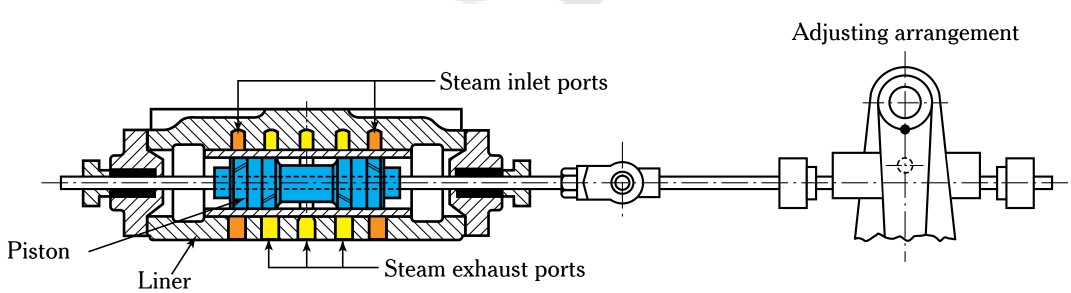


Fig. 2.21
Sliding high pressure steam balanced piston valve.

[8] ***Yoke*** is the link or shaft used for joining or adapting parts to a mechanism (e.g. the yoke between the engine parts).

used on ***duplex steam pumps*** when the slide-type valve cannot be used because it does not provide the necessary balance to the movement of the pistons. The valve piston is shaped in such a manner as, depending on its position, to communicate the appropriate ports for the admission and the removal of the steam of the cylinder. The balanced piston valve can also be used without lubrication at pressure above 17 bar and temperatures above 260 °C. At much higher pressures, a steam cutting (or wire drawing) mechanical system can be used, which operates in conjunction with the piston as it slowly crosses the steam ports.

2.6.2 Power pumps.

The pumps in which the plunger used for the transfer of liquid moves through crankshaft, connecting rod and piston are called ***crank piston pumps***, motor-driven plunger-type reciprocating pumps (fig. 2.22). The rotary movement of the crankshaft becomes linear at the piston rod, while the piston may be single-acting, double acting, piston or plunger, depending on the pump type.

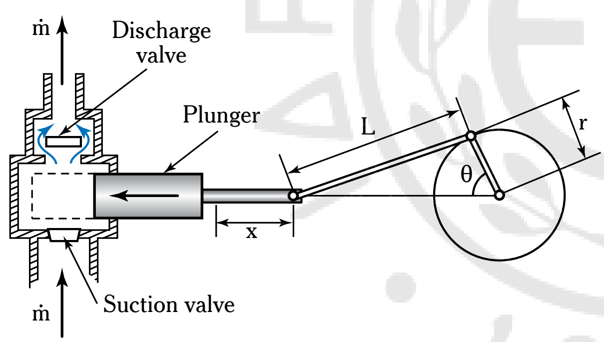


Fig. 2.22
Slider-crank mechanism in a power pump.

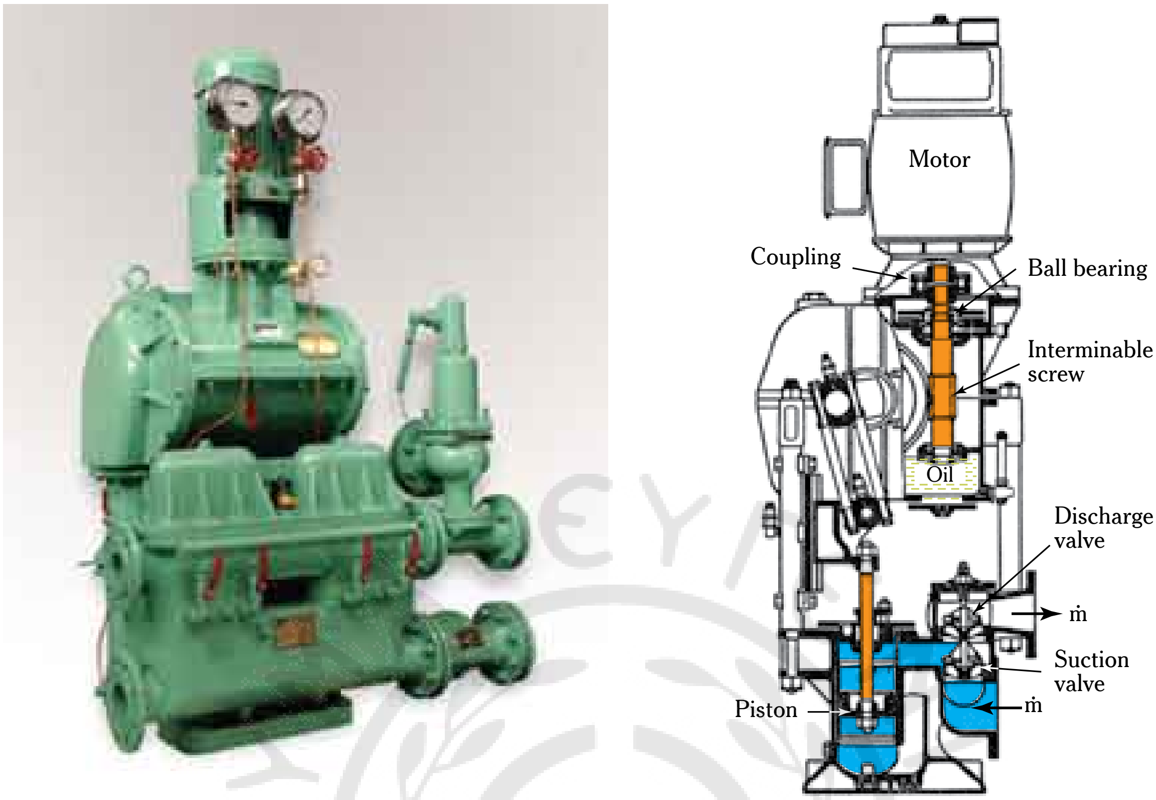
The flow of fluid through a positive displacement pump is performed by a repetitive transition state, called ***dynamic pumping cycle***. The event that initiates this cycle is the linear motion of the piston, and its movement is related to the pump efficiency.

During the pump operation, as the piston moves toward BDC and is “removed far” from the fluid movement cylinder head, the volume of the chamber formed inside the cylinder increases. Removal of the piston has the effect of reducing the pressure inside the cylinder as its volume increases. Since most of the fluids which are managed by positive displacement pumps are relatively incompressible, a very small piston movement is required to cause the pressure reduction. When pressure drops sufficiently in the cylinder, below the suction pressure, the difference in pressure acts in order to begin opening the suction valve. The valve opens gradually and smoothly at the beginning of the suction stroke, because the speed and acceleration of the piston are small. The fluid flows through the suction valve and, since it follows the movement of the piston, it fills the space that is created inside the cylinder. As the movement of the piston slows when approaching at the end of the suction stroke, the suction valve gradually returns to its seat and the flow rate of the fluid is interrupted. Ideally, the suction valve is fully closed, as the piston reaches the end of the suction stroke (BDC), and ***stops*** moving.

The movement through the sliding mechanism from the crankshaft causes the reversal in the direction of movement of the piston. So, after an instantaneous break in the BDC, it will start moving towards TDC for fluid discharging. The liquid, which is trapped inside the cylinder, is pressed, until the pressure in the cylinder sufficiently exceeds the existing pressure in the piping discharge system, so as to begin to open the discharge valve. The discharge valve remains open until the piston reaches the limit where its course will be completed, or until the flow rate of the fluid velocity through the valve is constant, when the pressure equals that of the discharge pressure system. As the piston speed is slowed, since it follows the rotational movement of the crankshaft to which is connected, the discharge valve moves back to its seat in the closed position and stops the liquid discharge. Similarly to the suction valve, the discharge valve is ideally closed when the piston movement stops once it reaches TDC.

The number of ***pumping cycles*** in a single rotation of the crankshaft, is equal to the number of cylinders that a pump has, with the pumping order of each cylinder being determined by the order that the pistons are connected on the crankshaft.

Therefore, the pumps can be single-cylinder or multi-cylinder, with the cylinders arranged in parallel, while the crankshaft may be positioned horizontally above the cylinders (fig. 2.23), or horizontally at the side of the horizontally arranged cylinders (fig. 2.24). Discharge of the liquid is carried out in a common discharge manifold in the pump body, on which the pipes are connected.



(a) (b)

Fig. 2.23
(a) Power pump shaft installed horizontally above the cylinders and (b) schematic representation and its parts.

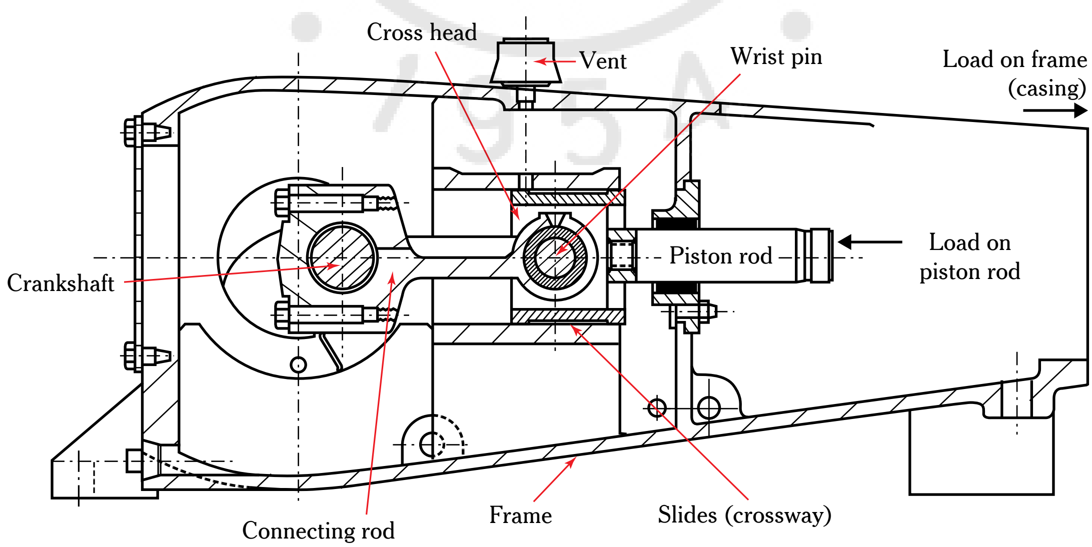


Fig. 2.24
Section of the power end in horizontal pump.

A common way of determining the movement of crank pumps in the industry is performed in accordance to the number of pistons connected to the crankshaft (tab. 2.1).

Table 2.1: *Industry terms for the number of plungers (or pistons) in power pump.*

Number of pistons	**Industry term**	**Number of pistons**	**Industry term**
1	Simplex	5	Quintuplex
2	Duplex	6	Sextuplex
3	Triplex	7	Septuplex
4	Quadruplex	9	Nonuplex

The pistons are made of cast iron, bronze, or steel sealing rings from reinforced rubber or other synthetic material, e.g. bakelite[9]. Their liners consist of Ni-resist[10] metal, which has properties that meet the pressure and temperature requirements, during operation. To align the movement of the piston, joints (commonly, cross) are usually used in the connection points of the connecting rod with the piston rods, which are suitably shaped so as to slide within guiding cylinders.

As mentioned, the movement of the pistons in power pumps is materialized through the piston rod, the cross head, the connecting rod, the crankshaft main bearings, and the bearings in the attachment points of the connecting rod to the crankshaft. All these together are enclosed in a casing, so that the aggregation can be called a ***power end*** (fig. 2.24). The load exerted on the elements (components) that comprise the power end is the load from the piston rod, and is equal to the product of the piston cross-sectional area multiplied by the maximum discharge pressure.

In an air compressor, or in a machine, this load is increased from 90° to 180° in crankshaft rotation before the maximum load is achieved. But, in a pump, the maximum load is reached in less than 30° crankshaft rotation by the relatively incompressible property of the liquid during pumping. Since this ***load cycle*** is repeated in every piston cycle, the pump charge is shown as a momentary increase in the high intensity load and should not be taken as a simple fatigue strength of the components (parts), which receive the load. Therefore, the design criteria of pump components, which receive the load from the piston rod, should include the ability of materials to absorb the high instantaneous load increase, with resistibility of their construction safety factor[11] greater than 3:1.

Another critical factor affecting the ability of the elements (components) of the power end to handle the load exerted by the piston rod is the manner in which this load is applied. So, depending on the pump arrangement, i.e., if it is vertical or horizontal, due to differences in the application of the load, the choice of material and the safety factor should be modified accordingly. For the movement of the crankshaft, and moreover of the pistons, electric motors are used, which transmit the movement to the crankshaft through an endless screw gear to a disc gear (peripheral serrated drive), or by pulley and belts. Applications of power pumps on ships are found in drinking water networks, sanitation, bilge-pumping systems, etc.

– Functional and constructional factors of power pumps.

The evaluation factors that are considered for choosing the installation of a power pump or that affect its performance during operation are:

1) The ***power***. As a function of the flow rate, the pressure difference and the pump's mechanical efficiency, power consists an essential criterion for the selection of the drive machinery of the pump. This does not apply in selecting the pump itself, because it is possible that the supply provided by installing a high-powered pump to the piping system can also be achieved through a low-powered pump which operates at higher speed. Therefore, if no reduction

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[9] ***Bakelite*** (bacelite) is a polymeric resin used in the industry because of its high endurance to heat and chemicals, and is also excellent material for electrical insulation.

[10] ***Ni-resist*** is a non-magnetic iron alloy containing 14-25% nickel and other alloying elements (usually chromium, copper or molybdenum) to increase durability to fatigue and to facilitate thermal processing.

[11] The ***safety factor*** (SF) is a term that describes the ability of a thing or a material to handle / accept loads beyond the expected or actual ones.

of the rotating speed is required for the operation of a pump installed in the system, the choice of pump based on the upper range of its characteristics is a most economical solution in an effort to reduce the cost of the installation.

2) The ***pump flow rate*** or pump capacity, i.e. the total volume of fluid discharged per time unit. The fluid discharged includes liquids, dissolved gases, and dissolved solids, during specified pumping conditions.

3) The ***displacement*** (D_v), namely the estimated pump flow rate without slip losses. For pumps with single-acting pistons, displacement is given as:

$$D_v = A \cdot m \cdot n \cdot s \tag{2.9}$$

wherein: $D_v = \dot{V}$, A the cross-sectional area of the piston, m the number of pistons, n revolutions per minute of the pump, and s the length of the pump stroke (half the linear distance of the piston in one revolution).

For pumps with double acting pistons, displacement is:

$$D_v = (2A - a) \cdot m \cdot n \cdot s \tag{2.10}$$

$$A = \frac{\pi D^2}{4} \quad \text{and} \quad a = \frac{\pi d^2}{4}$$

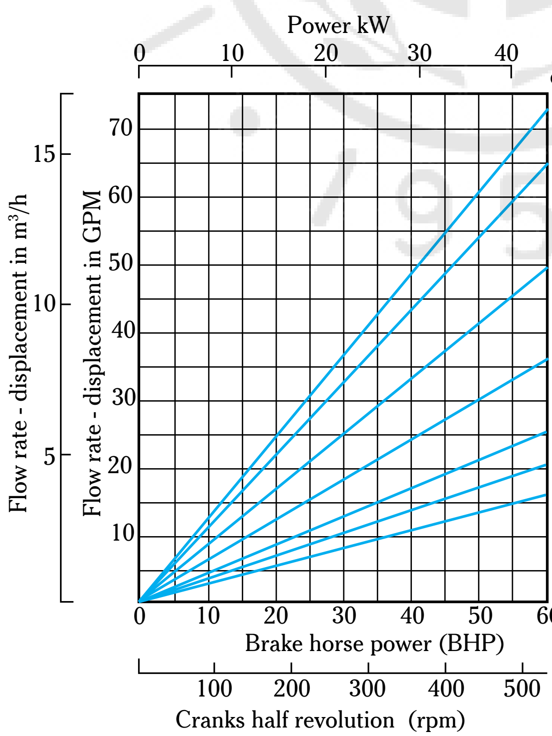
where d is the diameter of the piston rod, D the diameter of the piston, and a the cross section area of the piston rod.

4) The ***pressure***, more accurately the ***pressure difference Δp***, used to determine the power. It is the difference of the discharge pressure and the suction pressure.

In most applications, the suction pressure is lower than the discharge pressure. However, when pumping certain liquid gases, such as methane and propane, the suction pressure can be as high as 20 to 30% of the discharge pressure. But, independently of the prevailing conditions, for an accurate power calculation, we should always include the suction pressure. In figure 2.25 is shown a typical performance curve for a power pump.

5) The ***slip***, i.e. losses in flow rate due to leakage inside or outside the pump. Internal leakage is primarily the backflow through suction and discharge valves, while external leakage occurs mainly through the gland, due to leakage of the gland packings.

Reversing the flow rate can appear for a fraction of a second from the valves, while they remain open during the change in direction of the piston motion at the end of its stroke. In double action pistons, slight leakages may occur from the compression side to the suction side. Slippage is expressed as a percentage of loss in the flow rate, which is typically about 4%. The viscosity of the fluid, the speed of the pump and the discharge pressure can affect the slip, as shown in tables 2.2 and 2.3.



Plunger diameter IN (mm)	p_g LB/IN²	p_g (BAR)
2⅛" (54)	1 265	(87)
2" (51)	1 425	(98)
1¾" (44)	1 865	(129)
1½" (38)	2 500	(172)
1¼" (32)	3 650	(252)
1⅛" (29)	4 510	(311)
1" (25)	5 700	(393)

p_g: Manometer pressure

Fig. 2.25
Standard power pump performance curve based on zero slip.

Table 2.2: *Slip on pump with plate valve (or poppet valve).*

Viscosity cSt	100	1 000	2 000	6 000	10 000	12 000
Percentage of slip (%)	8	8.5	9.5	20	41	61

Table 2.3: *Slip as a function of pump speed and pressure.*

	Percentage of slip (%)		
Pressure bar	440 rpm	390 rpm	365 rpm
275	11	22	34
207	9	20	31
138	7	18	30
69	7	15	27.5

The slip is given as:

$$S = B + V + L \tag{2.11}$$

where: S the slip, B the external leakage by sealing or equivalent flow rate unit, V the inverting of the flow through valves and L the internal leakage.

6) The ***mechanical efficiency*** of a power pump, i.e. the sum of all the losses in the power imparted to the fluid by the drive machine (motor) and caused by the frictions of the mechanical power transmission parts. These parts include the pistons, the crossheads, the sealing materials of the shafts and the bearings of the crankshaft. The mechanical efficiency of power pumps is given as:

$$\eta_m = \frac{P_{out}}{P_{in}} \tag{2.12}$$

where P_{out} and P_{in} the power delivered by the pump and the power which enters from the drive machine respectively. $P_{out} = P_{in} - P_{ls}$ is the reduction of power due to losses (P_{ls}) during transmission. The mechanical efficiency in a single-acting pump often exceeds 90%, while in a double-acting piston pump it reaches 88%, due to losses by frictions on the additional sealing of the piston rod. If the pump is equipped with an internal gear of energy transmission, there are additional losses of around 2%. Most power pumps are designed to facilitate a range of different piston sizes. With larger pistons one is bound to use larger sealing elements, and also larger contact surfaces are created, which result in an increase of losses due to frictions. Generally, doubling the diameter of the piston leads to a decrease of the mechanical efficiency by 8%. Also, the mechanical efficiency is affected by the speed and, to a lesser extent, by the pressure increase (tab. 2.4 and 2.5).

7) The operating ***speed (velocity)***, which is one of the most critical criteria for the selection of power pumps. The rotating parts, as well as those which reciprocate, including the power end, are usually able to operate at a speed twice the real speed that has been set for the pump. However, irrespective of the strength of the rotating parts, the maximum speed of the pump is determined by:

a) The capabilities imparted during the design of the side in which the fluid is handled,

b) the ability of fluid suction from the piping system and

c) the desired duration of operation of the pistons, the sealing parts, and connections with minimum wear.

The specifications of most of the pumps limit the speed of the pistons from 0.71 m/s up to 1.42 m/s (140 with 280 ft/min). The piston speed is given by:

$$v_p = s \cdot n \tag{2.13}$$

where: vp the piston speed, s the stroke and n the pump rpm.

All pumps have a minimum operating speed limit, which is usually determined to ensure sufficient lubrication of the bearings in the power transmission device.

8) The ***volumetric efficiency*** (η_v), i.e. the ratio of the discharge volume of the fluid to the suction volume, expressed as a percentage (%), taking into

Table 2.4: *The effect of speed on mechanical efficiency at constant pressure development.*

Percentage of full speed, %	44	50	73	100
Percentage of η_m, (mechanical efficiency), %	93.3	92.5	92.5	92.5

Table 2.5: *The effect of pressure on the mechanical efficiency at constant speed development.*

Percentage of pressure increase, %	20	40	60	80	100
Percentage of η_m (mechanical efficiency), %	82	88	90.5	92	92.5

account the losses from slip (S). It is given by:

$$\eta_V = \frac{\dot{V}_{out}}{\dot{V}_{out} + \dot{V}_s} \qquad (2.14)$$

where $\dot{V}_{out}$ is the volume of pump outlet and $\dot{V}_s$ the flow rate losses from slip.

9) The ***torque*** (moment M). The average torque which is required by a power pump is independent of the speed of the pump, assuming that the suction and discharge pressures are kept constant. This means that a power pump is a constant torque device, and, in contrast to the centrifugal pump, the illustrated curve on a speed-torque diagram will be a flat straight line[12].

Starting torque is called the torque required to start the pump and to be able to accelerate so as to reach a stable speed. ***If the starting of a pump is performed with an open bypass between discharge and suction, the required starting torque is 25% relative to the operation torque. When the starting is performed under pressure in the discharge, the torque required is 125% of the operating torque.*** Therefore, when the pump is starting under load, electric motors with high torque are required. Also, when using a coupling system from the motor to the pump, one must take into account the losses in the torque that are ultimately attributed to the pump.

10) The power reduction ***(derating)***, expressing the function of a device at a lower than maximum power. It is applied in order to extend the operational 'life' of a pump. In positive displacement pumps, it may be necessary to reduce the operating power, in order, either to achieve improvement to the performance of the pump, or to reduce the damages to some of the pump parts when pumping fluids in a variety of applications with specific characteristics. The most common practice to reduce the power of a pump is to reduce power provided by the drive device, in order to reduce the rotational speed. Experience has shown that efficient operation of the pump parts which come in contact with the fluid, such as the pistons, the sealing materials and the valves, might be extended, and their wear might be limited if the speed of the pump, and, by extension, the speed of the piston stroke, is reduced during pumping of certain fluids. Also, for pumping high viscosity liquids, a reduction of the speed of the pump may be required, while, if the pump is designed specifically for pumping high viscosity liquids, a moderate speed reduction may be required.

11) The ***pulsations*** in the flow rate (or fluctuations), which are a result of the piston pumping cycles, and, depending on the piston direction, create the suction and discharge flow of fluid from the pump. The intensity of pulsations in the flow rate is significantly influenced by the number of pistons, and especially by depression fluctuation, because of the high potential energy produced when the resistance from the discharge system is large, in order to create pressure. Given that the size of the depression fluctuation is mainly affected by the number of the pistons, increasing their number, and thus the number of cylinders, the pulsations in flow rate would be reduced.

12) ***Separation of liquid from the piston***. As mentioned, the liquid flows through the suction valve in order to fill the volume of the cylinder which increases. If the acceleration of the piston is greater than the velocity of the incoming liquid flow, the liquid will lose its contact with the surface of the piston, because the vacuum that is created would have the lowest pressure than on any other point inside the cylinder. Then, if the liquid entering the pump chamber entrains gases, they will be separated from the liquid solution and would be concentrated in an area with lower pressure. However, the gas bubbles, during the stroke, would be compressed, resulting in their breaking due to the pressure, which would cause cavitation and lesions that would appear on the surfaces of the piston and cylinder.

The geometry of the sliding mechanism driveline of the crank affects the point at which the liquid separation takes place, and is dependent on the ratio of connecting rod length to the radius of the crank, relative to the rotation speed. As the ratio of the length of the connecting rod to the crank radius increases, the rotation speed of the crankshaft of the pump, where the separation of the liquid from the surface of the piston appears, would be reduced. Since liquid separation may be the determining factor for Suction Pressure (NPSHr), during pump design the geometry of the sliding drive mechanism and the crankshaft should be carefully evaluated, in order to optimize the hydraulic efficiency of the pump.

[12] The torque required at the crankshaft of the pump is given as follows: $M = 9\,554 \cdot P / n$, where: M is the torque at the pump in $N \cdot m$, n is the speed in rpm and P the power in kW.

13) ***Unbalanced forces.*** Due to the relatively slow speed of a piston pump, the loads that are developed by the inertia of rotating or reciprocating parts are quite low compared to the loads developed at centrifugal pumps, thereby minimizing problems which may be caused by vibrations. For this reason, the crankshafts are usually not balanced with great precision. But when the pump is coupled with a high speed electric motor, or the coupling of the motor pump is done through a gear unit (or gear reducer), it may be required to analyze the moments of the moving parts of the pump, and the mechanism of movement transmission. The forces deployed are:

a) ***Unbalanced Reciprocating Parts Force – F_{rec}***. These components are usually one-third of the weight of the connecting rod, the cross head with bearings, the wrist pin and the plunger. In vertically arranged pumps, the piston rod and the piston nut are included in these components.

b) ***Unbalanced Rotating Parts Force – F_{rot}***. These components are typically about two thirds of the weight of the connecting rod, the crankshaft and its bearings, and the crank pin.

2.6.3 Diaphragm pumps.

Diaphragm pumps are positive displacement pumps, and their operating principle is the same as the one used in reciprocating pumps. The pistons used in those pumps are replaced by diaphragms, which are flexible membranes, usually elastic or made of teflon, that move either mechanically or hydraulically or pneumatically (by using air), thereby characterizing the types of diaphragm pumps. Specific types are:

1) ***Mechanically moving diaphragm pumps*** in which the diaphragm reciprocates from the piston that moves through a cam. By pushing the diaphragm through the piston, the pressure to the liquid, present in the chamber that is created by the pump housing, is increased. By increasing the pressure, the non-return discharge or exhaust valve opens, and the liquid is discharged to the piping system. With the continuous rotation of the cam and the assistance of the return spring, the piston moves towards the opposite direction, resulting in the diaphragm returning to its initial position. At this time, vacuum is created in the pump chamber, which causes the closing of the discharge valve and the opening of the non-return suction valve, so that a new amount of liquid can enter into the chamber. With the reciprocating movement of the piston caused as the cam rotates, and through the non-return valves, the continuous suction and discharge of the pump is accomplished (fig. 2.26). On ships, diaphragm pumps are also found as dosing pumps, due to their ability to displace budgeted quantity of the liquid during each piston movement towards the casing cover. The amount of discharge of the pump can be varied according to the desired one through the adjustable knob (hand wheel) of the piston stroke.

2) ***Hydraulically moving diaphragm pumps***, consisting of the pump body inside which there is the diaphragm, the hydraulic liquid, and the piston that reciprocates (fig. 2.27).

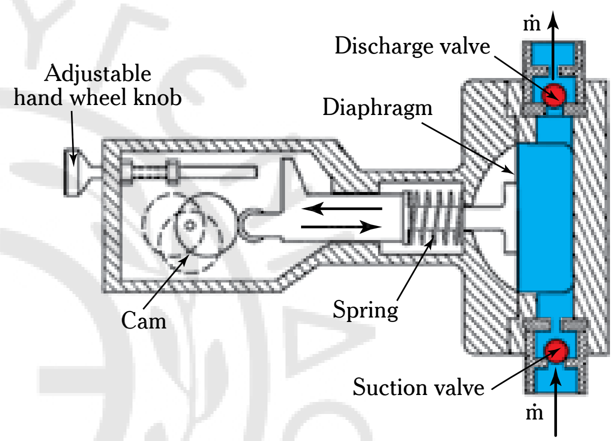


Fig. 2.26
Section of mechanically driven diaphragm pumps.

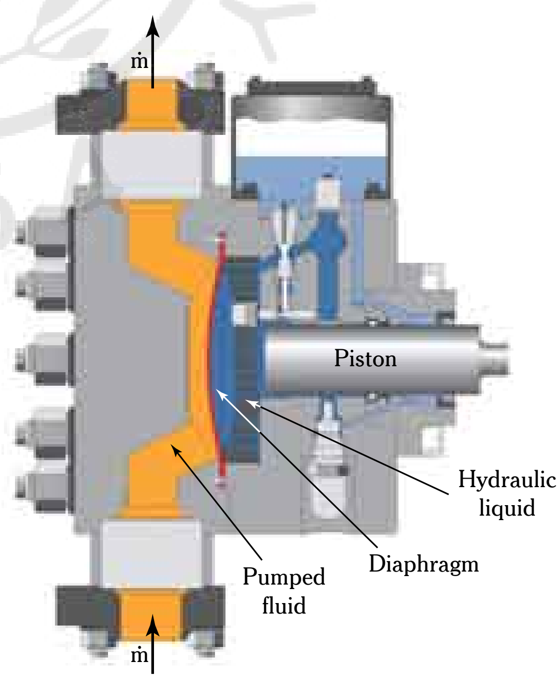


Fig. 2.27
Section of hydraulically driven diaphragm pumps.

In pumps of this type, the reciprocating movement of the piston is applied to the hydraulic fluid, which causes the diaphragm to flex forward and back by varying the volume of the chamber, without the piston contacting the diaphragm. Thus, the diaphragm follows the reciprocating movement of the piston, through the hydraulic fluid, and generates the suction and discharge conditions of the respective valves. In hydraulic diaphragm pumps, the operating pressure between the hydraulic and the pumped liquid is usually the same. This eliminates the fatigue of the diaphragm, since the pressure is substantially equal on both sides, at any moment. In order to achieve a smooth operation of the pump, relief valves and automatic filling valves are installed, with which the flow and the hydraulic liquid pressure is controlled. Also, an automatic vent valve is installed in order to continuously remove any air entrained in the hydraulic operating liquid.

The diaphragms are made of a resilient material, such as polytetrafluoroethylene (PTFE, commonly known as teflon), and metal, when the diaphragm pumps are used in handling liquids under high pressure, high temperature, or liquids that are highly corrosive. Due to the reduced number of parts of the pump that contact the pumped liquid, since, apart from the diaphragm, the only moving parts in contact with the liquid are the specially designed valves for its entry and exit, a diaphragm pump requires less maintenance compared to a piston pump.

3) ***Air-operated diaphragm pumps (AODPS)*** (or pneumatic pumps with diaphragms) [fig. 2.28(a), (b)] which consist of:

a) The ***pump body***, that must have an appropriate formation, so that the liquid can flow easily through it.

b) The ***air distribution valve***, for the air that is needed when the pump operates.

c) The flexible ***diaphragms*** (or membranes), made from rubber or teflon.

d) The connecting ***rod***, at the edges of which the two diaphragms are connected.

e) The non return ***valves***, used for suction and discharge.

Each of these diaphragms is settled inside a closed cylinder that is created by the body of the pump. Thus, the volume of the cylinder and the pressure inside it can increase or decrease whenever the diaphragm regresses.

The air that is needed for the operation of the pump is provided through a distribution valve, which is settled on the pump body. While the air enters the cylinder, which is created by the diaphragm and the body of the pump, the diaphragm stimulates, causing an increase of the pressure that pushes the liquid inside the cylinder on the other side of the diaphragm. With the opening of the non return exhaust valve, as a result of the increase of the liquid pressure, the discharge procedure is completed. At the same time, the second diaphragm stimulates through the rod, which is moving axially, causing a parallel regression of the two diaphragms, because of their connection through the rod.

In this way, while one diaphragm is used to decrease cylinder volume and complete the discharge process, the other diaphragm is used to increase the cylinder volume and complete the suction from the piping system. Furthermore, with an appropriate arrangement, through a shaft that is activated by the movement of the diaphragm, the air distribution valve transposes, so that the air can also be used by the second diaphragm. Thus, the procedures of suction and discharge take place, one at a time, each of them on one side of the pump. The types of valves that are used on pneumatic pumps are: 1) ***Flap valves***. Liquids that contain solid particles that have nearly the size of the diameter of the pump can easily flow through it. 2) ***Ball valves***, that provide better impermeability and resistance and 3) ***Poppet valves*** that close, affected by the power of a spring [fig. 2.28(c)].

Air-operated diaphragm pumps with diaphragms, due to their structure, have the ability to operate dry, without the existence of liquid and without being destroyed. They are appropriate for pumping viscous liquids. Their supply is regulated by the operating air with which the distribution valve is supplied, they have an automatic suction system, and they discharge on a high height that depends on the pressure of the operating air.

2.7 Rotary pumps.

Rotary pumps or ***volumetric type pumps*** are a category of positive displacement pumps which force the liquid to flow by pressure. Their operation is based on one or two rotating rotors within a closed chamber (or cavity), which is defined by the pump casing.

The pumped fluid is trapped in appropriately de-

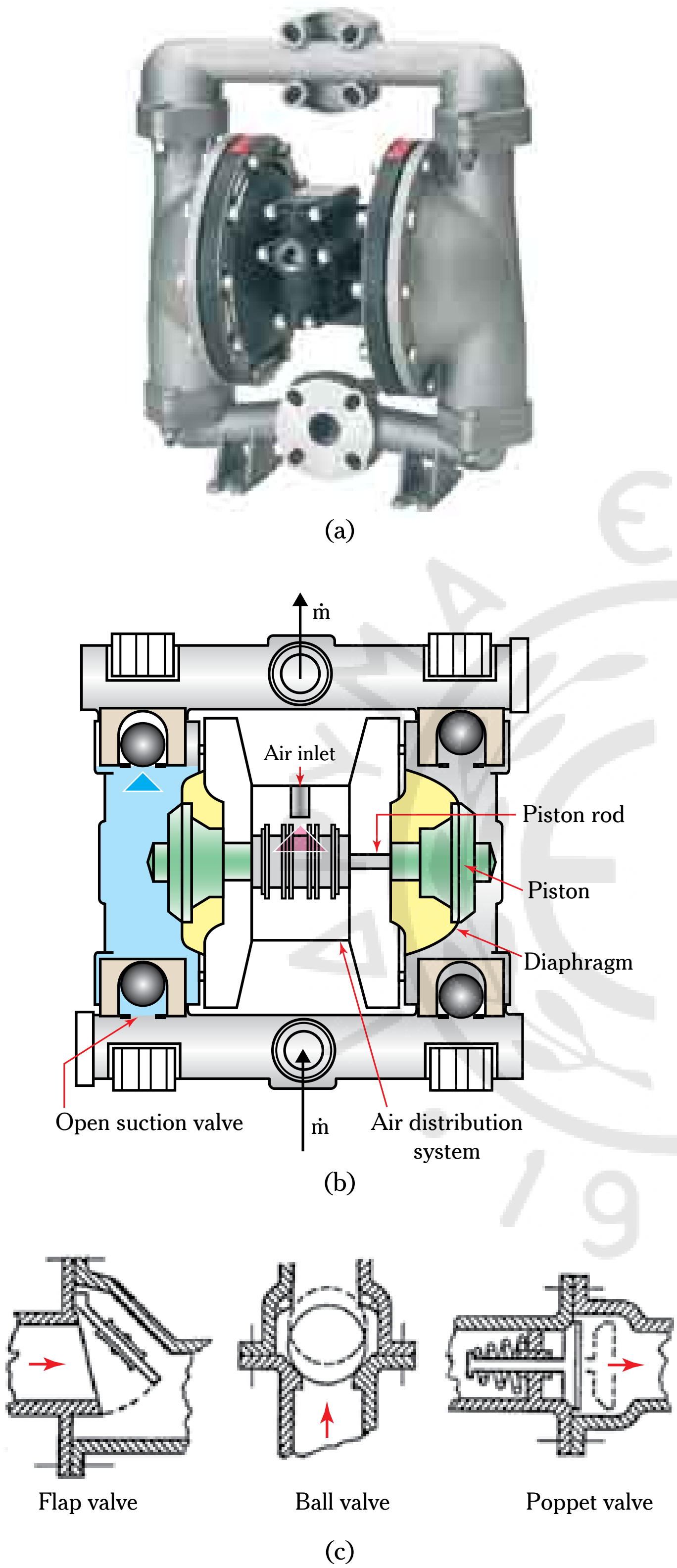


Fig. 2.28
(a) Air-operated, double-diaphragm pump with ball valves, (b) Section and representation of the parts, (c) types of valves.

signed smaller chambers formed between the rotors and the housing, so that, as the rotors are rotated, the fluid is transported by suction to the discharge. Rotary pumps have the same features as piston reciprocating pumps, since the pressure at the pump outlet is increased, as the energy of the discharge pipe load increases. The difference is that the moving parts in piston reciprocating pumps follow a reciprocal motion, while in rotating pumps the moving parts rotate and displace a finite amount of fluid in each revolution of the drive shaft.

Rotary pumps consist of a ***body*** or casing on which there are ***gates*** for the inlet and outlet of the liquid. The casing with the ***chamber covers*** (one on each side) and the moving parts of the pump create ***chambers***, containing the transferred liquid. The mobile parts or the rotor have small interstices with the fixed casing and the covers, in order to achieve the free rotation of the rotor.

The kind of the rotor depends on the specific type used in each rotary pump, and it, accordingly, may consist of flaps, screws, lobes, gears, sliding pistons, etc., so as to distinguish the rotary pumps into equivalent types.

The rotor is mounted on a shaft, which is supported by a suitable sealing device torque on the covers of the casing, in order to transmit the necessary for rotating the rotor of the drive machine. The sealing on the caps from where the rotational shaft passes to connect to the drive machine, is achieved by creating ***seal chambers***, where there are placed ***mechanical seals*** or ***gland packings*** or oil seals-simmering (synthetic reinforced with metal sealing rings).

During pump operation, in order for liquid to be transferred, liquid is trapped within small rotating chambers (or cavities) of variable volume, which are formed between the movable and stationary portions of the pump. As these chambers pass through the inlet port, they reach and reveal their maximum volume, which is filled by liquid. Then, by the rotation of the rotor, the volume is gradually reduced and this results in the liquid exiting towards the piping system under pressure, through the discharge ports. The absence of sealing between the moving and stationary parts of the pumps (i.e. between the rotor and the shell or the casing caps) means that the ***clearance*** (gap) must be very small in order to ensure the minimization of leakages, but it must also be enough to provide the rotation of the rotor freely. Thus, the amount of fluid that is leaking between the rotating chambers will be so small that it will not have the ability to affect the performance of rotary pumps. A further factor that should be considered, when controlling the rotation clearance, is the thermal expansion of components, which may cause the reduction of the gap, during the operation of the pump, by limiting the free rotation of the rotor. Therefore, during inspection or repair, particular attention should be given to the assembly, so as to preserve the clearances specified by the manufacturers.

The application field of rotary pumps extends to all kinds of liquids having a lubricity and sufficient viscosity to prevent excessive leakage through the gaps, at the pressure required to operate the pump.

The ***operational features*** of rotary pumps are:

1) The ***displacement of the pump*** $\dot{V}_d$. This is the total volume ($\dot{V}$) of the liquid that is displaced in given shaft rotations (n). It forms the theoretical pump capacity, assuming that all of the pump chambers that are created by the rotor and the pump casing are completely filled with liquid, and there are no losses between them.

$$\dot{V}_d = V \cdot n \tag{2.15}$$

2) The ***slip volume*** $\dot{V}_s$, which represents the liquid flow leaking from discharge to suction side through the gaps of the chambers that are created by the pump rotor.

$$\dot{V}_s = (V \cdot n) - S \tag{2.16}$$

The slipping is increased by the pressure depression and decreases by the differences in the fluid viscosity.

3) The ***volume flow rate***[13] V. It is given by the following equation:

$$\dot{V} = \dot{V}_d - \dot{V}_s \tag{2.17}$$

By this, the volumetric efficiency is calculated (η_V), which is defined as the ratio of actual to theoretical flow rate.

As mentioned, the suction and discharge conditions affect the pump capacity. Therefore the viscosity of the liquid, the tendency to create vapors in the suction, the amount of air entrapped, or the air that is dissolved in the liquid, as well as the big total height, are factors which are causing reduction of the flow rate.

Rotary pumps are used for low capacity and medium pressures, by addressing various manometric heads without significant changes in their flow rate. The reduction in flow rate of rotary pumps is shown when the total head required for the discharge of the liquid exceeds the maximum pressure at which the pump can maintain the sealing of its chambers.

– Types of Rotary Pumps.

Positive displacement rotary pumps, based on their shape and the construction details of their rotor, are distinguished into:

1) ***Single rotor***, comprising vane pumps, liquid piston pumps, pumps with pistons of variable stroke (radial or axial), eccentric helical rotor pumps and peristaltic pumps.

2) ***Multiple rotor***, comprising gear pumps, pumps with lobes and pumps with screws.

Depending on the way their drive machine is actuated, they may be:

1) Driven pumps (attached pumps or ***dependent***), when their motion is provided by a moving part of another engine, which transmits the motion to the pumps with gears.

2) ***Independent pumps*** (or just pumps, due to their independent motivation), when they are moving by an electric motor or diesel engine.

On ships, rotary pumps are used as fuel oil boiler feed pumps, on lubricating or cooling systems of diesel generators, in the transfusion or drainage of lubricating oil tanks, in tanks for the transfer of fuels or water, for the movement of hydraulic rudders, for the movement of winches, etc.

For the selection of the construction materials, the following material properties should be taken into account:

1) The modulus of elasticity, in order to resist deformations from the forces exerted during operation.

2) The thermal expansion coefficient, which, due to the variation in temperature of the conveyed liquid, affects directly the clearances between the

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[13] ***Volume Flow Rate*** ("flow rate" or "pump capacity $\dot{V}$") is the volume of liquid per unit time delivered by the pump.

rotors and the shell and, hence, the smooth and efficient operation of the pump.

3) The friction coefficient, which affects the wear resistance between contacting surfaces that are sliding or dragging.

Therefore, depending on the purpose, materials that are used in the manufacture of the rotary displacement pumps are:

1) For the ***shell***: cast iron, cast steel or brass.

2) For the ***rotor***: synthetic rubber, cast steel, forged steel or brass.

3) For the ***valves***: cast steel, phosphor bronze, copper-tin alloy, stainless steel or Monel metal.

The rotary pumps, which are commonly found on ships, and belong to the single and multiple impeller type, are the following:

1) ***Vane pumps*** (fig. 2.29). Pumps of this type are consisted of a cylindrical casing, within which the rotors are rotating. Radially, on the cylindrical rotor drum exist slots, and in each slot, sliding vanes or blades are fitted. These vanes are usually made of rigid non-metallic material (e.g. bakelite). As the impeller rotates, the vanes slide, due to the centrifugal force, and their contact with the internal surface of the cylindrical casing is maintained.

Since the slotted rotor is eccentrically placed into the cycloidal casing, the center of rotation does not coincide with the center of the casing. Due to the impeller eccentric rotation in relation to the pump cylindrical cavity, the sliding vanes which are in contact with the internal circumference of the casing are forced to reciprocate inside the slots, and so, the surface of the vane changes at every point of the rotation. Hence, between the vanes, the casing, and the covers, cavities with variable volume are created. While the rotor moves by the arrow impetus (fig. 2.29c), the vanes are retracted from the center and the cavity which is created increases. This increase causes the creation of vacuum resulting in the suction of the liquid, which enters into the cavity through the inlet port.

Then, the volume of the cavity, which is created due to the change of the rotor distance from the internal circumference of the casing, starts to progressively decrease. The result of this decrease in the cavity volume causes the pressure of the pumped fluid that exists in the cavity, and its outlet from the discharge

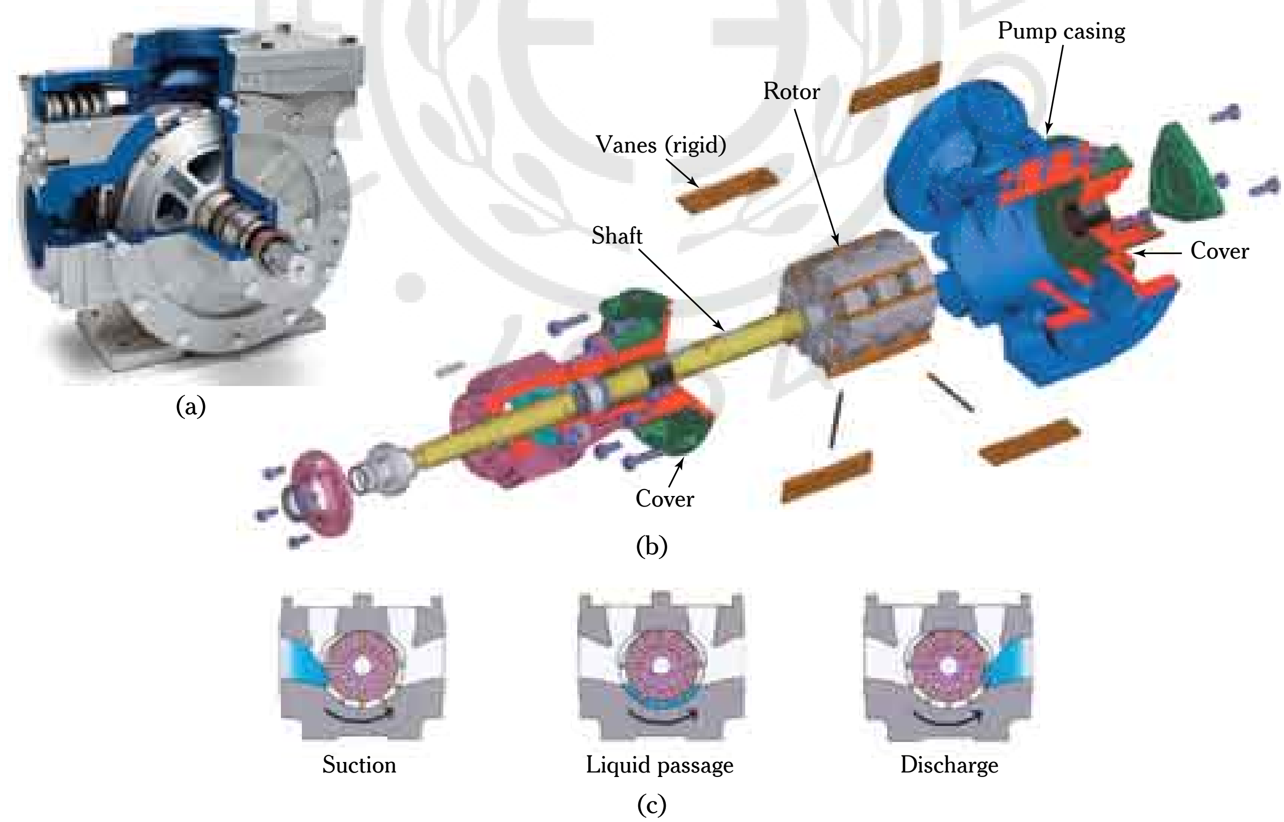


Fig. 2.29

(a) Vane pump, (b) pump parts, (c) flow of liquid within the pump.

ports. To facilitate the reciprocal motion of the vanes and their contact with the periphery of the pump casing, springs are placed in order to push the vanes towards the periphery, when the rotor rotation speed is low and sufficient centrifugal force does not develop inside the socket of the vanes. The installation of three or four sliding vanes between the suction and discharge prevent the back flow of liquid to the suction where low pressure exists, especially on pumps with low rotational speeds. For pumps which handle low viscosity fluids, satisfactory, for the category, capacity is achieved by operating speeds at 1 000 rpm.

Vane pumps have a great degree of efficiency, low noise operation and long life. They are used in oil transfer from a tank to another, while, when high-speed vane pumps are used to increase the oil operating pressure in hydraulic systems, vanes are manufactured of rigid, high strength, metal alloys. Typical alloys are those of steel with nickel, magnesium and carbon.

2) ***Liquid piston pumps***. Pumps with liquid pistons are manufactured in two types:

a) Pumps with elliptical casing, which have suction and discharge ports on the inner hub of the rotor and which are referred as ***nozzle type pumps*** and

b) Pumps with ***eccentric circular casing***, where the suction and discharge is achieved through ports in the side transversely plate of the housing and are called ***plate type pumps***.

The rotor in both types is comprised of vanes, which create the liquid suction and discharge cavities. The sealing between the cavities is accomplished by providing water, which rotates with the rotor. As water follows the shape of the casing, it enters and exits in the cavities between the vanes, increasing or reducing the volume over the liquid surface (fig. 2.30).

Each time the water emerges from the space between the vanes, the volume within the cavity is increased. As a result, the vacuum which creates the suction through the inlet port increases, while, when entering, it creates a reduction of the volume within the cavity and pushes the liquid to the discharge. This pump, due to water losses (or other working fluid losses), which occur as water is entrained with gases to the depression, is provided with a regulating constant flow rate arrangement that supplies the exact amount of the sealing liquid which is required.

Liquid piston pumps are used for removing air and creating suction vacuum into (main) pumps, where the suction is located above the free surface of the fluid. Their operation lasts until all the air is removed from the suction tube, in order to start suction from the main pump.

3) ***Eccentric helical rotor pumps*** (progressive cavity pump or roto pumps). With these pumps, smooth liquid transfer is ensured, without turbulences or pulsations.

They provide continuous flow rate, while flow reversing is achieved by changing the rotor rotation.

The pump consists of a steel helical rotor, while the shell is made internally of elastomeric material[14] (fig. 2.31). The geometry and the dimensions of these parts are designed to form, as the rotor is rotated within the casing, a double row of sealed discrete cavities (or chambers). The rotor has the form of a single helix with a circular cross section, and, as it rotates, it creates a vertical circle which moves along the axis, passing through the center of rotation of the rotor. The rotor rotates within the fixed housing while abutting its inside, so that the circle delineated by moving along the rotor corresponds to the displacement of the chambers containing the fluid. This creates a volumetric flow rate proportional to the rotation rate (bidirectionally), and moves the liquid by the suction to the discharge. If greater pressure is required to be developed by the pump, a corresponding increase should be effected in the pump length, in order to achieve this pressure.

The eccentric helical rotor pumps are suitable for transfusion of viscous liquids, such as oil or petroleum dirty residues (sludge), and liquid having high solids content, fibrous materials, gases, and fluids that create foam.

4) ***Geared pumps*** belong to the category of rotary pumps with multiple rotors and are divided, according to their construction and the operation of their rotors, in two categories:

a) ***External gear pumps*** (or pumps with external toothed wheels), that consist of two shafts, which bear gears. The two toothed wheels have the same diameter, the same number of teeth, and the same pitch.

[14] ***Elastomeric material*** (elastomer, elastic polymer) is a polymer that shows elastic properties. It is a polymeric material that combines both viscosity and elasticity, has very weak intermolecular forces and low elastic modulus as well as high tear strength compared to other materials.

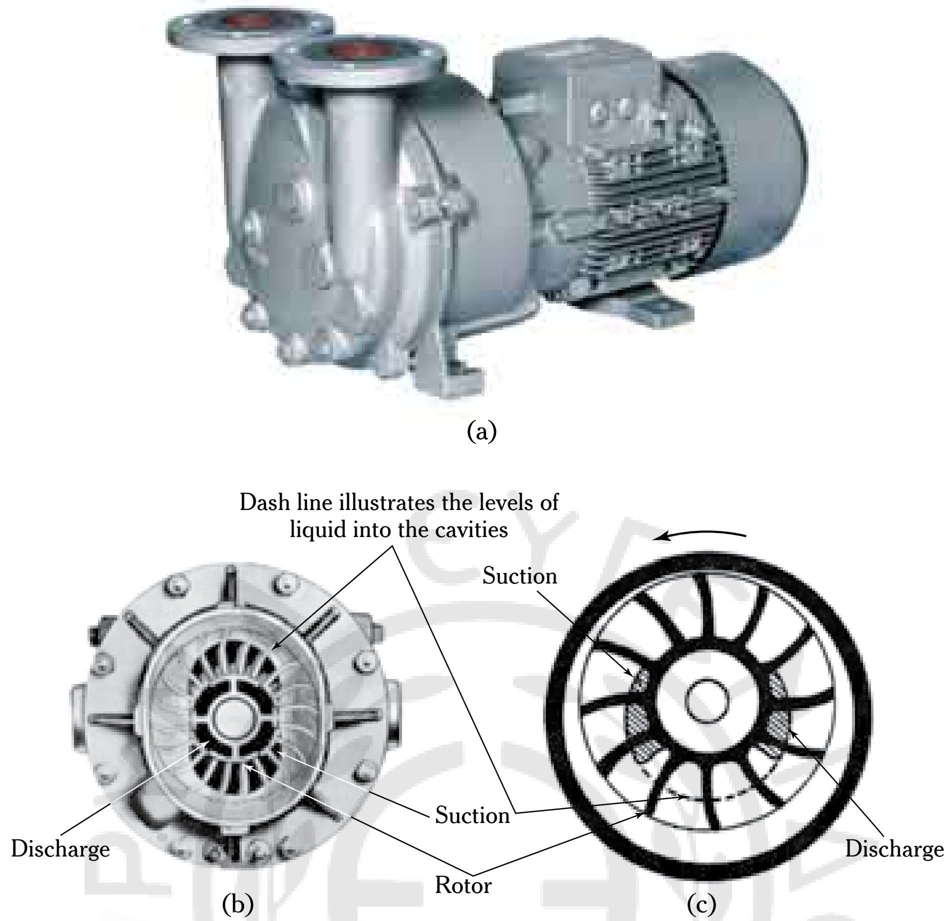


Fig. 2.30
(a), (b) Liquid piston pump and (c) cross-section of the pump.

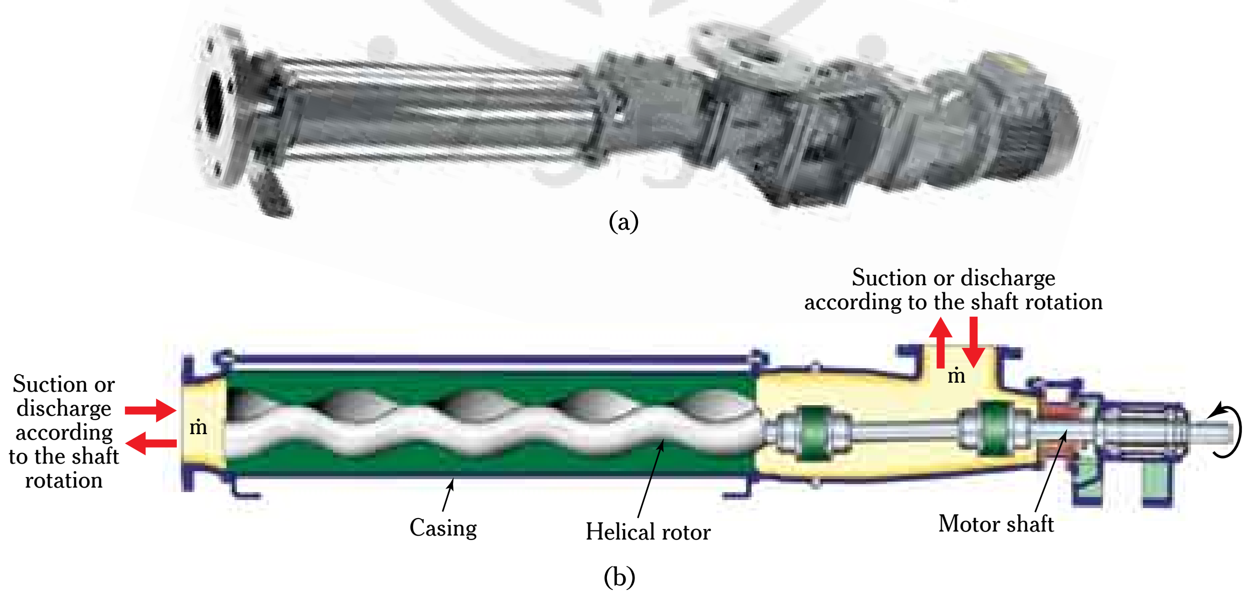


Fig. 2.31
(a) Helical rotor pump and (b) schematic illustration of its parts.

Their rotation is performed within a casing that is surrounding both wheels, while the two side plates, which are placed in the casing, create the liquid chamber. The support of the shafts of the gear pumps is carried out in suitably shaped slots of the side plates, on which lies the shaft of the output for connection with the drive rotational machine. The sealing of the shaft, on the outlet of the side plates, is usually achieved by oil seals.

The motion for rotating the geared wheels is provided from the drive unit to one of the two shafts and, as the teeth engage, the movement is transmitted from one wheel to the other. As the wheels rotate in opposite directions, they are carrying the liquid from the suction into chambers which are formed circumferentially with the pump housing. When the teeth engage again, the liquid is pushed by pressure to the discharge (fig. 2.32).

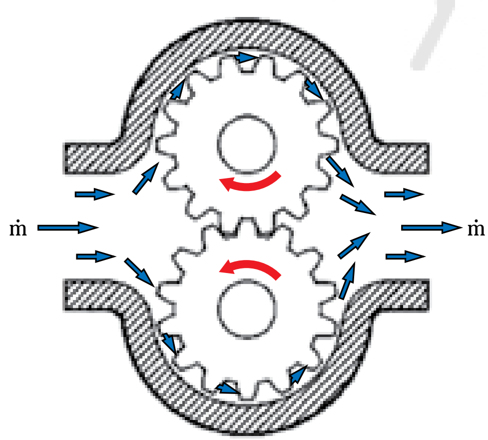
The pressure developed due to the tightness between the gears is too high, thus creating stresses in the gears and on the bearings of the pump, resulting in their radial loading. In order to reduce the radial load, either diametric slots are created on the gears, allowing diametrically the communication between chambers, or the high pressure side is connected to the suction side and the low pressure side to the discharge side through connecting tubes on the casing. Certain types of pumps with connecting tubes on the shell have valves for controlling the pressure. Also, the load that is created, due to the trapped fluid as the teeth of one gear are engaged between the two teeth of the other, is eliminated by creating slots in the side plates or the bearings. The gears are made of chrome nickel steel alloy and subjected to hardening by surface dyeing; the shell consists of cast iron or duralumin, while the side plates are made of phosphor bronze or other special alloy. The shape of the teeth of the rotor gear, may, other than linear, be helical or herringbone, according to their structure.

The relative size of self-priming gear pumps is small, providing a small amount of liquid, which depends on the volume of the chambers and the speed of their rotation. Because of the pressure, that can reach 17 bar, relieve valves are installed on the discharge side. Their operating speed varies between 800 rpm and 3 600 rpm, and reaches high flow rate. It should be noted that the high levels of noise and the fatigue of the pumps during the operation, leads the rotation speeds in practice not to exceed the limit of 1 500 rpm. The pumps of this type may be operated in the opposite flow direction; it is sufficient to reverse the rotation of the driving machine. The safe use of the pumps in this case is achieved by expansion valves which are located on both sides of the pump. The ability to reverse the operation of the pumps, in combination with the high pressures attained, is utilized to provide oil under pressure in hydraulic systems, for operating auxiliary machinery.

The performance of these pumps is affected by the clearance between the teeth of the rotor and the clearance between them and the shell, so the smaller they are, the higher the degree of their efficiency.

In certain types of pumps with external gears, the transmission of motion takes place on one of the shafts, and then is transmitted to the rotors which convey the liquid. In this type of construction, the

(a)



(b)

Fig. 2.32
(a) Gear pump and (b) section and flow in the pump.

clearance between the rotor teeth should be greater than the spacing of teeth on the drive wheels, so as to avoid the contact between rotor wheels, providing freedom of movement without affecting the level of pump performance.

b) ***Internal gear pumps*** (fig. 2.33), which have one larger gear with gear teeth cut internally on the major diameter meshing with a smaller externally cut gear. The movement is usually transmitted to the gear with the externally cut teeth, which is constructed with less teeth than the internally cut teeth gear. As a result, because of the difference between the number of teeth and recesses, the rotations of the ring will be less than those of the gear. The movement could be given to the internally cut teeth gear with similar functional principles and results.

As the gear with the outer teeth rotates to the direction that is shown in figure 2.33(b), the outer ring is drifted and the volume of liquid chamber increases, creating a vacuum for the suction of the liquid. Then, the teeth enter into the recesses of the ring, so that the liquid is driven to the discharge. The system gear-ring rotates within the shell, while in both sides are placed plates that create the fluid circulation chamber.

In other pump types with inner gear, in the space between the teeth of the two gears, there is a crescent, so that it can keep sealing chambers between the suction and the discharge [fig. 2.33(a)]. The crescent is necessary because it can block the return of the liquid from discharge to suction. The clearances between the parts which constitute the pump are small, so that the sufficient seal between the chambers can keep the pump efficiency in satisfying levels.

The function rotations of these pumps reach 1200 rpm for low viscosity fluids, but when they are used in pumping high viscosity liquid, in order for enough time to exist for the viscous liquids to enter the chamber created between the gears, the rotation speed is low.

5) ***Lobe pumps*** (fig. 2.34), constructed and operated like the gear pumps, but each rotor has two or more lobes. The surfaces of the rotors are shaped to cooperate, constantly ensuring seal between themselves and with the inside circumference of the casing of the pump.

At the top of each lobe, depending on the type of the pump, metal spokes are appropriately applied, which are pushed internally by intensive springs, reinforcing the spoke contact with the interior of the casing, so as to increase the sealing during rotation of the lobes. The timing of the rotors is performed externally of the pump casing, with gears placed in the rotation shafts of the rotors. Due to the small number of lobes, fluctuations may be presented in the flow rate during the operation of the pumps, while they present greater strength relative to the geared pumps, because the metal parts of the lobes at their contact points are usually covered by a protective pad. Their use is advisable for the transfusion of viscous liquids, and liquids containing a large amount of suspended solid particles, with the provision that their operation is performed at low speed.

Usually the manufacture material of lobe pumps is stainless steel, plastic or ceramic materials.

6) ***Screw pumps*** are composed of the casing and the interminable screws which rotate inside the casing. Depending on layout and how they operate they are distinguished in two types:

(a)

(b)

Fig. 2.33

(a) Internal gear pump and crescent and (b) section of internal gear pump.

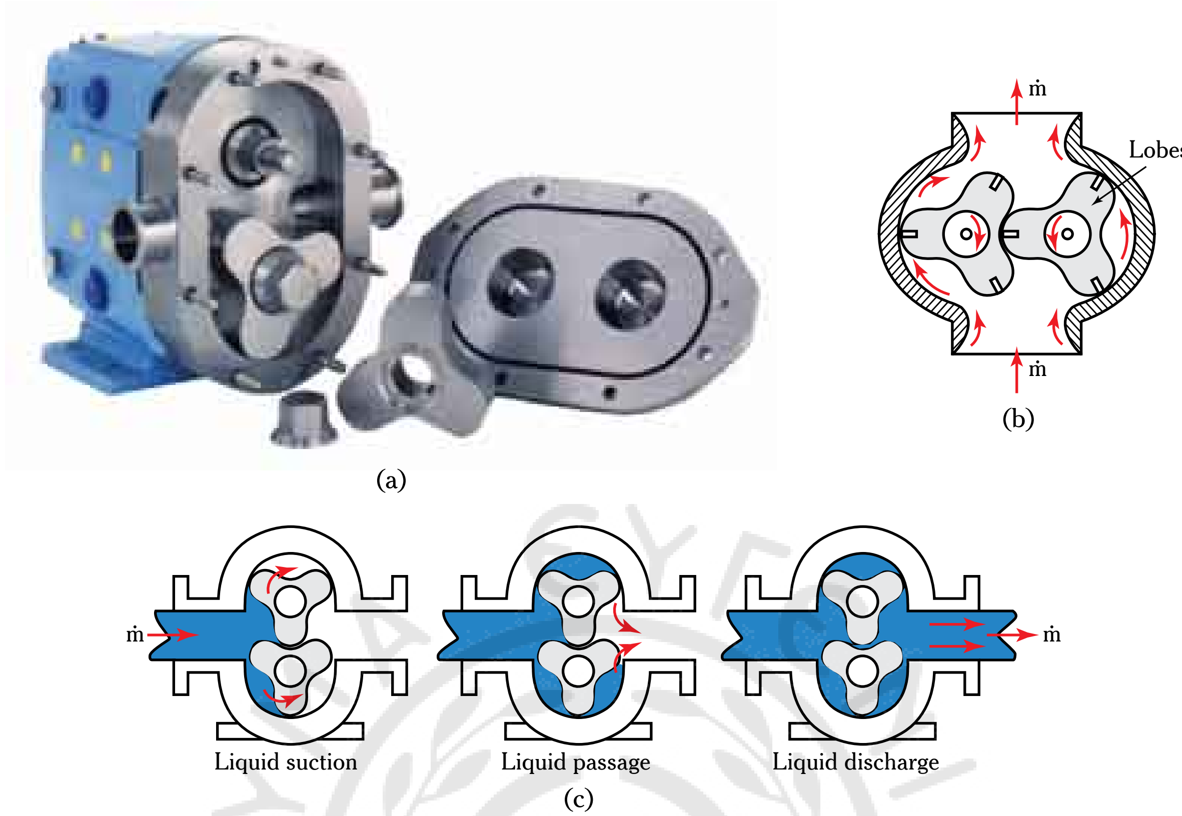


Fig. 2.34
(a) Lobe pump, (b) pump schematic illustration with metal spokes and lobes (c) liquid flow through the pump.

a) The ***timed screw pumps***, which consist of two screw shafts, where, the one is rotating in a clockwise, and the other in a counterclockwise direction (fig. 2.35). One of the two shafts is connected to the drive machine and provides the motion to the other one with external gears, so that the screws, while being involved with accuracy, are not in contact. The screws with the casing form chambers which are being moved in parallel to the axes of the screws, as they rotate transferring the liquid from the suction to discharge.

b) The ***untimed screw pumps***, which are constructed by three screw shafts (fig. 2.36). These pumps have no driveline gears on each shaft, but the central shaft (rotor) receives the motion and transmits it to the lateral screws (idles) directly by contact. The principle of liquid transfer is identical to that of timed screw pumps through the chambers that are created by screws with the casing. The support of the shafts is formed at the appropriate point on the casing retaining them in the desired position. These types of pumps are used for pumping only completely pure liquids, because the screws are in direct contact with pressed liquids, and any solid particles in the liquid would cause the destruction of the surfaces and reduction in pump efficiency. They also have high suction capacity and achieve a laminar movement of the fluid without mixing it, preventing the formation of emulsions.

7) ***Rotary pumps with variable piston stroke***. These pumps can develop extreme pressures. Their peculiarity lies in the ability to develop high rotation speeds, despite the fact that the suction and the discharge are generated by the reciprocating motion of the pistons.

During their operation, the pistons reciprocate in an equal number of cylinders which are arranged either radially (vertically) or parallel to the pump rotation axis. Because of this arrangement, the pumps are divided into two main categories:

a) Those of ***radial pistons***, where the rotor of the pump consists of the cylinder block (of the pistons), which are arranged perpendicular to the pump shaft and rotated within a reaction (or floating) ring, which can also be moved within the cylindrical shell of the pump. Correspondingly, the pistons that are fitted into the cylinders, are, by the rotor rotations, due to the centrifugal force, pushed towards the periphery, with their axis perpendicular to the rotor rotation axis. In the center of the rotor are the suction and discharge ports of the pump (fig. 2.37).

When the pump is in operation, the pistons are pushed towards the circumference and slide on the reaction (floating) ring, while maintaining the same distance from the center of rotor rotation, since the center of the ring coincides with the center of the rotor [fig. 2.37(b)].

If the reaction (floating) ring is moved towards the pump housing, the pistons which slide on the ring are forced in reciprocal motion. Thus, the volume within the cylinders is varied, so as to accomplish suction and discharge. Hence, with the reaction (floating) ring on the left position [fig. 2.37(c)], the suction

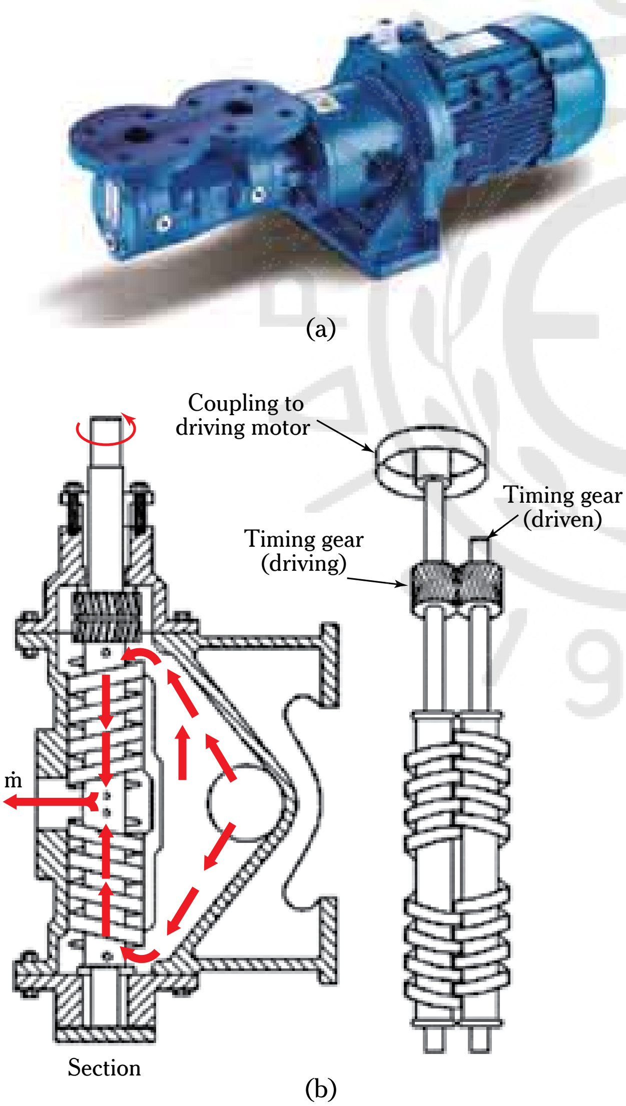


Fig. 2.35
(a) Timed screw pump and (b) schematic illustration.

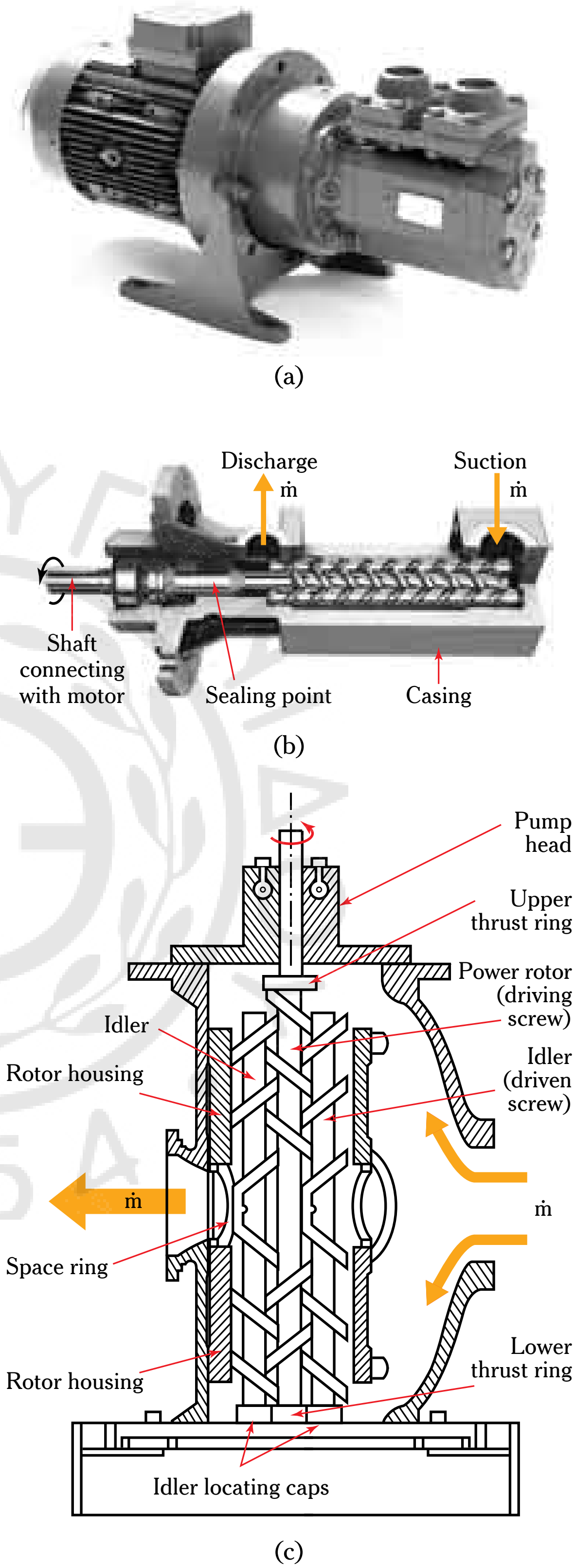


Fig. 2.36
(a) Asynchronous screw pump, (b) section and (c) schematic illustration.

is performed by port B and the discharge by port A. Respectively, with the ring on the right side [fig. 2.37(d)], the suction is performed by port A and the discharge by B port.

The rotor, during the operation of the pump, is rotated in the same direction, so that the discharge can be achieved by the movement of the sliding ring only, without the interruption of rotation of the pump to be required. Similarly a change of the flow rate of the pump is achieved, which is performed by changing the relative position of the rotor center to the center of the ring, So, the flow rate is altered depending on the displacement of the cylinder and, hence, the amount of liquid discharge from the pump changes too.

b) Those of ***axial pistons***, where the pistons are arranged in parallel around the rotor rotation axis (fig. 2.38). The cylinders block of the pistons constitute the rotor of the pump and is rotated by the motor shaft that gives the movement to the pump. The inlet and outlet ports of the liquid are located on the cover of the cylinders, which is a part of the pump body. Therefore by the rotation of the rotor, as the cylinders pass over the ports, they suck or discharge the liquid respectively.

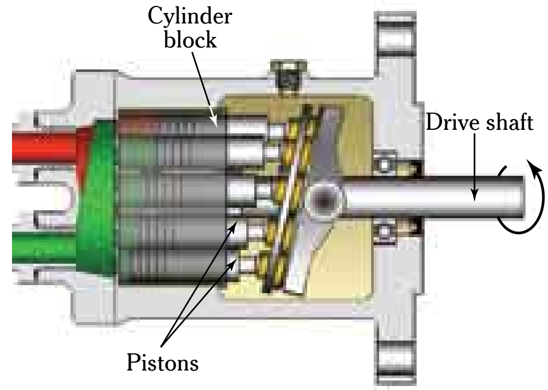


Fig. 2.38
Axial piston pump.

The free end of the rod of the piston abuts on suitably shaped slipper pads that slide on a disk surface (swashplate), with a plane perpendicular to the rotor rotation axis. The disk has the ability to change its tilt to the rotation axis under the influence of an external control device, which sets the length of the active stroke of the pistons and hence the flow capacity of the fluid that the pump handles.

When the cylinder-piston system rotates and the disk position is in vertical plane to the axis of the drive shaft, the pads slide on the disk without the reciprocating movement of the pistons, so no suction or discharge occurs (fig. 2.39). Discharge

(a)

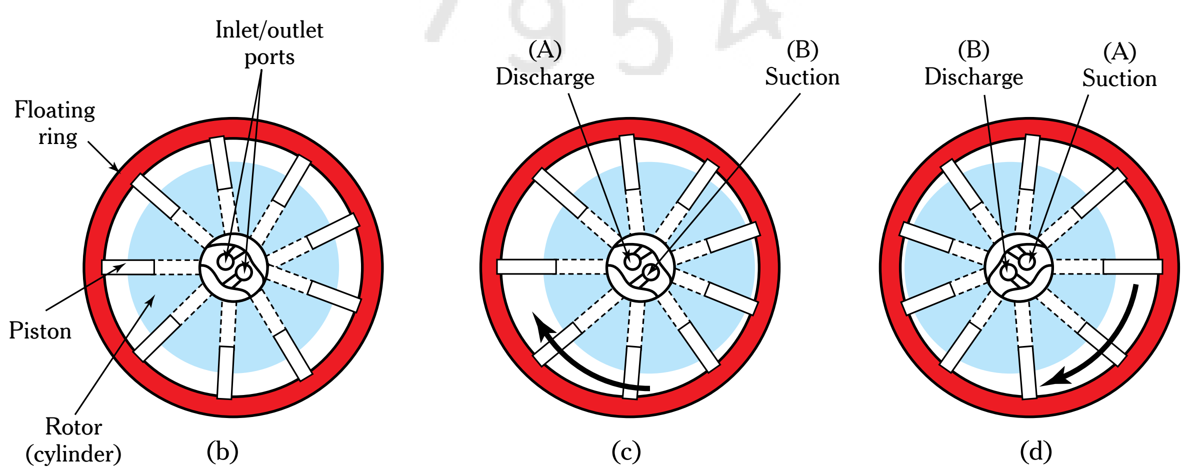


Fig. 2.37
(a) Radial piston pump, (b), (c) and (d) phases of operation.

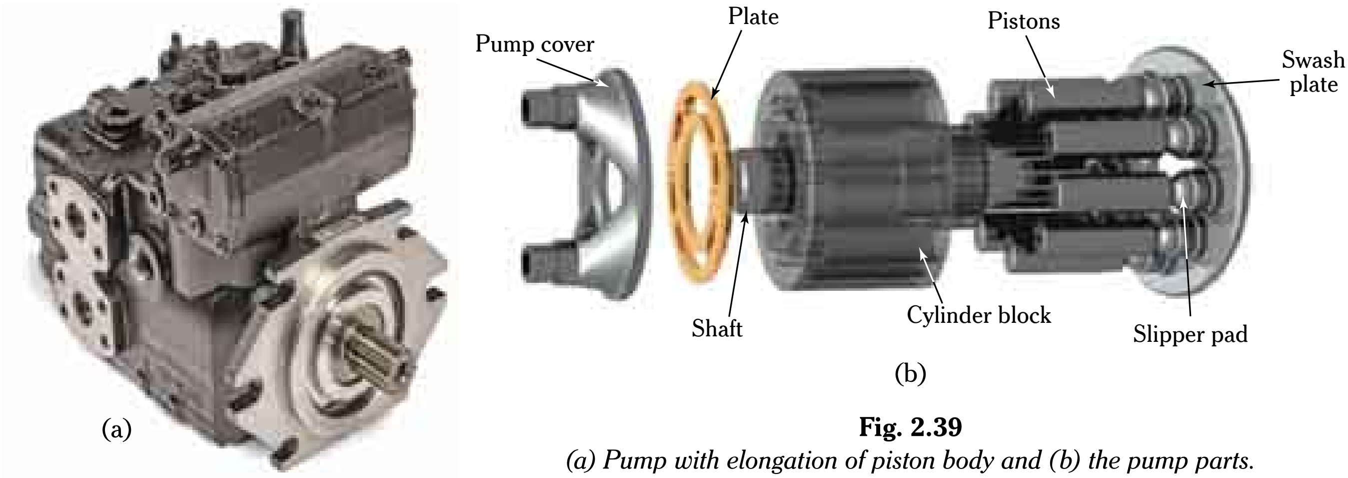


Fig. 2.39
(a) Pump with elongation of piston body and (b) the pump parts.

and suction is achieved in the pump ports by the relative displacement of the pistons from the pump cover, as they approach or are removed respectively from it, since they follow the tilts of the slide disk. Thus, the capacity of the pump changes according to the disk tilt, and provides more or less quantity of the fluid to the system.

The high discharge pressure by the pistons of the pump, as well as the high rotation speed which may be developed, create the appropriate conditions for the usage of these pumps in the supply hydraulic oil for the operation of ships steering systems, hydraulic winches, etc.

Depending on the construction of the pump, the same results in the supply could be achieved if the tilting is given to the pump rotor, besides being given to the sliding disk where the pads slide. Also, the pistons of the pumps may have a rod, or their bodies may be elongated, thus creating spherical ends, which are in direct contact with the slide pads that slide on the variable tilting disk.

8) ***Peristaltic pumps*** (fig. 2.40). These pumps constitute one more type of positive displacement rotary pumps. Pumping, as well as leaking prevention, either between discharge and suction, or between the high and low pressure compartments during liquid transfer, is based on the elasticity of the flexible member. This can be a hose, a flexible impeller, or a resilient sleeve, distinguishing the pumps as follows:

a) ***Pumps with flexible tube*** (or hose), consisting of a rotor, a shell and a tube installed circumferentially within the pump casing [fig. 2.40(a)]. The drive unit transmits the motion to the rotor, which is constructed with circumferential gaps. These hollows aim to create chambers along the tube, for transferring the fluid. By the rotor rotation the tube is squeezed gradually, achieving pumping and transfer of liquid. Because a specific amount is transferred in each displacement of the

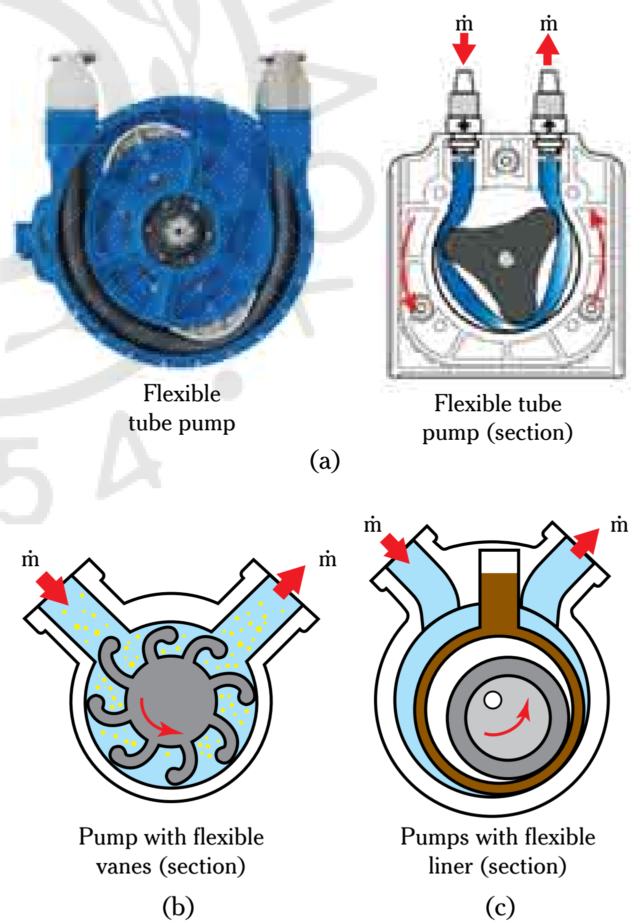


Fig. 2.40
Peristaltic pump types.

rotor relative to the tube, the pumps are used as dosing pumps, providing a constant volume of the pumped fluid for each revolution of the rotor.

b) ***Pumps with flexible vanes*** (or fins), in which suction and discharge is achieved by a rotor consisting of flexible elastic flaps located radially around the driveline axle [fig. 2.40 (b)]. The shell, which surrounds the rotor, together with the elastic flaps create chambers for transferring the fluid from the suction to the discharge, located in the periphery of the housing. They are usually dependent pumps, because the drive shaft of the rotor is driven by a suitable arrangement of the machine they serve. They are used for water circulation in roll cooling system in low horsepower engines, in compressors, etc.

c) ***Flexible liner pumps***, consisting by an eccentric, metal or hard synthetic material rotor, which is rotated within a rubber liner [fig. 2.40(c)]. The rubber liner is installed inside the pump body, so that, between liner and body, a peripheral chamber is created for transporting the liquid. With the rotation of the eccentric rotor, the inner surface of the rubber liner is pushed on the inner periphery of the pump body, and, as a result, in each rotation an amount of the liquid is discharged from the suction to the discharge. These pumps are used to provide a specified quantity of liquid chemicals (as dosing pumps) and are operated where low speed is needed for low flow requirements.

9) ***Circumferential piston pumps.*** These pumps consist of only two rotors moving in the inner part of the liquid movement casing (fig. 2.41). Each rotor is consisted of the shaft where a piston is positioned, and its movement is realized by two rotating shafts. The pistons are made of steel alloys suitable for the liquids being moved. Also, the pistons, with suitably formed gear cut internally at the center of their hub, are stably mounted on the spline on the edge of the moving shafts. Hence, both pistons are moved with the same rotation speed. The shafts come out through the casing, and one of them is connected to the drive engine. The movement of the other shaft is achieved with external timing gears. Therefore, there is no contact between the piston surfaces, as in pumps with lobes and screws, rendering the circumferential piston pumps more reliable for pumping liquids with small suspended particles.

The suction during pumping is achieved by circumferential pistons, since the volume created between the pistons and the casing is increased, as they rotate. Thereby, the liquid enters through the suction port. Subsequently, as the rotation of the pistons continues, the volume is gradually reduced and causes the liquid to come out by discharge, where it is pressured to the piping system.

The sealing required between the pistons, as well as among those on the inner surface of the casing and the covers, is achieved with very small clearances, whereas the sealing of the shafts at their outlet from the casing, where the liquid is transferred, is accomplished by mechanical seals. The shafts are supported by ball bearings, which are suitably installed to the casing of the pump.

These pumps are used in a wide range of applications because of the relatively low intensity at rotating speeds, while they are suitable for pumping viscous liquids, since they achieve high discharge pressures, compared to other pumps. Furthermore, they are simple in terms of their design and maintenance.

2.8 Flow rate of rotary pumps.

Rotary pumps, as described in the previous paragraphs, are constructed in various types. The calculation of the fluid flow rate is a function of the geometrical elements of the pump rotor. So:

1) For gear pumps with two gears (straight or helical), the volume of liquid (V) which is moving by the rotor in one turn of each wheel is the product of the interval cross sectional area between two consecutive gear teeth (A), the tooth length (l) and the number of gear teeth of each wheel (z).

The sectional area of the space between two consecutive teeth (A), is the product of the arc of the base circumference between two consecutive gear teeth and the height of the tooth (fig. 2.42).

Since the pump has two wheels, then the mobile liquid volume is given by:

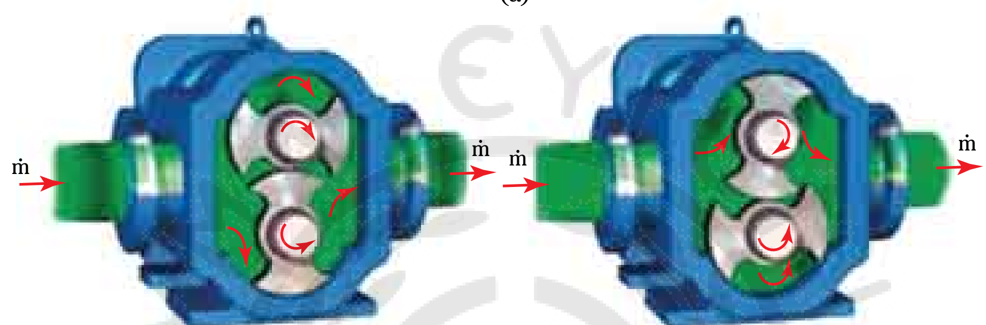
$$V = 2 \cdot A \cdot \ell \cdot z \qquad (2.18)$$

Then, the theoretical volume flow rate $\dot{V}_{th}$ for pump rotational speed (n), results from the relationship:

$$\dot{V}_{th} = V \cdot n \qquad (2.19)$$

2) For screw pumps, the theoretical volume flow rate is calculated by multiplying the volume of liquid conveyed and the number of the rotations. Fur-

(a)



(b)

Fig. 2.41
(a) Circumferential piston pump,
(b) phases of operation.

thermore, when the form of the rotors is helical, the annular volume of transferred liquid is given as:

$$V = \frac{\pi(D^2 - d^2)}{4} \cdot \beta \qquad (2.20)$$

where, D is the external diameter of the screw in m, d is the internal diameter of the screw in m and β is the screw pitch in m.

So the theoretical volume flow rate for (n) rotational speed is given by the relationship:

$$\dot{V}_{th} = 2 \cdot V \cdot n$$

By replacing the volume the theoretical flow rate can be written as:

$$\dot{V}_{th} = 2 \cdot \frac{\pi(D^2 - d^2)}{4} \cdot \beta \cdot n \qquad (2.21)$$

The actual flow rate of the pumps due to losses is given by the product of the theoretical flow rate V_{th} and the volumetric efficiency (η_V) as follows:

$$\dot{V} = \dot{V}_{th} \cdot \eta_V \qquad (2.22)$$

where: V_{th} is the theoretical flow rate, and η_V the volumetric efficiency.

Table 2.6 shows the advantages and disadvantages of various types of rotary pumps.

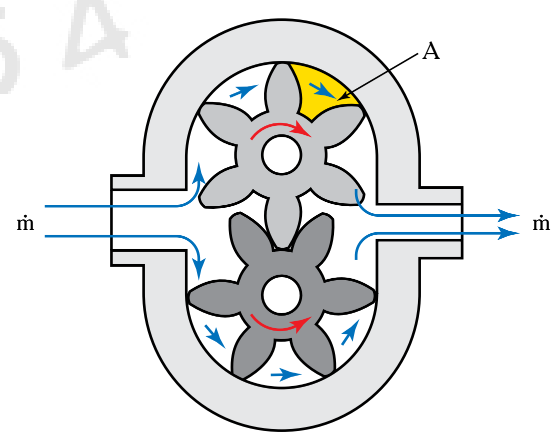
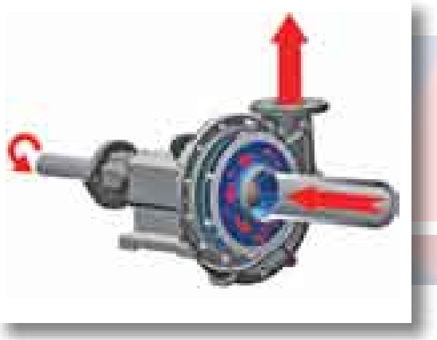


Fig. 2.42
Cross-sectional area between two consecutive teeth (A).

Table 2.6: *Advantages and disadvantages of rotary pumps.*

Pump type	***Advantages***	***Disadvantages***	***Pumped fluids***
Gear	They consist of fewer moving parts and have a simpler construction as compared to the other types.	They lack the ability to pump liquids containing suspended solid particles.	Pure fluids and viscous fluids.
Lobe	Indicated as wastewater pumps, and their pumping capacity is mild.	They present low performance for lifting low viscosity liquids.	Viscous fluids with small solids content.
Peristaltic	No sealing on the drive shaft is required, so reducing leakage as well as pollution probabilities, and they have low maintenance costs.	Are not appropriate for pumping liquids with a relatively high viscosity.	Corrosive liquids, with low viscosity, containing small solids.
Screw	Suitable for high flow demands, providing laminar flow, and the fluid during pumping does not form emulsions.	They are high cost pumps and must be big in size in order to achieve high pressures.	Viscous liquids, without containing suspended solid particles.
Vane	Effective in vacuum creation, and they perform well even without lubricant.	The rotor consists of several components, their use is not recommended for fluids with high viscosity, they do not develop high pressure.	Low viscosity liquids, gases, solvents, aqueous solutions, alcohols, aerosol, etc.
Piston	They have the ability to develop high pressure while operating at high speed. They can invert the flow to the piping system without interrupting their operation.	Pumps with several rotor parts, high cost.	Hydraulic oils and low viscosity fluids handling at high pressure.
Liquid piston	Appropriate for the removal of air from the suction system or from the inner casing space of centrifugal pumps.	They present low performance, and are not able to pump different liquids, because the pumped liquid is mixed with the operating fluid.	Gases.
Eccentric helical rotor	They are effective in vacuum creation and for pumping viscous liquids, they have good performance even without lubricant.	They are not suitable for pumping liquids with abrasive properties.	High viscosity fluids with content of small solids.
Circumferential piston	They are able to develop high pressure, and operate at high or low speeds.	They are not suitable for pumping liquids with abrasive properties.	High viscosity fluids with content of small solids.



CHAPTER THREE
Dynamic pumps

General introduction.

In the most common types of ***dynamic*** or ***kinetic type pumps***, one or more rotors with vanes are attached on the pump shaft that rotates inside the casing (pump body). The rotor with the blades (or vanes) mounted on it is called ***impeller***. The liquid enters the center of the impeller and acquires kinetic energy as it is pushed by the blades. There are two major categories of dynamic pumps, namely the rotodynamic and the special effect ones, as noted previously in the classification shown on page 6.

In centrifugal types of pumps, which are the most widespread kind of rotodynamic pumps, through their operation, the distance between the elementary mass of the liquid and the rotation axis is increased, i.e. the rotation radius increases (fig. 3.1). According to the laws of circular motion, the linear velocity of rotating elementary mass of liquid is proportional to the rotation radius which is expressed by the formula $v = \omega \cdot r$. Consequently, the kinetic energy of the liquid increases, as it moves away from the center to the casing. The design is such that, before the liquid exits from the pump, the cross section that is perpendicular to the flow increases (by single or composite structures that form flow watercourses) and this results in the conversion of a substantial portion of the kinetic energy into pressure energy. The structures that form the flow watercourses are called ***diffusers***. In these types of dynamic pumps, the fluid moves radially (viz. from the center to the periphery). For this reason rotodynamic pumps are called ***centrifugal*** or ***radial flow pumps***. Also, there are rotodynamic pumps where the impeller and the casing are designed in such a manner that the liquid moves, either parallel to the direction of the pump shaft (these are called ***axial flow pumps***), or using a design of intermediate type which combines radial and axial flow principles (these are called mixed flow pumps). Another special category are the peripheral flow pumps, vortex or regenerative turbine pumps, where the liquid enters the periphery of the impeller and largely moves therein.

The common feature of dynamic pumps is the initial transformation of the mechanical work that the pump transfers to the fluid into kinetic energy (within the pump) and then into pressure energy (the pump outlet). This feature incorporates within the category of dynamic pumps non rotating machinery such as ***jet*** or ***ejector pumps***, pressurized air or gas lift pumps, hydraulic ram pumps and electromagnetic pumps.

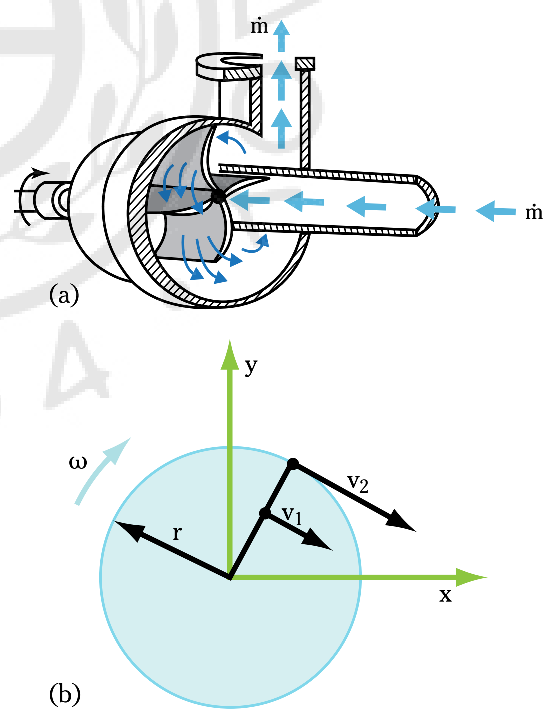


Fig. 3.1
(a) Schematic and (b) linear representation of the proportion between linear speed and rotation radius (radius from the axis), wherein as the radius increases the linear velocity increases too.

An additional feature of dynamic pumps is the fact that the supply from the suction part to the discharge part is continuous and constant.

The functional and structural characteristics of these pumps are presented in the following paragraphs.

3.1 Dynamic pumps.

The classification of dynamic pumps on the basis of the way they transform mechanical work to kinetic energy (and then to pressure energy) is presented on page 6.

Dynamic pumps are subdivided into two main categories, the ***rotodynamic*** and the ***special effect*** pumps.

Rotodynamic pumps comprise a major category of pumps according to the basic pump classification system we have applied. Their operation is based on the alteration of the kinetic state of the fluid through the conversion of the imparted energy. Initially, mechanical work is transformed into kinetic energy, and subsequently, part of the kinetic energy is transformed into pressure energy in the outlet/discharge of the pump. So, in accordance to the ***Bernoulli principle***[1], the large flow velocity which is initially imparted to the fluid by the impeller is then converted into pressure. The size of this conversion depends on the pump dimensions, the shape and texture of the chamber surface where the liquid passes through, and the rotor operating speed.

The rapid development of dynamic pumps during the past decades, their efficiency, and the wide array of applications they serve in, can be attributed to the following reasons:

1) A high performance is achieved by using relatively small volume and light weight pumps, in relation to the potential amount of transferred liquid.

2) They have relatively low manufacturing costs.

3) Their rotary motion is continuous and uniform, without any fluctuations in pressure and flow rate (the opposite happens in reciprocating piston pumps where fluctuations are actually present).

4) They have a simple design.

5) They can easily be connected to various types of motors.

6) They can handle large quantities of liquids.

For these reasons, the rotary dynamic pumps, especially centrifugal ones constitute up to 80-90% of the pumps installed in the oil industry. However, it should be noted that their widespread use became possible after the major construction developments achieved in the field of electric motors, steam-turbines and internal combustion engines. Prior to these developments, the use range of positive displacement pumps was larger, due to the fitting response of their characteristics to previous operating requirements.

A pump consists of fixed parts and moving parts. The fixed parts are the casing (***volute casing***), the seal (***gland*** or ***retainer***) which seals the rotor shaft, and the ***ball bearings*** (or supporting bearings) of the rotor shaft housed internally or externally in the pump body. The moving parts are the rotor or ***impeller*** and the ***rotation shaft*** of the impeller.

The free end of the rotor shaft is suitably connected to the pump motor through a coupling or a gear. As the impeller rotates on its shaft, it displaces the liquid present inside the pump casing and a new quantity of liquid passes through the impeller by substituting the amount of displaced liquid. The liquid passes into the impeller and its casing through the suction and discharge openings, which are on the pump casing. Under the influence of the impeller the speed increases, by mechanical action, and the energy goes to the pumped liquid. This creates a constant flow from the suction to the discharge, and the liquid is supplied to the pipe network.

To maintain the pressure inside the pump, a seal is used. Appropriate seals are placed on connecting points of the casing and on the entering point of the shaft into the casing.

Rotodynamic pumps are classified in various ways, such as according to:

1) The liquid flow direction under the influence of the impeller.

2) The construction design of the pump casing.

3) The entrance of the liquid into the pump impeller.

4) The type of impeller.

A further subdivision of these pumps is performed by:

1) The stages which depend on the number of impellers enclosed in a pump and attribute energy to the fluid during their operation. The subdivisions are, single-stage pumps when there is one impeller,

[1] The ***Bernoulli principle*** states that an increase in the speed of a fluid occurs simultaneously with a decrease in pressure or a decrease in the potential energy of the fluid.

two-stage pumps with two impellers, three-stage pumps with three impellers and multi-stage pumps with more impellers.

2) The arrangement of the rotary shaft (horizontal and vertical pumps).

3) The type of driving device which is used for the pump operation steam, diesel, electric and hydraulic energy driven pumps.

3.2 Classification of pumps based on the liquid flow direction.

According to the criterion of fluid flow direction, dynamic pumps are classified into centrifugal or radial flow, axial flow or propeller, mixed flow and peripheral flow or regenerative pumps.

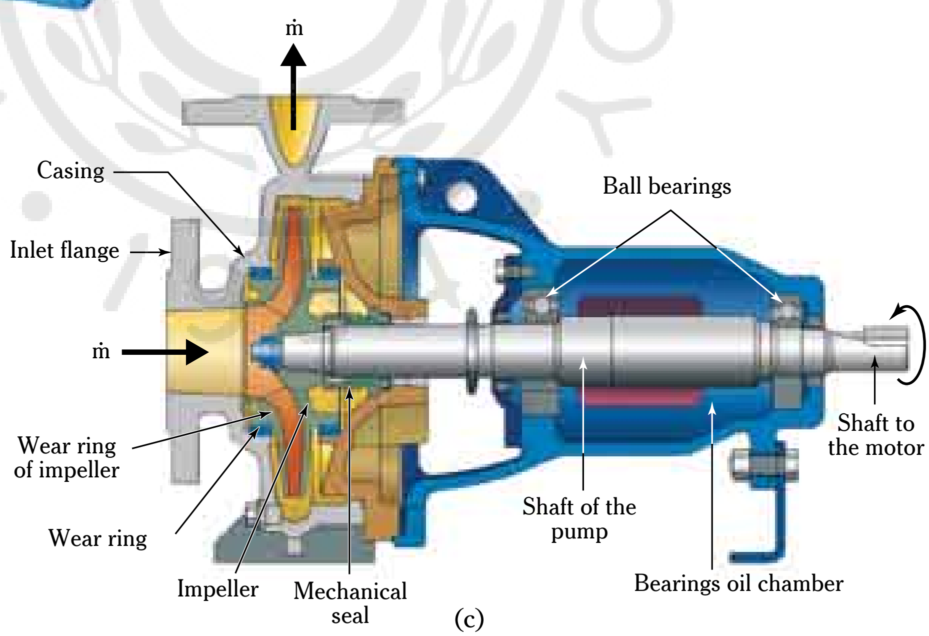
3.2.1 Centrifugal or radial flow pumps.

Centrifugal pumps (fig. 3.2) are the most common type of dynamic pumps in use today. The number of applications results from their ability to combine satisfactorily the advantages of dynamic pumps, such as high performance and relatively simple construction, with the attributable amount of ***given height***[2] (or manometric head).

Blades or vanes are placed on the impeller hub of these pumps and the whole impeller is enclosed within the casing (fig. 3.2). The impeller is placed on the rotating shaft of the pump and driven by the drive machine (usually an electric motor). As the impeller rotates, the liquid enters the center (eye of the impeller), and then accelerates as pushed from the surface of the vanes at a speed which depends on the rotational speed and shape of the vanes. In centrifugal pumps the liquid is influenced by the impeller surface and ejected to the volute casing due to the centrifugal force. The volute casing actually consists the pump periphery. Then, the liquid is driven to the discharge pipe. The pumping is achieved by the vacuum (low pressure) developed at the center of the impeller as the liquid moves to the outer periphery of the impeller under the effect of the vanes. The developed vacuum is supplemented by an amount of

(a)

(b)



(c)

Fig. 3.2
(a) Horizontal (b) vertical centrifugal pump and (c) illustration of different parts of a centrifugal pump.

[2] Head height (h) is called the vertical height, which a pump has the ability to reach (see par. 4.5).

new liquid through the suction pipe to the low pressure point, near the center of the impeller (eye of the impeller). The new liquid passes again over the impeller in a direction towards the periphery creating a constant flow from the suction to the discharge. Due to this direction of liquid flow the centrifugal pumps are also called ***radial flow*** pumps.

The pressure and manometric head developed by a centrifugal pump are directly related to the diameter of the impeller, the number of impellers, the size of impeller center where the liquid enters and the shaft speed rotation where the impeller is installed. Also, the pump flow rate depends on the width of the impeller outlet (Ch. 6). Therefore, depending on the pumping requirements, the discharge head and the supply are main factors that influence the magnitude of the motor power to be used. As a result, the larger the amount of liquid that must be pumped, the greater the energy amount that is required.

Also, when a centrifugal pump operates with constant speed (n), it has a certain flow rate at a minimum head height (minimum head height is when the friction resistances inside the pump are taken under consideration). Hence, by increasing or reducing the pump rotor speed, the liquid flow rate attributed to the system increases or decreases. If the number of the shaft revolutions changes from n_1 in n_2, then a corresponding change will occur to the flow rate as $\dot{V}_1$ to $\dot{V}_2$, and the following relation would apply:

$$\frac{n_1}{n_2} = \frac{\dot{V}_1}{\dot{V}_2} \quad (3.1)$$

In view of the above, if we have a constant number of revolutions (n) and the discharge head changes, alterations are accordingly observed in the flow rate. (fig. 3.3). Specifically, for pump head height (H_0) the pump supply becomes nonentity. This height, where no discharge occurs from the pipe, is called ***head of the pump***, and it is also known as the "No flow or zero flow height" ($\dot{V}_0$), "Lift" "Shut off", etc. It does not matter which pipe size is used, as this height remains the same. Respectively, if the head height (or manometric head) is reduced, the flow rate increases.

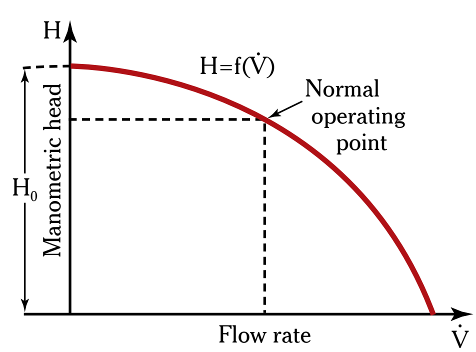


Fig. 3.3
Flow rate changes as the head increases.

The manometric head is related to the pressure which is developed by the pump, since during the operation of the pumps we have a geometric lifting height (z_g), which is the height difference between the free surface of the pumped liquid and the discharge point. The height difference (z_g) will cause pressure at the interior of the tube bottom, equal to the pressure of the water column (z_{WC}). Consequently, for water to flow in a pipeline, there should be given an additional pressure that can overcome:

1) The resistance to the water flow inside the pipe and in addition resistances that are presented by the pipeline fittings (e.g. the suction valves, valves installed in the network, the elbows of the network piping, etc.).

2) The resistances of water flow to the pump components.

So, for a constant speed (n_1), if we change the manometric head of the pump, inverse change will be observed at the pump flow. By representing these values in a coordinate plane, where the abscissa is the flow rate and the ordinate the respective discharge manometric heads, the curve form given by [$H = f(\dot{V})$] is shown in figure 3.3.

The centrifugal pumps, being dynamic pumps, do not have a stable liquid pumping volume. The volume depends on the suction depth and the discharge pressure (Ch. 6). Accordingly, the lower the place of pump suction depth position, the smaller the quantity of liquid that passes by the pump. Respectively, the pressure of the quantity of liquid pumped decreases as the pressure increases, when the pump discharge is performed on a pipeline or a tank. The attainment of bigger discharge pressure by centrifugal pumps can be achieved either by using pumps which have a large number of speed rotations and could exceed the pressure of 10 bar, or by multi-stage pumps where impellers are used in a series. In these pumps the discharge of one impeller is the suction of the next one.

For these reasons, it is essential that a pump that will be installed in a pipeline must be designed to meet the specific pumping requirements. Also, in order to achieve the maximum pump efficiency, the shape of the hydraulic passageways of the impeller and the casing are important.

Under normal circumstances, the operating speed of centrifugal pumps varies from 1 500 rpm to 3 000 rpm, whereas some types are designed to operate approximately at speeds of 5 000 – 25 000 rpm.

3.2.2 Axial flow or propeller pumps.

Axial flow pumps are dynamic pumps, meaning they utilize fluid momentum and velocity to generate pump pressure. The pump usually consists of an impeller with a few number of vanes, typically only three or four vanes, which are mounted on a hub and the hub is mounted on the pivot axis (fig. 3.4). So, such pumps are called axial flow or propeller because the impeller is propeller–shaped. The position of the vanes is angled in the vertical plane of the rotation axis, while the entire system rotates in a tube which constitutes the pump casing. The liquid is not centrifuged, but, as it enters through the suction, it is pushed by the rotating vanes in such a way that the pumped fluid exits axially, rather than radially. The result is the augmentation of the kinetic energy of the liquid and the conveyance of the liquid in a direction parallel to the rotation axis.

In order to increase the liquid flow velocity before the liquid reaches the impeller, the casing is constructed with a slightly smaller suction diameter. The casing diameter expands slightly after the impeller. Hence, a decrease in fluid velocity is caused, with a corresponding increase in pressure, due to the conversion of part of the velocity (dynamic energy) to pressure (velocity and pressure are inversely proportional).

The rotation speed of the helical impeller is high, and, as a consequence, intense turbulence is developed in the flow of the pumped liquid, despite the axial direction of the liquid. This turbulence causes a reduction in the pump efficiency. In order to reduce the turbulent flow and to increase the effectiveness of the pump, fixed guide vanes are mounted on the pump suction (fig. 3.4). These vanes have an inclination that is opposite to that of the propeller and are used to ensure the smooth passage of the fluid through the pump, eliminate circulation and achieve a reduction of turbulent flow.

The fixed vanes are called ***diffusion vanes***. Axial flow pumps are used in applications that require very high flow rates at very low amounts of pressure (low discharge head). Due to pump construction, the suction is achieved only when the pump is positioned below the free level of the liquid to be pumped, while

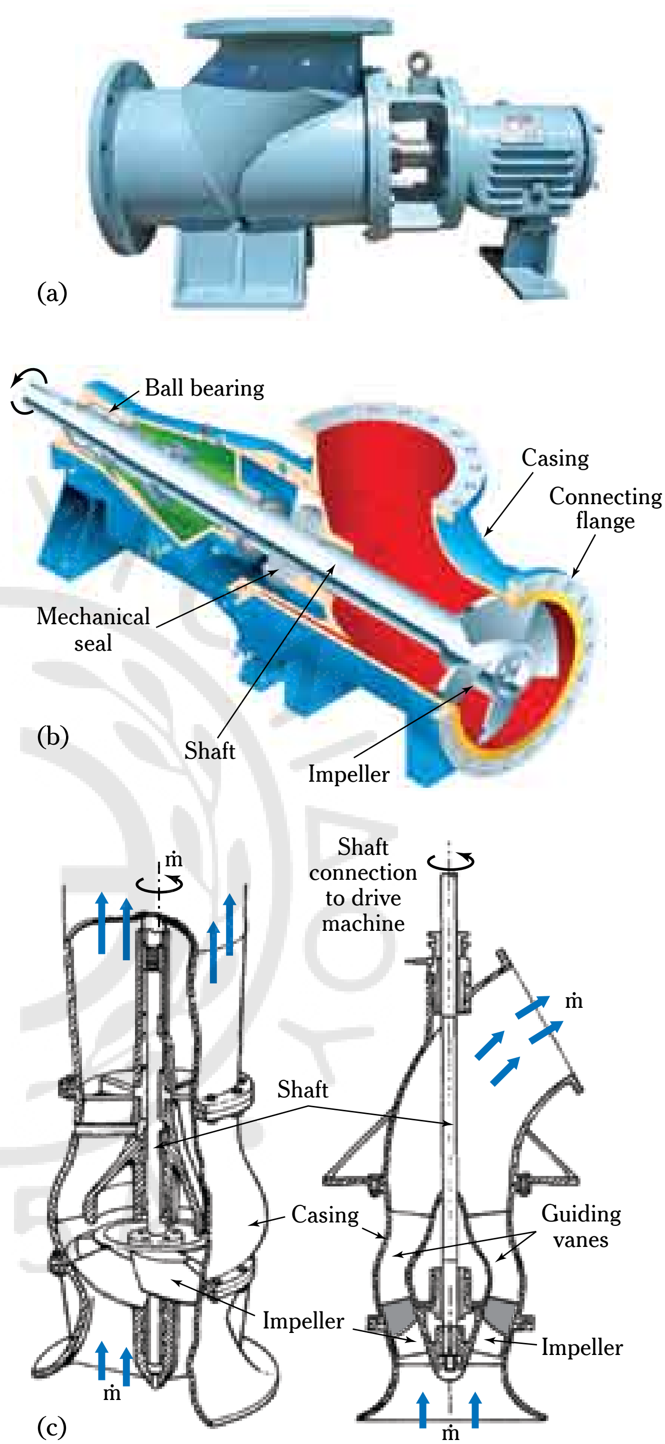


Fig. 3.4

(a) Axial flow pump (b) Section of horizontal arrangement and (c) Section of vertical arrangement axial flow pump.

the discharge can reach a pressure up to 1.5 bar.

The pump operation can be reversed either by reversing the rotation of the shaft by a drive machine or by using a variable pitch propeller and adjusting the position of reversible blades. Therefore, the suction becomes discharge and alternately, achieving, with fast operation and high precision, the desired quantity supplied into a tank or to the sea. Examples are the axial flow pumps that are used as ***ballast pumps***, as they provide the possibility to adjust the ship trim accurately due to the rapid variation of the sea water supply (see Ch. 8). The arrangement of the pumps could be vertical or horizontal, with the same functionality.

3.2.3 Mixed flow pumps.

The operation and shape of ***mixed flow pumps*** combines the characteristics of the two previous pump types. The pump consists of an impeller, with vanes placed at an angle to the rotation axis. The liquid enters the pump axially and debouches thrust with the vanes of the impeller in a diagonal component that is, moving simultaneously axially and radially (fig. 3.5). In this way, the increase of the liquid pressure is simultaneously created with the thrusting of the vanes and under the centrifugal force influence, due to the direction towards the circumference of the casing. During the liquid removal from the impeller, the velocity increases, but this augmentation is smaller in comparison to the velocity that develops by the radial flow pumps.

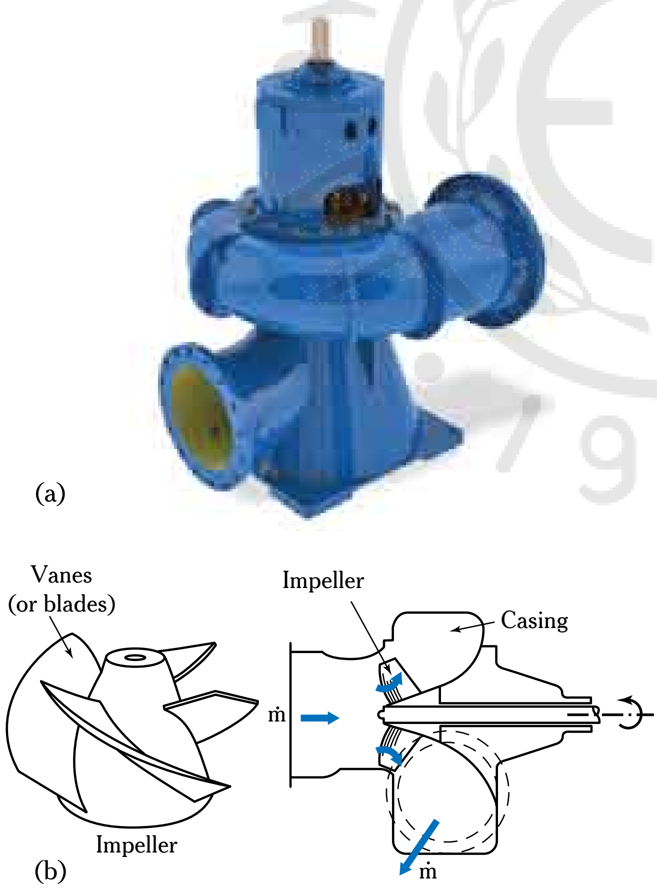


Fig. 3.5
(a) Mixed flow pump and (b) schematic depiction of a mixed flow pump.

The suction is realized from a tube, but the discharge is realized either in a casing or in a tube (or pipe) with wider diameter and appropriately stable guide vanes, which convert the kinetic energy into pressure energy.

The installation of guide vanes provides the possibility of placing more impellers on the same axis, thus accomplishing an increase in the energy attributed to the fluid, as long as each impeller is supplied with liquid by the previous one. Thus, multi-stage mixed flow pumps, that have satisfactory height of discharge and high volumetric flow rate, are created.

3.2.4 Peripheral flow or vortex pumps.

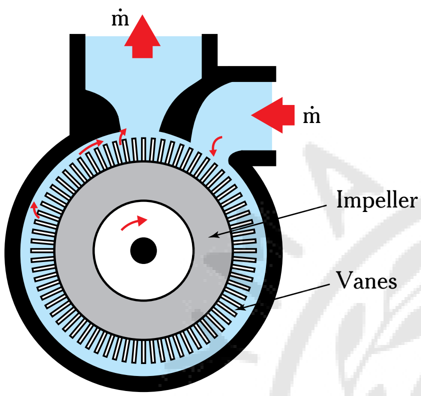
The peripheral flow, regenerative or ***vortex pumps*** are characterized by the consecutive move of the liquid from the impeller circumference to the casing and from the casing to the impeller circumference. The configuration of the casing and the impeller is presented in figure 3.6. The impeller comprises vanes or blades, which are plane (flat) and placed circumferentially at its edge and in radial arrangement. The pumped liquid whirls in the space which is created between two vanes, and rotates with the impeller. This swirly mass between the vanes debouches into the narrow peripheral tube, which exists between the impeller and casing and, subsequently, it enters the place between the vanes again, while the impeller rotates. Thus, the liquid delineates two moves simultaneously, move ***b*** and ***d***, from and to the vanes of the impeller, and move along the casing.

This swirling, as well as the fluid insertion and outing from the vanes, is repeated until the fluid, which is drifted by the impeller, reaches, from the suction, the discharge of the pump.

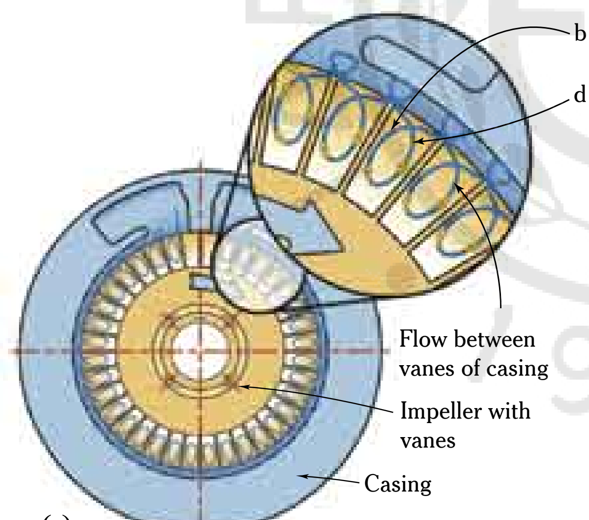
It is noted that in every entry of the liquid between the vanes, its kinetic energy increases due to the swirling and, as a result, it is much higher in comparison to the kinetic energy that is transfused to the liquid in a centrifugal pump. With the pump operation, as the liquid reaches the discharge, because of the gradual depletion in the casing width, its velocity depletes, having as a result (according to Bernoulli's theorem) its transformation into pressure towards the pipeline.

The functional particularity of this type of pumps

(a)



(b)



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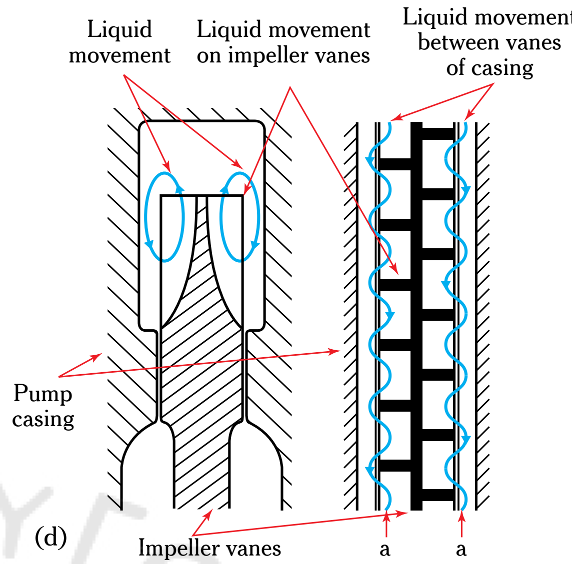


(d)

Fig. 3.6
(a) Vortex pump (b) movement of the liquid into vortex pump (c) section with liquid flow between vanes and shell and (d) the motion of liquid.

allows them to develop high pressure in order to discharge the fluid in great height and function with a satisfactory efficiency, when gas quantities, which reach 20% by volume, are drifted in the derived liquid. However, they are at a disadvantage compared to the rest of dynamic pumps, because they do not have a big flow rate when there are solids in the liquid mass and when the fluid has got a high viscosity (viscous).

3.3 Dynamic pump classification based on the casing design.

Dynamic pumps are the ones most commonly used. A dynamic pump consists of rotating and fixed parts. Due to the rotation of the impeller, as liquids are drawn from a tank through the inlet casing into the impeller eye, where pressure is less than atmospheric pressure, suction is realized. The function of the impeller increases the kinetic energy of liquid pushing it outwards, by increasing the angular momentum of the liquid. So, both pressure and velocity are increased within the impeller. Then, the liquid is thrown to the casing by the vanes of the impeller that surrounds it.

Outside the impeller the fluid is collected into a volute casing, where further kinetic energy is converted into pressure energy as it is discharged to the outlet pipe. Furthermore, guiding vanes are located between the impeller and the casing before the liquid reaches the casing. Depending on the construction of the volute casing, the liquid velocity is gradually decreased, while, respectively, according to the flow continuity equation and the Bernoulli equation, its pressure is increasing until the output to the discharge tube. The discharge is realized through a suitable liquid outlet point on the casing perimeter and delivered into the outlet pipe.

Consequently, the shape of the casing affects the proper functioning and performance of the pump, because, depending on the form of the casing, the conversion of the kinetic energy of the fluid into pressure energy is affected, and vice versa. Thus, depending on the purpose which is served by the pump, on the pumped liquid and on the pump technical characteristics, the construction of the casing must satisfy the cooperation with the impeller, in order to optimize performance and to achieve a smooth operation without vibrations. Depending on the shape of the casing, these pumps are divided into ***spiral casing pumps***, ***diffuser type pumps*** and ***mixed flow type casing pumps***.

3.3.1 Volute casing pumps or spiral casing pumps.

In ***volute casing pumps*** (or ***spiral casing pumps***), the duct diameter, which constitutes the pump casing and encircles the impeller, increases progressively creating a spiral-shaped surrounding duct. Due to the increase in the sectional area of the spiral-shaped duct, the velocity of the liquid that seeps through it is depleted gradually until the outlet to the discharge pipe, whereas the pressure increases. The impeller may be placed in the center or eccentrically of the casing, whereas there is a gap-clearance (small distance) between the impeller and the casing, deterring the touch of movable and fixed parts, when the pump functions. Therefore, the pump acquires corresponding characteristics, depending on the impeller position in relation to the casing. These positions are:

1) The pump impeller is concentric to the circular casing; these are defined as ***cycle casing pumps***. The distance of the impeller perimeter from the casing is circumferentially stable.

2) The impeller is placed eccentrically; these are named ***volute casing pumps.*** In them, the impeller distance from the casing is modified circumferentially, so that it is minimized near the outlet point of the fluid from the pump, as a 'pinpoint' is formed from the casing. This point is named ***cutwater*** or ***tongue*** [figs. 3.7(b) and 3.8], because of its shape.

3) The impeller is placed in such a way that the pumps constitute an intermediary form of pumps, which combines the characteristics of the two previous types of cycle casing and volute casing pumps [fig. 3.7(c)].

As the volute casing pump operates, the distribution of the liquid pressure around the impeller, as well as the pressure which develops at discharge, are not even, but depend on the impeller revolution and the opening of the discharge valve. This creates function conditions that must be confronted, because they affect the pump efficiency.

These conditions are the following:

1) The ***unequal distribution of pressure in the pump interior***, which creates radial thrust around the impeller. The radial thrust is exerted on the bearings of the rotary shaft and of the impeller, causing quick detriment of the bearings and bending of the rotary shaft, as well as the detriment of mechanical seals, having as a result the loss of sealing.

The radial thrust is minimized in pumps of impeller eccentric rotation, when the pump operates in its design conditions (Best Efficiency Point), whereas it increases when it deviates from them in amounts un-

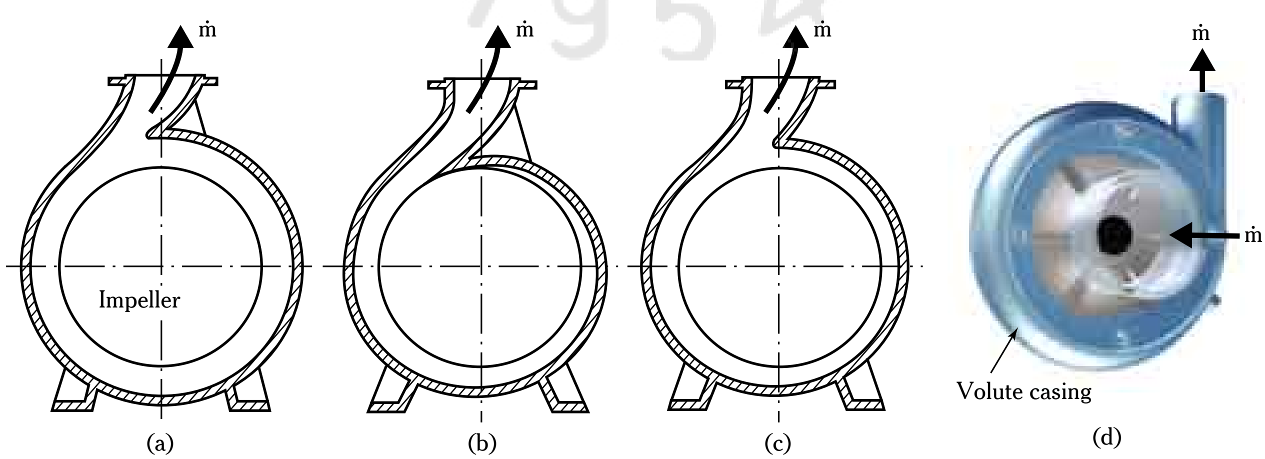


Fig. 3.7
(a), (b), (c) Types of spiral casing pumps and (d) three dimensional image of a spiral casing pump.

der 30% or over 120%. In order for the radial thrust to be confronted, the casing is constructed with a separating wall, so that the double casing created, balances the hydraulic loads through the separation of the flow. Pumps with such kind of casing are named ***double volute casing pumps***. In this case, the separating wall creates two diametrically opposite cutwater points, preserving the balance in the radial thrust which is exerted, causing in that way its depletion (fig. 3.8). Basically, this design consists of two 180° volutes, and a passage, external to the second volute, joins the two into a common discharge.

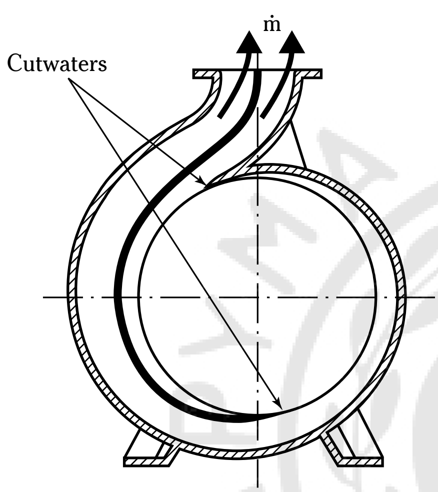


Fig. 3.8
Pump with diametrically opposite liquid cutwaters.

In the concentric pumps instead, where the distance of the impeller perimeter from the casing is circumferentially stable, the radial thrust is maximized in the optimum point of the pump function. Hence, as the radial thrust is taken into account in the pump design conditions, these pumps are constructed in such a way, so that the radial thrust is confronted and all the rest possible fields of function, in which the radial thrust may cause breakdown, are covered.

2) The ***number of revolutions with variations in the valve position*** (open or closed). When the number of revolutions of the impeller is stable, the discharge pressure will be reduced as the discharge valve opens, correspondingly increasing the liquid flow rate. On the contrary, when the discharge valve is closed, the pressure will increase and the flow rate will be reduced, and it will be eliminated when the valve is totally closed.

The closing of the valve causes the internal circulation of fluid, provided that a small quantity is discharged from the impeller in the interior and comes back through the clearances in the suction. This circulation assimilates energy from the axis rotation and turns it into kinetic energy that is spent in swirls and frictions inside the pump and it finally comes to loss, having as a result the overheating of the pump.

3) ***The position of the discharge valve at variable pump speed***. When the opening of the discharge valve is stable, the pressure increases by increasing the rotations of the impeller, whereas when the rotations are reduced, the opposite happens.

3.3.2 Turbine or diffuser type centrifugal pumps.

In turbine or diffuser type centrifugal pumps, the casing has got internal stable guiding vanes, which form divergent ducts with a progressively increasing cross sectional area. While the liquid is ejected from the impeller with very high kinetic energy, it passes through the duct, having as a result the reduction of its velocity and therefore the reduction of its kinetic energy, which is smoothly converted into pressure energy (Bernoulli) (fig. 3.9).

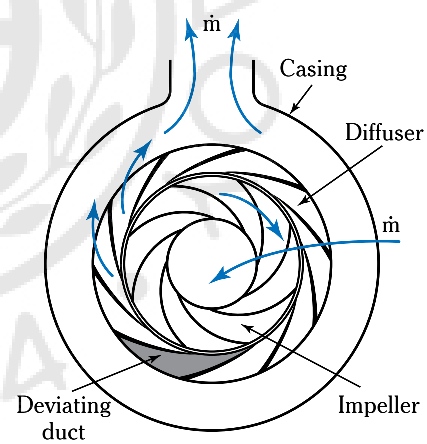


Fig. 3.9
Turbine or diffuser type centrifugal pump.

The casing shape is cylindrical, whereas the impeller is placed in its center. The vanes of the casing are encircled by an annular constant cross section duct, which leads the fluid to the discharge.

The ducts with the progressively increasing cross section area are named ***diffusers*** and contribute to the increase of the degree of pump efficiency, when it works under its design conditions. The increase of the efficiency degree can reach up to 90%. However, when the function conditions deviate from the design conditions, flow detachment from the fixed vane surface is caused, resulting in a significant decrease of the efficiency degree.

3.3.3 Pumps with mixed flow type casing.

Pumps with mixed flow type casing or volute turbine pumps, consist of a casing with diffusion vanes, which is externally surrounded by a volute hollow discharge casing (fig. 3.10). The increase in the liquid pressure is performed by combining the two previous type of pumps structural features. So, in volute turbine pumps, the liquid enters the center of and is driven by the rotating vanes, ejected in the periphery of the impeller.

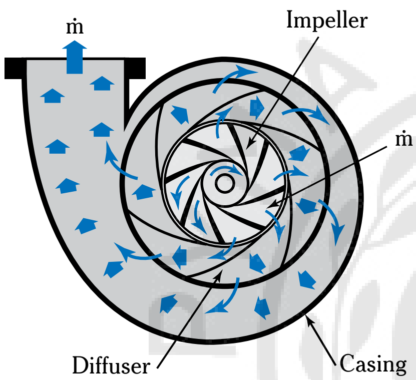


Fig. 3.10
Pump with mixed flow type casing.

Then the liquid passes through permanent diffusers, where gradual decrease of velocity is caused as well as a pressure increase of the liquid.

The combination of permanent diffusers and an impeller creates a system, which is placed eccentrically within the surrounding volute ***casing***. As the area of the volute casing gradually increases, the speed of the liquid that passes through, is, after the diffusers, further reduced, achieving the conversion of additional kinetic energy into pressure energy. This results in further improving the pump efficiency degree.

3.4 Pump classification based on the liquid inlet.

Another way of classifying dynamic pumps is associated with the liquid inlet. In particular, these are:

1) The ***single sided or center suction pumps*** (or one side suction pumps). The liquid is guided by the suction pipe, through the inlet, to the impeller, only on the one side of the pump casing. As shown in figure 3.11, the suction on the casing is adapted, in order to guide the liquid to the impeller center (eye).

The supporting of the shaft on the pump casing is achieved by external bearings. Furthermore, between the suction and discharge area of the impeller, the sealing is accomplished by appropriate rings, placed inside the casing. Onto the casing, to prevent leakages from inside of the pump, a stuffing box is configured where packings are placed. These packings are compressed by a gland (glad follower or pusher). Apart from gland packing, another suitable sealing arrangement suitable for rotodynamic pumps is mechanical seals.

During pump operation, the suction generates axial thrust on the shaft. The appearance of this thrust is caused by the vacuum which tends to displace the impeller towards the suction mouth. This trend is the

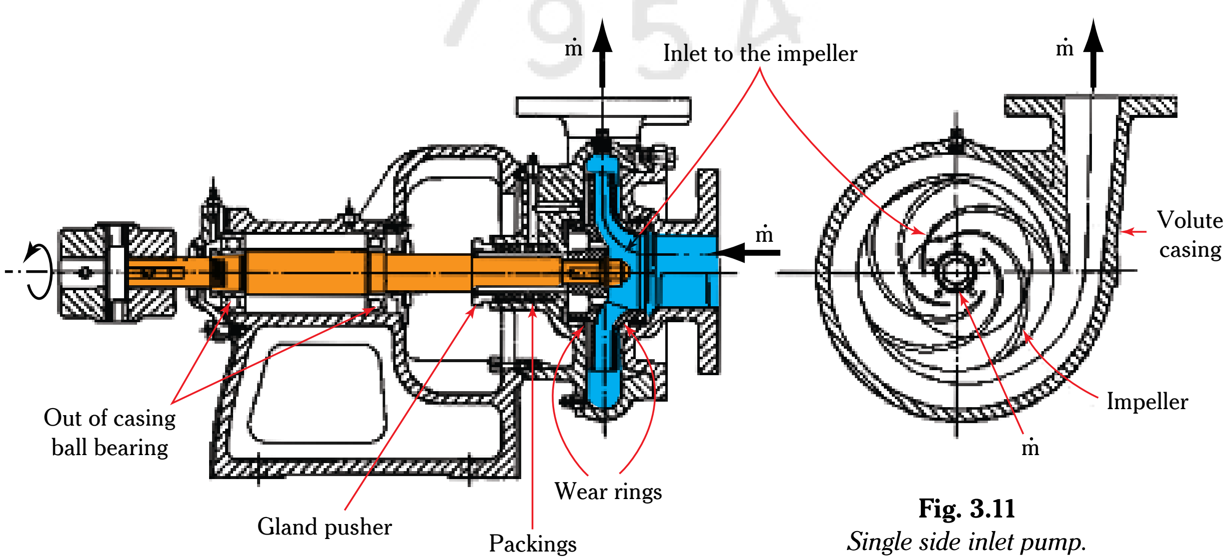


Fig. 3.11
Single side inlet pump.

result of the pressure difference created on both the outer sides of the impeller. A small axial thrust is received by the pump bearings. Otherwise, a greater thrust is compensated with holes in the rotor disk (fig. 3.12), or communication between the suction and discharge chambers.

This communication can be realized by connecting the upper side of the impeller casing (the point of attachment to the drive shaft) and the suction beneath the impeller with a pressure equalizing pipe.

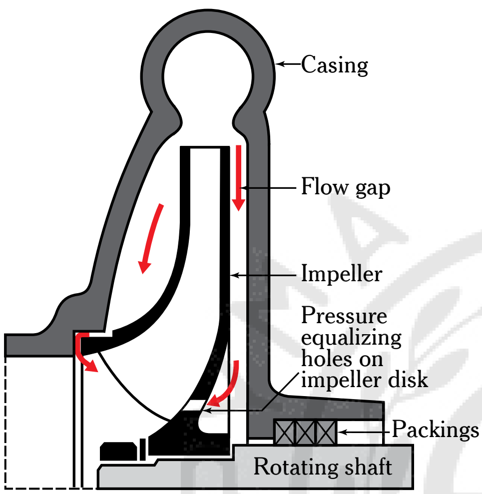


Fig. 3.12
Pressure balance holes on impeller plate.

The flow rates, depending on the type of pump, range from 15 m^3/h to 6 000 m^3/h to a manometric head of 10 m to 180 m. Also, the axial arrangement can be vertical or horizontal.

2) The ***twin volute casing pumps*** (fig. 3.13). The inlet of the liquid in these pumps is performed on both sides of the impeller, through an appropriately shaped casing, which is symmetrical to the perpendicular bisector plane of the pump shaft.

The casing does not have vanes, and is divided axially into two half-shells. The pump shaft may have vertical or horizontal orientation. The liquid inlet and outlet openings are embedded to the lower half-shell, in horizontal arrangement pumps, and in order to facilitate repairs and controls, they are positioned on the fixed part of the split casing.

Given the symmetry presented in the pump construction, the pressure developed on both sides of the impeller is theoretically the same, so as to seem hydraulically balanced, without axial thrust occuring by the liquid passing through the impeller. However, due to unavoidable manufacturing imperfections and the uneven wear of components, a very small axial thrust appears at the pump, which is not enough to cause malfunctions, in practice. These pumps (fig. 3.13) are used to serve high flow rates and are manufactured with diameter impellers analogue to the liquid and the desired discharge amount. The flow rate reaches up to 12 000 m^3/h.

3.5 Dynamic pump classification based on the type of impeller.

Another pump classification is associated with the available type of impeller. The impeller types are characterized by their mode of construction.

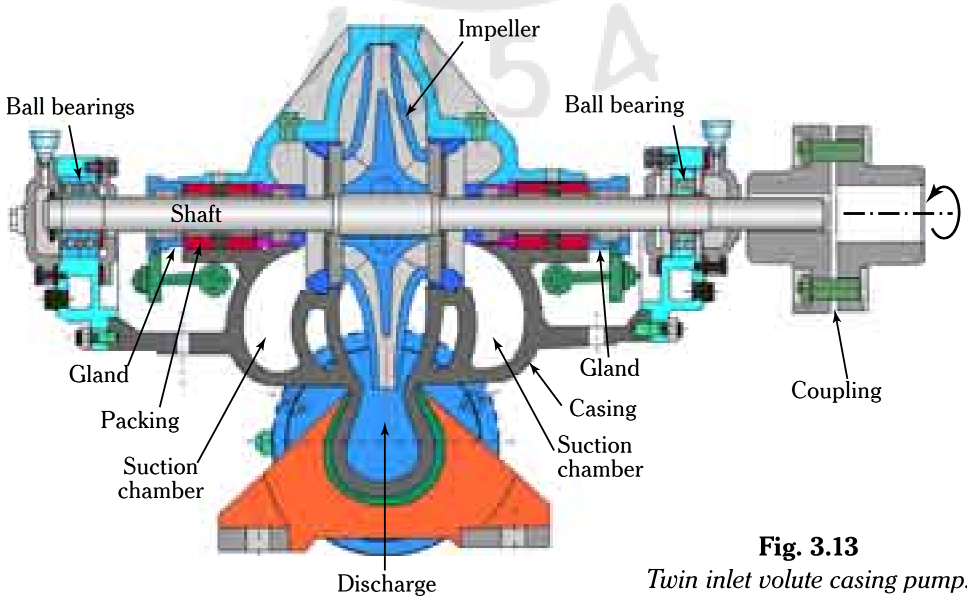


Fig. 3.13
Twin inlet volute casing pump.

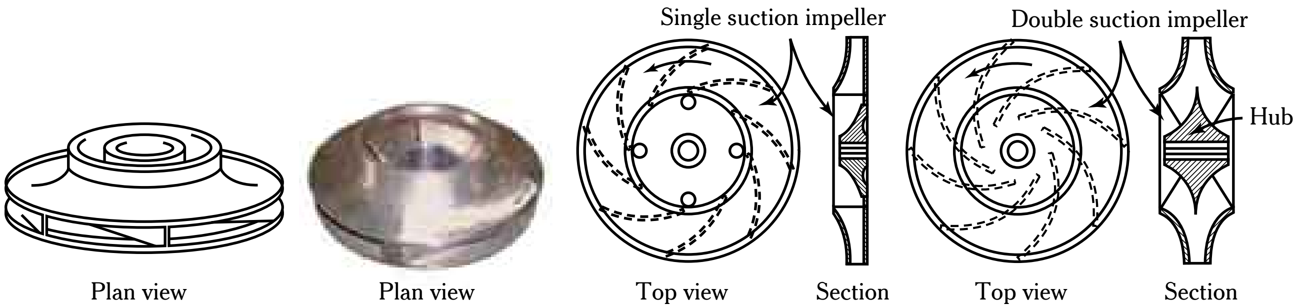


Fig. 3.14
Closed type impellers.

There are three types of impellers: the closed, the semi-open and the open type and pumps are named accordingly. In more detail:

1) ***Closed impellers*** (fig. 3.14). The impellers are constituted of two plates and between them are usually placed 3 to 7 vanes. For single inlet pumps on the one side of the impeller plate, there is a circular opening. This opening is called eye or eye of impeller. Through the opening of the impeller plate, the liquid is inserted into the vanes and pushed to the inner circumference of the casing. Respectively, for double entry pumps, the installed impeller has a circular opening in both plates. The smooth entry of the liquid on the impeller vanes is achieved by designing the diameter of the circular opening such as to coincide with the inner diameter on the casing suction side. In order to separate discharge from suction spaces into the pump casing, the sealing is achieved by ***wear rings*** (or friction rings), installed on the pump casing, that minimize the gap between the impeller and the casing. These types of impellers have high efficiency, develop high pressures and have low axial thrust.

2) ***Semi-open impellers*** (fig. 3.15). Semi-open impellers are constructed with a circular plate attached on one side of the vanes since there is no plate on the suction side. The vanes form an extension of the hub. They have higher efficiency than closed type impellers, due to the elimination of frictions that are produced by the plate at the inlet side of the liquid. Such impeller types are used even when the liquid contains suspended particles or fibers because they allow the passage of large diameter solids. Furthermore, the distance of impeller vanes from the casing at the open side should be minimum in order to achieve the reduction of leakage between the vanes and the casing. The disadvantage of these impellers, compared to closed-type ones, is the higher axial thrust.

3) ***Open impellers*** (fig. 3.16). In these impellers the vanes are attached on the central unit of the hub without plates. They are used for transporting large amounts of liquid (i.e. in axial flow pumps) without the ability to develop large head height of discharge due to the lack of segregation between suction and discharge. They are also ideal for high speed pumps or pumps that carry fluids with large suspended particles.

The liquid flow to the pump outlet (Ch. 4) is determined by a significant parameter. This is the specific speed (Ns) of the pump, which does not depend

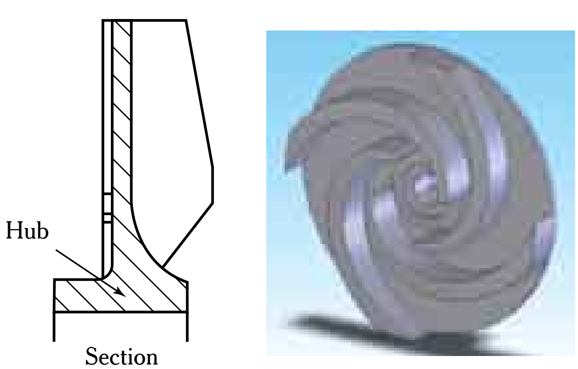


Fig. 3.15
Semi-open impeller.

Fig. 3.16
Open impeller.

on the type, the nature of the pumped liquid or the size of the pump, but on the type of shape and the number of impellers. The specific speed (see par. 5.5) for single-stage pumps is given by:

$$N_s = \frac{n \cdot \sqrt{\dot{V}}}{H^{\frac{3}{4}}} \qquad (3.2)$$

Where: n is the pump speed, V the flow rate to the point of maximum efficiency and H the total discharge head at the point of maximum efficiency. For multi-stage pumps, the relation (3.2) becomes:

$$N_s = \frac{n \cdot \sqrt{\dot{V}}}{\left(\frac{H}{i}\right)^{\frac{3}{4}}} \qquad (3.3)$$

Where: i is the number of pressure stages, because the total head which grows in each impeller is H / i.

The relation (3.2) for pumps with double entry impeller becomes:

$$N_s = \frac{n \cdot \sqrt{\frac{\dot{V}}{2}}}{H^{\frac{3}{4}}} \qquad (3.4)$$

as each side of the impeller provides half of the total flow rate $\dot{V}$.

As the pump performance demand increases, it becomes necessary to change the impeller construction from radial type to axial, since the specific speed depends on the pump design features. In general, it can be concluded that for low specific speeds, low flow rates and high altitude discharge, closed or radial type impellers are used, whereas for high specific speeds, high flow and low altitude discharge, open or axial type impellers are used.

3.6 Classification of dynamic pumps based on the number of stages.

A ***stage*** in a dynamic pump is determined as the combination of an impeller and the flow guiding arrangement of the liquid. By this combination we achieve the total or part of the total discharge head which is developed by the impeller. The guiding arrangement can be a fixed volute casing or an annular casing, with vanes, or more, in combination. So, dynamic pumps with stages are divided into the following categories:

1) ***Single-stage centrifugal pumps*** (fig. 3.17) are the pumps which consist of one impeller within the casing, and this impeller develops the total head. The reached pressure depends on the impeller diameter or the pump speed or on both, so that, for developing greater pressure each parameter must be increased accordingly. But, due to the simultaneous increase in losses, to achieve the desired pressure, it is preferable to use pumps with more stages (multi-stage).

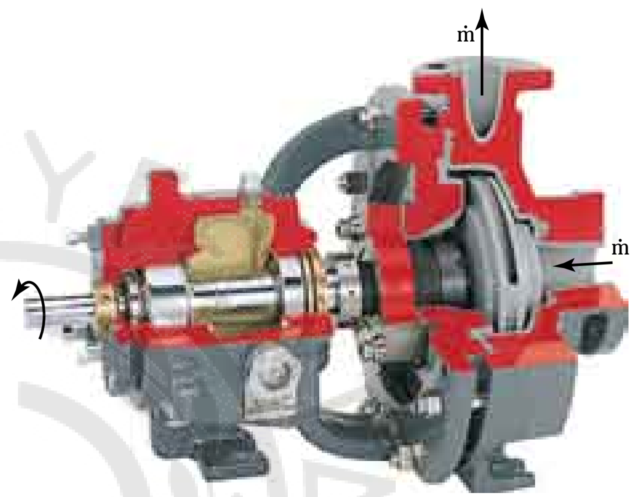


Fig. 3.17
Single-stage centrifugal pump.

2) ***Multi-stage pumps*** (fig. 3.18) are called pumps where the total head develops by two or more impellers in series inside a common appropriately shaped casing. They may be constituted by two or more impellers that are placed on an axis and are alternating with fixed guiding vanes on the casing. Hence, the discharge of one impeller is the suction of the next one, and so on. There is a wide variety of types which derives from the diversity of impellers or casings. Due to the installation of more impel-

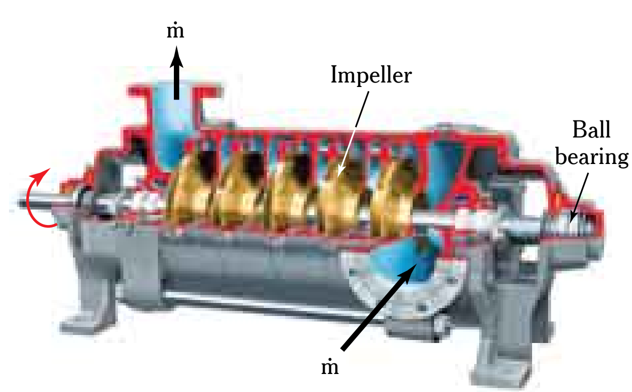


Fig. 3.18
Multi-stage centrifugal pump.

lers in series, the axial thrust in multi-stage pumps is bigger than in single-stage pumps. Therefore, to balance the axial thrusts, double entry impellers are used, or single entry impellers are placed opposite each other in pairs. By this arrangement, opposite impeller thrust is achieved and as a result the axial forces are balanced. ***Multi-stage pumps achieve high flow rates in high discharge pressures and are used as feed pumps to introduce water on steam boilers, overcoming the pressure inside the steam-water drum.***

These pumps are usually manufactured in horizontal arrangement and the casing is divided axially into two half-shells in the shaft plane. The shaft is mounted in ***ball bearings*** (or sometimes in bearings) which are placed at the outer casing. The sealing of the casing is accomplished by gaskets that are placed in a suitable gland on the casing or by ***mechanical seals***, when the pumping liquids are at high pressures and temperatures.

3.7 Special effect dynamic pumps.

Special effect dynamic pumps are classified into nozzle pumps, hydraulic piston, gas lift and electromagnetic pumps.

3.7.1 Nozzle or jet pumps.

Nozzle or ***jet pumps***, which are also known as educators or ejectors or giffard (from the name of the French mechanic Giffard who was the first that created them), are not rotary. Their function is based on the energy which is transmitted from a fluid (liquid or gas) that is named ***operating fluid*** or ***ancillary*** to another fluid, which is called ***pumped fluid***. Essentially, this constitutes the general operating characteristic of pumps with nozzles. Pumping with nozzle pumps – ejectors is accomplished by the creation of vacuum (negative pressure) in a duct, for the liquid suction, with the help of the operating fluid, that is provided to the ejector from a different pipe. The discharge of both fluids happens in a common throat, the diffuser, that is why there are limitations during the use of these pumps in ejector applications. These limitations do not pose a problem, if these pumps are used for the pumping of similar fluids, the characteristics of which coincide (e.g., when the operating fluid and the pumped fluid is water).

According to the above requirements, the main parts that a ejector pump consists of are the ***nozzle***,

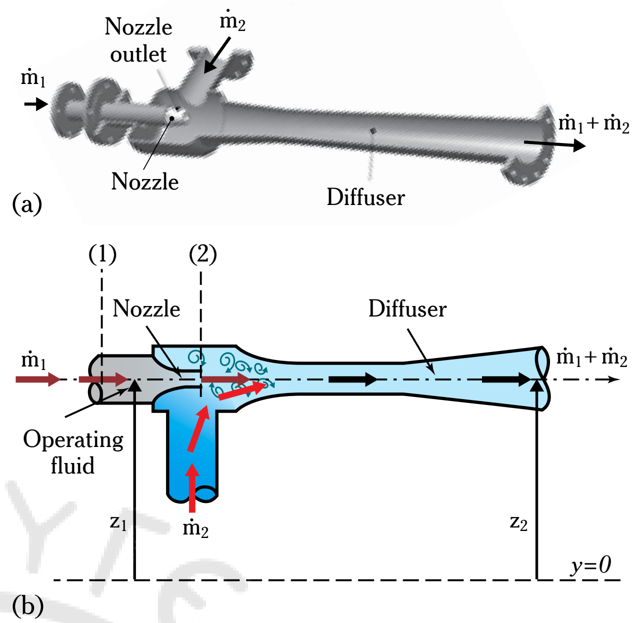


Fig. 3.19
(a) Section of ejector and (b) Operating principle of jet pump.

the ***suction duct*** and the ***diffuser*** (fig. 3.19), whereas the whole system operation is based on Bernoulli's equation and on the function of Venturi's effect in a tube. In order to study the flow through the ejector pumps, the pressure and velocity modification must be considered, which are going to create the vacuum conditions in the ejector interior and so the pumping. Thus, if the diameter in the inflow of the operating fluid in the nozzle is d_1, its velocity is v_1, pressure is p_1 and the corresponding values-dimensions at the nozzle outlet are d_2, v_2 and p_2, it applies that: $p_2<p_1$. More specifically, if the operating fluid is liquid, according to the continuity equation, it applies that:

$$\dot{V} = \frac{\pi \cdot d_1^2}{4} \cdot v_1 = \frac{\pi \cdot d_2^2}{4} \cdot v_2$$

or
$$v_2 = v_1 \cdot \left(\frac{d_1}{d_2}\right)^2 \qquad (3.5)$$

According to the Bernoulli equation:

$$z_1 - z_2 + \frac{p_1 - p_2}{\gamma} + \frac{v_1^2 - v_2^2}{2g} = h_f$$

Because $z_1 = z_2$, it applies:

$$\frac{p_1 - p_2}{\gamma} + \frac{v_1^2 - v_2^2}{2g} = h_f$$

or
$$p_1 - p_2 = \frac{\rho \cdot (v_2^2 - v_1^2)}{2} + \gamma \cdot h_f \qquad (3.6)$$

Where: p is the pressure, v the speed, ρ the liquid density, h_f the head loss and g the gravitational acceleration.

And considering the relation (3.5), the pressure difference becomes:

$$p_1 - p_2 = \frac{\rho \cdot v_1^2}{2} \cdot \left[\left(\frac{d_1}{d_2} \right)^4 - 1 \right] + \gamma \cdot h_f \qquad (3.7)$$

Equation (3.6), and taking into account the amount of height losses h_f, shows the pressure drop[3] that occurs at the outlet of the nozzle. For example, if the operating fluid is water, which enters the nozzle at a speed of 1 m/s and the ratio of the nozzle diameter is ($d_1/d_2 = 4$), even if the losses are considered negligible, from the equation (3.6) it is estimated that the pressure drop is: $p_1 - p_2 = 255$ kPa.

The pressure downswing on the nozzle outlet in the figure 3.19(b) leads to suction of the pumped fluid (m_2) from the suction pipe, with flow rate and velocity that depend on the pressure difference and the dimension of the suction pipe. The pumped fluid is commingled with the operating fluid in the chamber which encircles the nozzle, and they acquire shared velocity, which is reduced, and the fluid pressure increases in the divergent section of the duct (diffuser).

An important advantage of the ejectors is that they do not have rotating or reciprocating components (impeller or piston), but they receive the energy needed for the pumping from the operating fluid and not directly from a drive machine. Steam, compressed air or water under pressure can be used as operating fluid. The disadvantage of this type of pumps, apart from the limitation posed by the necessary compatibility of the fluids in use also include the efficiency reduction, in case that the discharge happens in an environment of pressure.

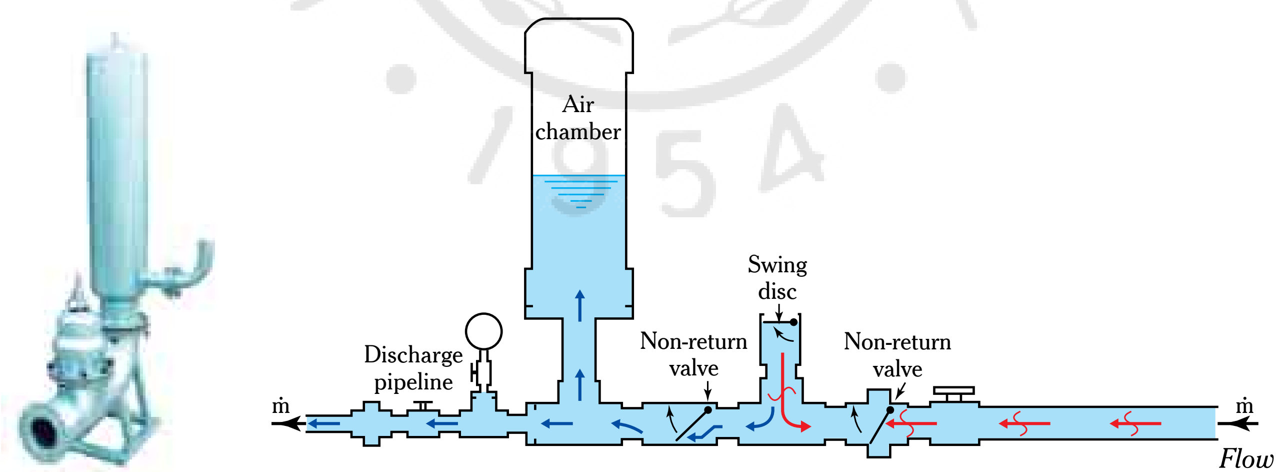
The ejectors are used for pumping fluids, which can also contain gases, due to their capability to suction the air when they are used for creating vacuum, etc.

In ships, we use ejectors (or educators) for pumping bilge, ballast water, wastewater, to create vacuum in the suction pipe of centrifugal pumps, and so on.

3.7.2 Hydraulic ram pumps.

The ***hydraulic ram pump*** (or pressure chamber pump) (fig. 3.20) constitutes an automatic pumping device, that has the capability of pumping liquids that are only higher than the place where the pump is positioned, and this is realized without usage of electric energy or any other energy source. The pump uses a few moving parts and therefore it is a very simple construction of high reliability and minimal maintenance demands for long time periods of operation.

The pump operation is based on the principle of water hammer (hydraulic shock) conditions, resulting in the creation of water pressure. Using this pres-



(a) (b)

Fig. 3.20
Hydraulic ram pump.

[3] If the initial speed height ($v_1^2/2g$) and height losses are considered as negligible, from the equation (3.6) the following relationship is resulted (which is used for an approximate calculation of pressure drop): $p_1 - p_2 = \rho \cdot v_2^2/2$.

sure, the pump can discharge the liquid at a point that is much higher than the pump.

The liquid, water usually, passes through the suction pipe through non return valves during the operation of hydraulic ram pumps (fig. 3.20), whereas a liquid quantity comes out from a swing disc valve (waste valve). The velocity that the water acquires while passing through the swing disc valve forces the disk to close abruptly. When the disc closes, the water has still got energy from the velocity that it has developed. This results in abrupt velocity decrease (Bernoulli Theorem) and creates pressure increase. The increase in pressure, due to the non return valve which ensures that the liquid cannot come back to the suction pipe, forces the water to flow towards the discharge pipe through another non return valve.

An air chamber (air cushion pressure vessel) is also installed in the hydraulic ramp pump system. It contains air and cushions the abrupt fluctuations of pressure with the fluctuation of water level in its interior. The compressed air exerts pressure on the water, forcing it up the discharge pipe, which carries the water up to its destination. As long as the water has got pressure, from the abrupt closing of the waste valve, it flows towards the pressure vessel and the discharge pipe. As soon as the compressed air decompression starts to decrease in the air chamber, the water dominates over its pressure, which comes from the suction side, resulting in the non return valve compulsion to close and a cycle to start anew with the water being inserted from the non return valve of the suction side. This cycle is repeated periodically, resulting in the creation of flow towards the pipeline. ***Pumps of this type are not found in installations on board ships.***

3.7.3 Air or gas lift pumps.

The air or gas lift pump maybe classified as a pump because of its ability to handle liquids. Actually, it is not a machine, but is essentially a pumping method which is used for pumping water or oil from shallow wells, pumping warm groundwater, lifting corrosive liquids or liquids containing solid particles (sand, etc.). The air lift pump operating principle is based on the fundamental Law of Hydrostatics ($P = \rho \cdot g \cdot h$), the principle of communicating vessels [fig. 3.21(a)] and the difference in fluid density. According to these, we observe that the submergence of pipes into the liquid to be pumped would create a type U tube, which operates due to the principle of communicating vessels. The one part of the tube contains only the liquid to be pumped and the other contains a mixture of liquid and air [fig. 3.21(b)]. The same p_{atm} (atmospheric pressure) is exerted on the free surface of the communicating vessels parts. Hence, by the air blown between the communicating vessels, a difference will be created in the contained liquid density. So, in accordance with the fundamental Law of Hydrostatics for each part of communicating vessels tubes, it will apply:

$$p_{hyd1} = \rho_1 \cdot g \cdot h_1, \text{ the part of tube (1) contains only liquid} \quad (3.8)$$

$$p_{hyd2} = \rho_2 \cdot g \cdot h_2, \text{ the part of tube (2) contains a mixture of air and liquid} \quad (3.9)$$

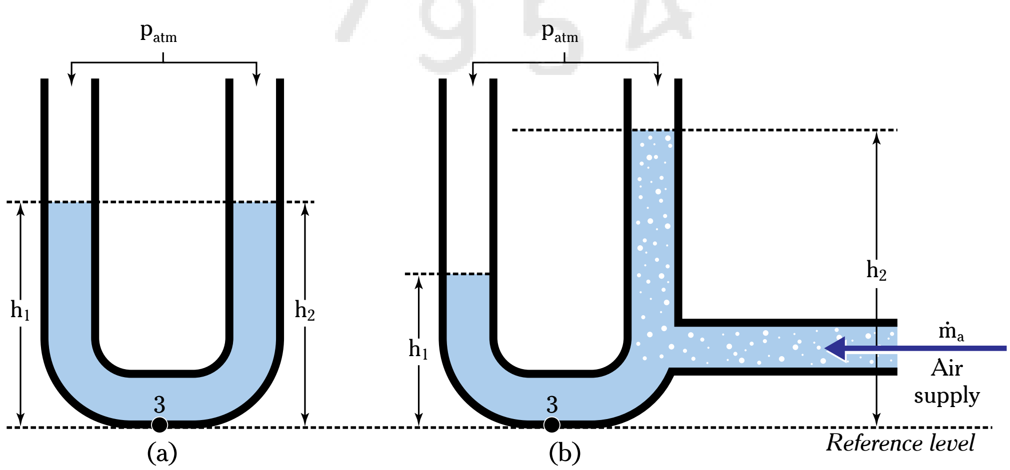


Fig. 3.21
(a) Communicating tube U type, (b) tube U type with air blow in one part.

where: p_{hyd1}, p_{hyd2} is the hydrostatic pressure on the free surface of each part of U tube, at reference point 3, g the gravitational acceleration, h_1 and h_2 the height in each part of the tube, ρ_1 and ρ_2, is the density of pumping liquid and the density of the mixture liquid and air, respectively.

Due to the existence of air, and the Patm on the free surface on the two parts of the U tube, from equations (3.8), (3.9) and for total pressure at the same reference point (3) the following equations apply:

$$p_{atm} + p_{hyd1} = p_{atm} + p_{hyd2}$$

or
$$p_{atm} + \rho_1 \cdot g \cdot h_1 = p_{atm} + \rho_2 \cdot g \cdot h_2$$

or
$$\rho_1 \cdot g \cdot h_1 = \rho_2 \cdot g \cdot h_2$$

or
$$\rho_1 \cdot h_1 = \rho_2 \cdot h_2$$

$$\frac{h_1}{\rho_2} = \frac{h_2}{\rho_1} \text{ or } \frac{h_1}{h_2} = \frac{\rho_2}{\rho_1}$$

or
$$h_2 = h_1 \cdot \frac{\rho_1}{\rho_2} \tag{3.10}$$

Since $\frac{\rho_1}{\rho_2} > 1$, due to the existence of air in one part of the U tube, from equation (3.10) it results that $h_2 > h_1$.

The pump performance depends on the pipe immersion into the liquid that will be pumped, and is proportional to the static pressure at the liquid free surface and the content of air in the liquid.

The air lift pumps are advantageous in terms of simplicity of construction and operation, but are at a disadvantage due to the low degree of efficiency ($\eta = 20\%$ to 35%). The entire installation of the pumping system is shown in figure 3.22, where the water rising pipe is submerged into the tank. When the system operates, an air compressor supplies the pressured air into the tube at the adapted point (A) of the outlet pipe. The mixture of air and water (or other liquids) formed in the riser pipe has lower density than the water surrounding the water pipe. So, in accordance to the principle of communicating vessels, the column of the mixture is raised, and results in the discharge of the riser.

3.7.4 Electromagnetic pumps.

Electromagnetic pumps are used for pumping liquids which are good conductors of electricity. Their principle of operation is identical to that of the electric motors (Faraday's Law). That is, as a conduit is affected by a magnetic or electromagnetic field (fig. 3.23), in electromagnetic pumps the influence of the magnetic field is applied to the conductive fluid that represents the conduit.

For the operation of an electromagnetic pump (fig. 3.24) the following are necessary:

1) The pipeline-tube where the conductive fluid flows through.

2) The magnets for generating the magnetic field, and

3) The power source which provides power to a conductor connected to the tube of the conducting fluid.

The magnetic field $\boldsymbol{B_{ef}}$, produced by permanent

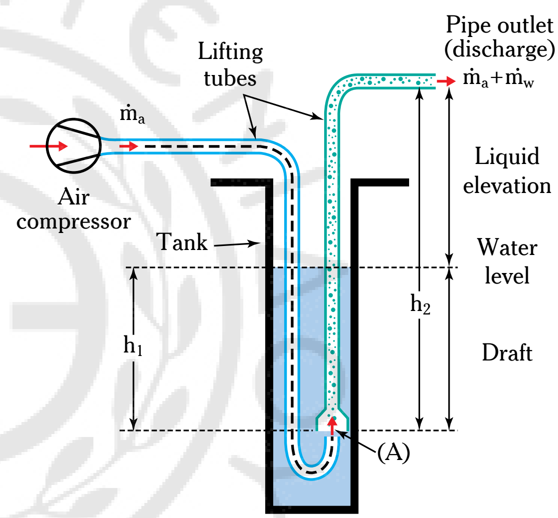
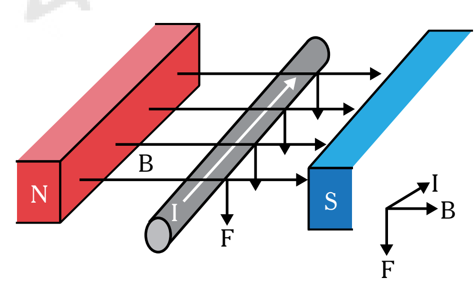


Fig. 3.22
Air or gas lift pump.



B, Magnetic field
I, Current flowing into conductor
F, Laplace force on the conductor

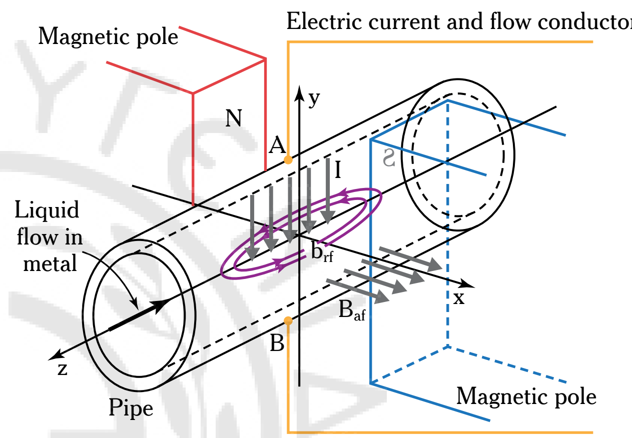
Fig. 3.23
Laplace force on a conductor in a magnetic field.

magnets which are mounted in a suitable pattern, passes perpendicularly by the conduit-pipe where the conductive fluid flows. Also, direct current is applied at points A and B from an external source (e.g. battery). The supplied current at points A and B passes from the conductive fluid, too, creating a magnetic field by induction. It has current density (I) and the magnetic field associated with this current can be called "Reaction magnetic Field" b_{rf}. As the two magnetic fields B_{af} and b_{rf} try to align with each other, this causes the mechanical motion of the fluid in the tube.

Hence, the magnetic field, which can be generated in several ways (e.g. battery or power source to electromagnets), acts on the fluid causing it to flow into the tube.

Electromagnetic pumps are used for pumping liquid metal, for cooling nuclear reactors and for casting by moving melted metal that flows through pipes of suitable material, e.g. ceramic.

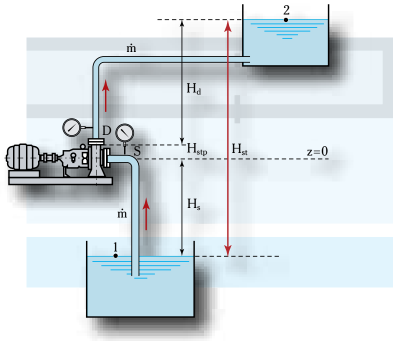
(a)



(b)

B_{af}, Applied magnetic field
I, Current flow between the electrodes
b_{rf}, Reaction magnetic field because of I

Fig. 3.24
(a) Electromagnetic pumps and
(b) schematic representation of the electromagnetic pump operation.



CHAPTER FOUR
Characteristic parameters of pumping systems

4.1 Basic principles of fluid mechanics and hydraulic systems.

Fluid is a substance in which the constituent molecules are free to move relative to each other. Hence, it exhibits free sliding flow in order to take the shape of the space occupied or the shape of the means wherein it moves without any fragmentation of the mass being observed when external forces are exerted on it. If the external forces exerted on the fluid are non-destructive (e.g., non-destructive conditions of temperature and pressure), the relative positions of molecules remain essentially fixed and it reacts by sliding continuously or by being displaced entirely, independently and constantly. Fluids are characterized by the following properties:

1) The ***ability to be compressed*** (compressibility).

2) The ***viscosity***, which indicates the presence of resistance to flow[1].

3) The ***affinity***, which is the traction force which develops between adjacent molecules (intermolecular) of different substances or materials, such as the fluid coherence with the walls of the vessel containing it.

Based on the above, the fluids, which are liquid or gaseous, are divided into:

1) ***Ideal*** or ***perfect fluids*** that are completely uncompressed, while no internal friction develops between the molecules during the flow. It is obvious that this is a theoretical state and is reflected in surveys and studies of the Fluids Laws, enabling conclusions with respective calculations.

2) ***Natural fluids*** which have properties that differ from the properties of perfect fluids, such as the ability to be compressed, the development of internal friction, and affinity.

The forces acting upon fluids are divided into:

1) ***External***, i.e. gravity, centrifugal force and atmospheric pressure, and

2) ***Internal***, i.e. pressures such as hydrostatic pressure and static head.

From the forces acting upon fluids, during their movement, be that internal or external, certain conditions, laws and equations have been raised, which specify how a pump or a pumping system operates. These are:

1) ***Atmospheric pressure***, which is the pressure that is exerted by the atmosphere's weight on the sea surface. According to Torricelli's theorem, it is equal to 760 mm Hg (millimeters of mercury).

2) ***Hydrostatic pressure***, which is the pressure exerted by a liquid to an object or a surface within the liquid (fig. 4.1).

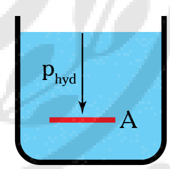


Fig. 4.1
The hydrostatic pressure exerted on a submerged surface.

The pressure is due to the external force of gravity, i.e. the weight of the liquid found above the surface or object. Thus, if the hydrostatic pressure is described as p_{hyd}, the force representing the weight of the liquid above the surface as F, and A is the surface on which the p pressure is exerted, the hydrostatic pressure equals the quotient:

$$p_{hyd} = \frac{F}{A} \tag{4.1}$$

3) ***Hydrostatic*** or ***pressure height*** (h) of a point located in a liquid is the vertical distance of this point from the free surface of the liquid mass. Thus, if ρ is the density of the liquid, then the force

[1] This resistance is caused by internal friction of the molecules from cohesive forces, to the extent that viscosity itself is a measure of liquid resistance to flow or a measure of liquidity.

F exerted on the surface A is given by:

$$F = V \cdot \rho \cdot g = A \cdot h \cdot \rho \cdot g \qquad (4.1a)$$

Then due to $p_{hyd} = \dfrac{F}{A}$ the (eq. 4.1a) becomes

$$p_{hyd} = \frac{A \cdot h \cdot \rho \cdot g}{A} \quad \text{or} \quad p_{hyd} = \rho \cdot g \cdot h = \gamma \cdot h \qquad (4.2)$$

thus $$h = \frac{p_{hyd}}{\rho \cdot g} \quad \text{or} \quad h = \frac{p_{hyd}}{\gamma} \qquad (4.3)$$

Therefore, the static height is the measure of the hydrostatic pressure at the point inside the liquid, and changes according to the change of pressure in proportion to the change in depth.

4) ***Archimedes' principle***. According to this, each body immersed in a fluid receives buoyancy equal to the weight of the fluid that is displaced, so if B is the buoyancy, it is expressed by the formula:

$$B = \rho \cdot g \cdot V \qquad (4.4)$$

wherein: ρ is the density of the fluid, g the acceleration of gravity, and V is the volume of the submerged portion of the object (volume of fluid displaced).

Thus, if V_l is the volume of the body and ρ_l its density, the weight G of the object is given by:

$$G = V_l \cdot \rho_l \cdot g \qquad (4.5)$$

5) ***Pascal's principle*** [Blaise Pascal, (1623 – 1662) French physicomathematician] is one of the three basic principles of hydrostatics and determines that any pressure applied to the surface of a liquid is evenly transferred to it in all directions and in all its depth. The application of this principle is found in pumps and hydraulic pressure machines.

6) The ***principle of communicating vessels*** (fig. 4.2). When two or more vessels of any shape contain homogeneous liquid and communicate to each other, the level of the free liquid surfaces in all the vessels is at the same height.

7) The ***fundamental equation of hydrostatic pressure*** (fig. 4.3) consists of the combination of hydrostatic pressure and Pascal's law. According to that, at a point in a liquid where on its surface an external pressure p_{ex} is applied (by force F), since the force of gravity is exercised as well, the total pressure p_T at this point is equal to the sum of external pressure p_{ex} and the relevant hydrostatic pressure p_{hyd}.

Where $p_T = p_{ex} + p_{hyd}$ since

$$p_{hyd} = h \cdot \rho_{hyd} \cdot g \quad \text{then} \quad p_T = p_{ex} + (h \cdot \rho_{hyd} \cdot g) \qquad (4.6)$$

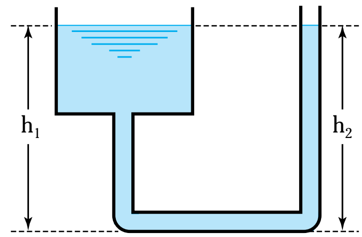


Fig. 4.2
Principle of communicating vessels.

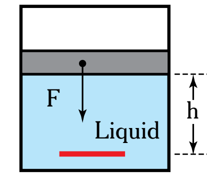


Fig. 4.3
Illustration of Hydrostatic pressure.

8) The ***law*** or ***equation of flow continuity*** (fig. 4.4). This law (because of the law of the conservation of mass) states that the mass flow rate through pipes, tubes and ducts on a steady flow process is constant along any section wherein the liquid flows. Hence, using the mass conservation law, as well as fluid mechanics, it is known that:

$$\dot{m}_1 = \dot{m}_2 \quad \text{or} \quad A_1 \cdot v_1 = A_2 \cdot v_2 \qquad (4.7)$$

where A is area of pipe cross-section and v is the fluid velocity.

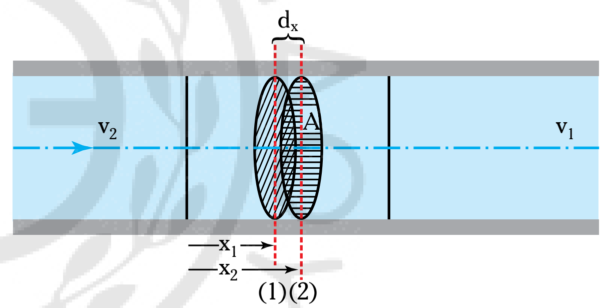


Fig. 4.4
Illustration of flow continuity equation.

Thus, if we have a pipe with varying cross-sections at different points of its length, either due to its shape or because of salts deposits, the constant flow rate $\dot{V}$ at the different points as the liquid is flowing in one direction inside the tube will be equal to:

$$\dot{V}_1 = A_1 \cdot v_1, \qquad \dot{V}_2 = A_2 \cdot v_2, \qquad \dot{V}_3 = A_3 \cdot v_3,$$

wherein: A_1, A_2, A_3 ...., A_n the area of various sections and v_1, v_2, v_3 ...., v are the respective velocities of the liquid therein.

Then we will have:

$$A_1 \cdot v_1 = A_2 \cdot v_2 = A_3 \cdot v_3 = \ldots\ldots\ldots = c = (\text{constant})$$

or generally $$A \cdot v = \text{constant} \qquad (4.8)$$

From this relationship it is determined that when the pipe intersection is increased, the liquid velocity decreases and vice versa, so that the product when we multiply the cross section by the velocity is always the same and equal to the constant mass flow rate of the pipe. This equation is a direct consequence of the mass conservation law, and the equation is applicable to the construction of piping systems (networks) and pipe branches, pipe contractions or expansions, etc.

$$\dot{m}_1 = \dot{m}_2 = \ldots\ldots = \dot{m}_n = c \tag{4.8a}$$

$$\rho_1 \dot{V}_1 = \rho_2 \dot{V}_2 = \ldots\ldots = \rho_n \dot{V}_n = c \tag{4.8b}$$

for incompressible fluids:

$$\rho_1 = \rho_2 = \ldots\ldots\ldots = \rho_n$$

or $$\dot{V}_1 = \dot{V}_2 = \ldots \dot{V}_n$$

or $$A_1 v_1 = A_2 v_2 = \ldots = A_n v_n = c$$

9) The ***volume flow through the pipe where the liquid is flowing***, is given by:

$$\dot{V} = A \cdot v \tag{4.9}$$

10) The ***energy equation***, which is commonly used in fluid mechanics for steady, incompressible flow, along a steamline in inviscid regions offlow, also known as the ***Bernoulli equation***, is another major tool that is used to analyse a hydrodynamic system. Daniel Bernoulli (1700–1782), Swiss physicist and mathematician, made important discoveries in fluid dynamics. Sometimes this and the continuity equation are needed to solve a particular problem. According to the Bernoulli equation, applied in two points on the centreline of a pipe, 1 and 2, and making the assumption that there is no loss of energy (the total energy at points 1 and 2 is the same), when the speed of a fluid increases, the pressure decreases, and vice versa. Hence, even if the energy may change from one form to another, the following equation is true:

$$\frac{p_1}{\rho_1} + \frac{v_1^2}{2} + g \cdot z_1 = \frac{p_2}{\rho_2} + \frac{v_2^2}{2} + g \cdot z_2 =$$

$$= \ldots\ldots\ldots = \frac{p_n}{\rho_n} + \frac{v_n^2}{2} + g \cdot z_n = c \;\; \text{(constant)} \tag{4.10}$$

4.2 Flow of real fluids.

The difference in flow of real fluids from the ideal fluids is that real fluids are influenced by their compressibility, cohesion of molecules (viscosity in nature) and the affinity. Note that the compressibility hardly affects the flow, because liquids are considered practically incompressible, unlike gases where the effect is great.

The coherence of the real liquid molecules results in the creation of internal friction. The affinity relates to the contact force between the liquid with the walls of the duct. It occurs in such a way that the liquid molecules that are in contact with the walls of a tube (boundary layer or lamina of liquid) have zero speed (the lamina is essentially stationary), while the liquid molecules that move at the center of the duct show the maximum speed. Thus, the flow of physical fluids is characterized as:

1) ***Laminar flow***, when in the cross section of the pipeline with section A, the distribution of liquid molecules is orderly with all particles moving in straight lines parallel to the pipe walls, and has the form of figure 4.5.

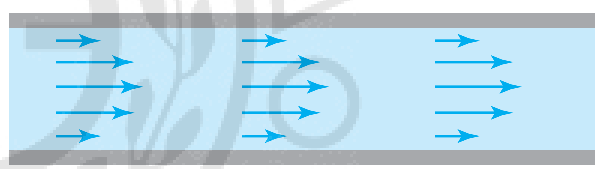


Fig. 4.5
Laminar flow.

2) ***Turbulent flow***, when in the cross section of the pipeline with section A, the distribution of molecules and flow lines of the liquid has the form of figure 4.6.

By observing the two streamline[2] forms illustrated, we note that the velocity of fluid molecules which are in contact with the pipe wall is zero, while the maximum speed is shown on the liquid molecules

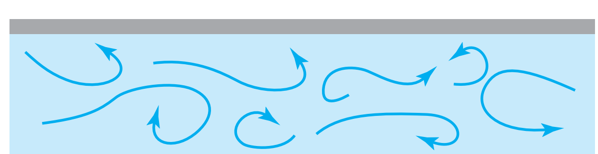
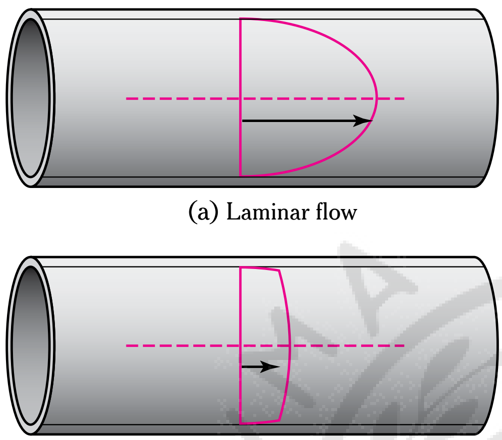


Fig. 4.6
Turbulent flow.

[2] ***Streamline*** is the path of any particle of liquid as it moves through the pipe.

at the center of the tube. ***The speed of movement of all particles moving in the full extent of the cross section A within the tube is calculated as an average speed represented by v_a.***

This speed is not the same in the laminar flow, but acquires the form shown in figure 4.7(a), while it is distributed differently in the conduit where the flow is turbulent [fig. 4.7(b)].



(a) Laminar flow

(b) Turbulent flow

Fig. 4.7
Distribution of fluid velocity.

4.2.1 Tube volumetric flow rate.

The simplest way to calculate the volumetric flow rate (or volume flow rate, rate of fluid flow) is by following the general formula providing and using the average velocity v_a, namely:

$$\dot{V} = A \cdot v_a \tag{4.11}$$

where: A is the section area and v_a the mean flow velocity, applies $v_a = 1/2\ v_{max}$, where v_{max} is the maximum speed.

The parallel liquid flow is converted to turbulent flow in the tube when, under Reynolds (Osborne Reynolds, physicomathematician, 1842–1912), the fluid flow rate exceeds a certain value called the ***critical flow speed*** (v_c), and is determined by the ***Reynolds number*** (Re).

Reynolds number is a dimensionless quantity which is determined experimentally for each case. It characterizes the flow under pressure and is used to help predict flow patterns in different fluid flow situations.

$$Re = \frac{\dot{V} \cdot D_h}{A \cdot \nu} \tag{4.12}$$

or

$$Re = \frac{v_a \cdot D_h}{\nu} \tag{4.13}$$

where: ν is the kinematic viscosity of the fluid (m^2/s), $\dot{V}$ the volume flow rate (m^3/s), A the cross section area (m^2), Dh the hydraulic diameter (m), and v_a the average velocity.

4.2.2 Losses due to resistance to flow.

Part of the energy provided by the pump to a fluid flowing inside a pipe is consumed in losses due to flow resistance. The total energy losses depend on the cross section, the internal condition of pipe surfaces and fittings connected to the piping system, the speed, fluid viscosity, friction and turbulences caused with changes of flow direction, the installed ***valves***, and other factors experimentally estimated.

The energy losses are caused by:

1) Resistances exhibited by pipes to the liquid flow due to frictions by the contact of liquid with the pipe surface, and the flow type that develops. The magnitude of the losses depends on the liquid flow velocity, the temperature and the condition of inner surface of the pipe (roughness). Resistance also depends on the viscosity of the flowing liquid and the texture of the inner surface of the pipe so that the higher the viscosity or surface roughness, the greater the resistance to liquid flow.

2) Resistances due to abrupt fluctuation of the pipe section and the existence of various joints on the connections, where, due to the interference of transverse wall and the differentiation of the inner surface, turbulences are created that have the effect of slowing the flow of liquid.

3) Resistances due to change of the pipe direction, which depends on the diameter of the tube, the radius, the fluid density and viscosity.

4) Resistance due to interference of various components and devices, such as valves, thermometers, gauges, etc. These create diversions of flow and turbulences that cause slowing down proportional to the fluid motion.

4.2.3 The flow of liquids through a siphon.

A siphon is a bent tube used for transferring a liquid from one vessel to another one at a lower level. Accordingly, to transfer liquid through the siphon, the bent tube is filled with liquid so that streamline flow can take place, as each of the two ends is placed in each vessel.

The difference in weight of the fluid column in each arm of the tube (siphon) connecting the two containers creates flow streamline. The flow is maintained due to the cohesion of the liquid molecules, the difference in weight of the fluid, the pull of gravity and the pressure at the free surface of liquids in both vessels. For determining the flow through the siphon, we assume that a U-tube is inverted and submerged in a container of liquid, while the same tube is filled with liquid.

By pulling the tube out of liquid (fig. 4.8), we observe that the fluid does not flow in the U-tube, because there is atmospheric pressure (p_{atm}) on the surface of the liquid while in the upper part inside the tube the pressure p is smaller than p_{atm}, because:

$$\left.\begin{array}{l} p_2 = p_1 + \rho \cdot g \cdot h \\ p_2 = p_{atm} \end{array}\right\} \Rightarrow \begin{array}{l} p_{atm} = p_1 + \rho \cdot g \cdot h \\ \text{or} \;\; p_1 = p_{atm} - \rho \cdot g \cdot h \\ \text{and} \;\; p_{atm} > p_1 \end{array} \tag{4.14}$$

where ρ is the density of the liquid, g is the acceleration of gravity, and h is the height at which the pipe is out of the liquid.

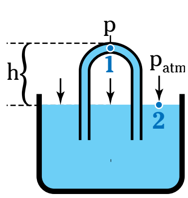


Fig. 4.8
Liquid flow in inverted U-tube in a vessel with liquid.

Thus, the atmospheric pressure forces the liquid to the interior of the inverted U-tube.

If the U-tube connects two vessels with liquid in different heights (fig. 4.9), then inside the tube at the upper point (1) the pressure will be p_1 and at the lower point (2) it will be p_2, for which applies:

$$p_{atm} = p_1 + \rho \cdot g \cdot h_1 \Rightarrow p_1 = p_{atm} - \rho \cdot g \cdot h_1 \tag{4.15a}$$

$$p_{atm} = p_2 + \rho \cdot g \cdot h_2 \Rightarrow p_2 = p_{atm} - \rho \cdot g \cdot h_2 \tag{4.15b}$$

According to equations (4.15a) and (4.15b), applies:

$$p_1 + \rho \cdot g \cdot h_1 = p_2 + \rho \cdot g \cdot h_2 \Rightarrow p_1 - p_2 = \rho \cdot g \cdot h_2 - \rho \cdot g \cdot h_1$$

or $\qquad p_1 - p_2 > 0 \quad \text{because} \quad h_2 > h_1$

or $\qquad p_1 > p_2$

This difference is caused by the weight difference of the fluid in each leg of the U-tube, as the height of column h_1 is less than height h_2, therefore the weight of liquid that exists inside the pipe of height h_2 is greater than the weight of the liquid contained in the pipe of h_1. So, because of the cohesion of the molecules, the liquid is entrained from vessel A to vessel B (fig. 4.9).

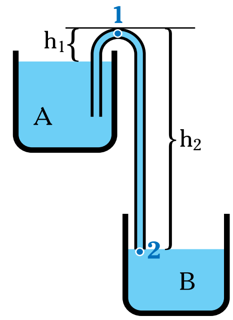


Fig. 4.9
Flow of liquid through siphon with P_{atm} on free surfaces of vessels.

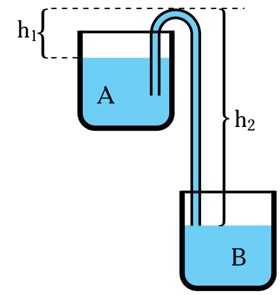


Fig. 4.10
Flow through siphon between closed vessels.

For the operation of the siphon, the existence of atmospheric pressure is not required, as can be shown in the closed circuit of figure 4.10. There, by providing the necessary tilt to the system in order to transfer an amount of the liquid contents of the two containers through the siphon to vessel A and then tilting the system to the opposite side, i.e. towards vessel B, initial flow is generated, which continues from vessel A to B when the system returns to vertical position.

4.3 Conservation of mass and energy in a pumping system.

In order to understand the function, role, and potential problems in the use of pumps, and also to formulate the criteria for their selection in relation to the requirements of the pumping system, they are installed in, it is necessary to investigate their operating environment. In any pumping system, the conservation of mass principle is applied and the Bernoulli energy equation for incompressible fluids.

A common pumping system is shown in figure 4.11(a), which consists of three modules:

1) The suction pipe 1-S.
2) The discharge pipe D-2.
3) The pump, which is interposed between the two pipes.

The pump draws liquid from the state 1, where its energy is characterized by an altitude z_1, pressure of fluid p_1 and speed v_1, and discharges in the state 2 to the respective qualities z_2, p_2, v_2.

It should be noted that the suction and discharge are not necessarily the points 1 and 2 but, when studying a system, different points may be considered. If there are suction and discharge tanks [fig. 4.11a(b)] the points 1 and 2 correspond to the free surfaces of liquid inside the tanks where velocity is zero (i.e. $v_1 = v_2 = 0$), while the velocities within the tubes are defined as v_s and v_d and are different from zero.

In case of a closed pumping system (e.g., recirculation in a ship's cooling system with fresh water), points 1 and 2 are identical.

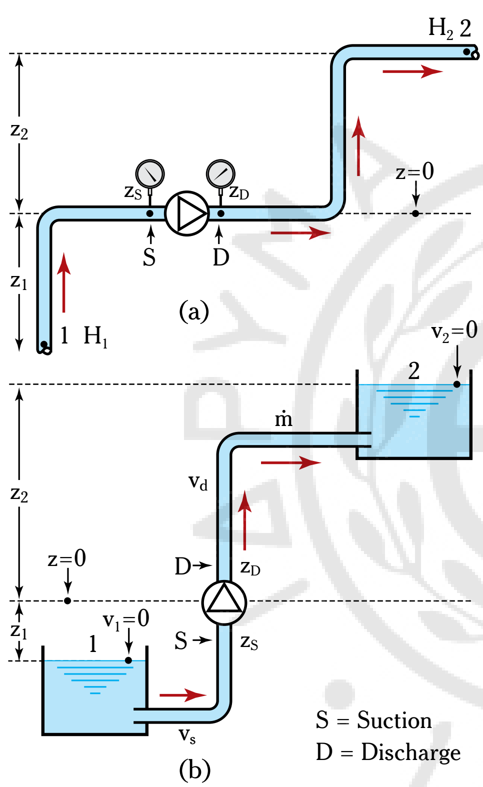


Fig. 4.11
Pumping system.

4.3.1 Continuity equation.

According to the continuity equation, in a pumping system with permanent operation, the flow remains steady.

Permanent operation means that there is no change of any parameter in the pipes. For example, a change such as a discharge valve opening, modifies the local loss factor and therefore alters the flow. The same will happen if the pump operating conditions change, e.g., if there is increase of speed in a centrifugal pump.

We should clarify what happens with the supply system in the case of reciprocating single-acting pumps. In these, the pump sucks in one stroke and discharges in the next. So the flow is not steady. The use of air accumulators reduces the problem but does not eliminate it. A more balanced condition occurs in double acting reciprocating pumps, but the problem of small fluctuations in the instantaneous values of flow still exists. Note that the instantaneous flow in reciprocating pumps periodically varies around a mean value, which is stable and constitutes the flow of the pumping system. Similar problems do not occur in other types of pumps, where the flow is stable.

Also, according to the continuity equation, in any section of the piping system perpendicular to the flow, the following formula applies:

$$\dot{V} = A \cdot v = \text{const.} \qquad (4.16)$$

For suction and discharge pipes with size $A_s = \pi \cdot d_s^{\,2}/4$, $A_d = \pi \cdot d_d^{\,2}/4$ respectively equation (4.16) becomes:

$$V = A \cdot v = \frac{\pi \cdot d_s^2}{4} \cdot v_s = \frac{\pi \cdot d_d^2}{4} \cdot v_d \qquad (4.16a)$$

$$d_s^{\,2} \cdot v_s \le d_d^2 \cdot v_d \qquad (4.16b)$$

where: v_s and v_d are the fluid velocities on the suction and discharge pipes, respectively.

Commonly, the discharge pipe has a slightly smaller diameter than the suction pipe. Consequently, the flow velocity is slightly higher in the discharge pipe so that:

$$v_d = v_s \frac{d_s^2}{d_d^2}, \quad \frac{d_s}{d_s} > 1$$

so $$v_d > v_s \ \text{for} \ \dot{V} = c \qquad (4.16c)$$

In the interior of the pump, the vertical cross-section to the flow is not circular (such as in the tube), but depends on the type and design of the pump. Currently, for pumping, the general relation (4.16) is applied. The specific application of the continuity equation within the centrifugal pumps will be discussed in Chapter 5.

4.3.2 Energy equation.

The Bernoulli equation for incompressible fluids is a statement derived from the conservation of energy.

For the energy study of the pumping system in figure 4.11, the Bernoulli equation is applied[3] between points 1 and 2.

Therefore, the total energy head (H) at any point in the system relative to a selected datum plane is equal to the sum of the elevation height or load of position (z), the pressure head $[p/(\rho \cdot g)]$ and velocity head or load of speed $(v^2/2g)$, so:

$$H_1 = \frac{p_1}{\rho_1 \cdot g} + \frac{v_1^2}{2 \cdot g} + z_1 \qquad (4.17a)$$

$$H_2 = \frac{p_2}{\rho_2 \cdot g} + \frac{v_2^2}{2 \cdot g} + z_2 \qquad (4.17b)$$

For water it applies $\rho_1 = \rho_2$

During the flow of liquid in suction pipes (1-S) and discharge (D-2) at a flow rate $\dot{V}$, the height of energy loss Σh_f is given as:

$$\Sigma h_f = h_{fs} + h_{fd} \qquad (4.17c)$$

As fluid passes through the pump, where the energy is added only by the pump and the specific weight (force) of the liquid does not change (for example, as a result of temperature), it receives energy head ***H*** which is called ***given*** or ***total head*** of the pump.

The energy balance is:

$$H_1 + H = H_2 + \Sigma h_f \Rightarrow$$
$$\Rightarrow H = (H_2 - H_1) + \Sigma h_f \qquad (4.17d)$$

Substituting the H_1 and H_2, the following general equation for determining pump total head applies:

$$H = (z_2 - z_1) + \frac{p_2 - p_1}{\rho \cdot g} + \frac{v_2^2 - v_1^2}{2g} + \Sigma h_f \qquad (4.17e)$$

Equivalent static head or ***static head of the pumping system*** is called the difference in liquid levels $z_2 - z_1$. Similarly, ***suction static head H_{sts}*** is called the altitude difference $z_S - z_1$ (in order to have flow in the suction tube) and ***static discharge head H_{std}***, the height difference $z_2 - z_D$ (in order to have flow in the discharge pipe) [fig. 4.11(a)]. So the static head of a pumping system is:

$$H_{st} = z_2 - z_1 \quad \text{or} \quad H_{st} = H_s + H_d + H_{\Sigma p} \qquad (4.17f)$$

where: $H_{stp} = z_D - z_S$ is the static head of the pump, which is usually negligible.

Substituting in equation (4.17e) the total head of the pump, it becomes:

$$H = H_{st} + \frac{p_2 - p_1}{\rho \cdot g} + \frac{v_2^2 - v_1^2}{2 \cdot g} + \Sigma h_f \qquad (4.17)$$

Equation (4.17) shows the energy forms which the mechanical work that the pump yields to the liquid that flows to the pumping system is converted into, when:

1) It increases the dynamic energy and is given as static head or potential energy:

$$H_{st} = z_2 - z_1$$

2) It increases the pressure energy and is given as a change of pressure head:

$$\frac{p_2 - p_1}{\rho \cdot g}$$

3) It increases the kinetic energy and is given as change in velocity head:

$$\frac{v_2^2 - v_1^2}{2 \cdot g}$$

4) Covering the friction losses and is given as sum of piping losses or frictional head losses: $\Sigma h_f = H_{f1S} + H_{fSD} + H_{fD2}$ [fig. 4.11(a)].

Of the four energy terms at the second part of the equation (4.17), the frictional head losses Σh_f always have positive value, while the remaining terms might be zero or may have a negative value. More specifically:

1) ***The friction losses exist and are unavoidable in all pumping systems.*** Particularly so in closed pumping systems, since the points 1 and 2 coincide.

2) ***The increase of potential energy*** is the major energy objective of pumping systems used for elevating liquids (e.g., water pumping from drilling, from the sea, etc.). In the quite common case of liquid elevation from one tank to another (fig. 4.12), if on the surface of both tanks atmospheric pressure prevails (where $p_1 = p_2$ and $v_1 = v_2$), the equation (4.17) becomes:

$$H = H_{st} + \Sigma h_f \qquad (4.17g)$$

In the case that point 1 is higher than point 2, the static head is negative (and reduces the pump load).

3) ***The increase of pressure energy*** is the

[3] It is emphasized that the energy heights are expressed in terms of height of column of the liquid.

most important energy change in case the pumping system transports liquid in an area with high pressure. Thus, the head yielded of the steam boiler feed pump is converted primarily into pressure head. If the head difference and the speed alteration are negligible, then:

$$H = \frac{p_2 - p_1}{\rho \cdot g} + \Sigma h_f \qquad (4.17h)$$

The alteration of the pressure energy in systems in which the discharge tank is a vessel under pressure is also significant.

4) ***The alteration of kinetic energy*** is in most cases relatively small. Specifically in a pumping system:

a) it is equal to zero when the pumping system has suction and discharge tanks (fig. 4.12), since on the free surfaces of liquid inside the tanks the speed is practically zero.

b) it is negligible or zero when the system we are studying begins and finishes in tubes of similar or equal diameter (in tubes with equal diameter in a system with constant flow rate equal speed prevails, as is clear from the continuity equation).

c) it is small (but appreciable) when the pumping system begins at a suction tank ($v_1 = 0$) and ends with a tube ($v_2 \neq 0$).

The only circumstance in which the amount of kinetic energy is a key factor is when the pumping system aims to increase the liquid speed. This happens in systems designed to extinguish fires and is obtained by placing a converging nozzle at the system outlet wherein the diameter d_2 decreases, and hence the v_2 speed increases.

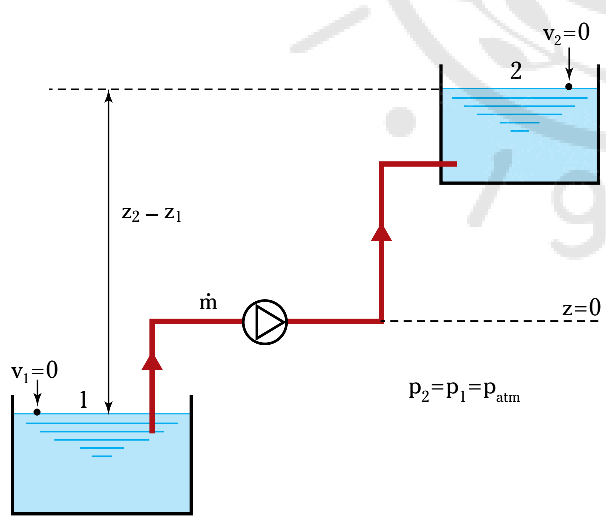


Fig. 4.12
Pumping system.

The flow rate of the pumping system directly affects the friction losses (which, according to the Darcy-Weisbach[4] equation, is proportional to the square of the flow rate), and the value of velocity in the pumping system. The modification of the equation for calculating the total head of the pump [eq. (4.17)], by replacing the amount of loss and the kinetic energy as a function of flow rate, gives the equation:

$$H = a + b \cdot \dot{V}^2 \qquad (4.18)$$

where $$a = H_{st} + \frac{p_2 - p_1}{\rho \cdot g}$$

is a coefficient equal to the sum of static head and the amount of pressure, and (b) is the coefficient which depends on the data of the suction and discharge pipes. For the calculations, the variations of the velocity of the liquid and the losses in the suction and the discharge are taken into account.

Equation (4.18) indicates that the assigned amount that the pumping system requires by the pump depends on the square of the volume flow rate.

In practice, much more important than the energy per unit weight is the work per unit time that the pump yields to the fluid, i.e. ***the power delivered*** to all liquid through the impeller P_I.

The total head H that is transferred by a pump to the fluid is calculated from equation (4.17). If through the pump passes amount of liquid $G = m \cdot g$, the delivered work by the pump is: $W = H \cdot m \cdot g$. Therefore, the delivered power (work per unit time) is expressed as:

$$P_I = \frac{W}{t} = \frac{H \cdot m \cdot g}{t} = H \cdot m \cdot g$$

The mass flow rate m is related to the volume flow rate through the density:

$$\rho = \frac{m}{V} \Rightarrow m = \rho \cdot V \quad \text{or} \quad m = \rho \cdot V$$

[4] In fluid dynamics, the Darcy–Weisbach equation (from the names of Henry Darcy and Julius Weisbach) relates to the loss of the energy amount of the fluid, or the pressure losses, due to friction along a conduit of a given length, with the mean velocity of fluid flow.

Substituting, and whereas $\rho \cdot g = \gamma$, results in the relationship that is used to calculate the required energy given by a pump to a quantity of fluid against a given total head, known as hydraulic power, as follows:

$$P_I = \gamma \cdot \dot{V} \cdot H \qquad (4.19)$$

wherein: γ is the fluid weight density, $\dot{V}$ the volume flow rate of the piping system, and H the total head of the pump.

4.4 Water hammer.

The water hammer (or ***hydraulic*** or ***pressure shock***), is a non-permanent phenomenon that occurs in hydraulic systems. The pressure shock is really a pressure wave with a velocity of propagation much higher than the velocity of the flow. It is frequently the result of rapid changes in the velocity of the fluid or strong variations of the pressure, especially in long runs of piping, that are usually much greater than the pressure developed during permanent flow of fluid therein.

The water hammer is caused when the continuous flow of liquid stops abruptly by closing a valve at the pipeline system. Then, the flow velocity is reduced to zero, which results in the conversion of the kinetic energy of the fluid to dynamic energy (dynamic pressure) and since its pressure suddenly increases, it causes a strong shock in the conduit.

That impact is called ***water hammer*** with significant stresses in the conduit, because, for a short time, strong alternating tension and compression are developed, which can cause fracture of the pipeline or other serious damage to the system.

The water hammer is observed in piston pumps, since their operation is not continuous but alternated, and occurs upon stopping the flow of liquid through the pipes or in the suction and depression valves. Prevention of the phenomenon is achieved:

1) by smooth closing of the valves in the pipelines, and

2) by adding equipment available to reduce pressure waves in pipelines of suction or discharge or in both, such as pressure storage tanks like air accumulators, pressure towers, damped or undamped non-return valves, when piston pumps are used.

In the air accumulators, depending on the pressure resulting from the liquid motion, as the air is compressed or expanded, it acts as a spring absorbing the effect of the steep pressure and velocity changes, making the flow smooth.

4.5 Characteristic parameters of pumps and pumping systems.

The pump, as mentioned, is the heart of the pumping systems, since it treats the energy needs and transmits to the liquid the mechanical work which is indispensable for its movement. The calculation of the system introduces several quantities of energy and flow that are related to the pump and characterize its effect or its capability. Therefore, the following four groups of characteristic parameters of the pumping system (and therefore of the pump) are distinguished: the equivalent ***energy heads*** (equivalent of velocity and pressure energy heads), ***volume flow rate***, ***power*** and ***efficiency***.

4.5.1 Equivalent energy heads.

Head is the height above the suction inlet that a pump can lift a fluid. The equivalent of velocity and pressure energy heads can be thought of as the height to which a vessel of liquid of constant density has to be filled, above the point of measurement, to create this same velocity or pressure. These are further explained in the following text.

1) Energy and head in pumping systems.

The pump has to face the energy heads of the pumping system wherein it is placed (fig. 4.13). According to the energy equation (4.17) these energy terms are the following:

1) The ***static head of the pumping system (*H_{st}*)***, expressing the change of dynamic energy from the beginning to the end of the pumping system. It is equal to the elevation difference between beginning and end of the system and so it is also called ***geometric head***. It consists of ***a suction static head H_{sts}*** ($= z_s$), ***a discharge static head H_{std}*** ($= z_d$) and the ***static head of the pump H_{stp}*** ($= z_d - z_s$). The static head is the sum of the vertical distance between the free liquid surface in the suction and discharge tanks [fig. 4.13(a)], and is expressed as:

$$H_{st} = H_s + H_d + H_{stp} \qquad (4.20)$$

The values of suction static head might be positive or negative and are dependent on the pump's position in relation to this.

So, if the pump is located below the liquid level in the suction tank, for calculations of the total amount of static head, the suction static head takes negative value because the liquid flows to the pump under the influence of gravity [fig. 4.13(b)]. Contrary, if the pump is above the liquid surface of the suction tank, it takes positive value [fig. 4.13(a)]. A small amount of static head that is presented by the pump (and is a structural characteristic) is, in most cases, negligible. Therefore the relationship of static head is given as:

$$H_{st} = H_s + H_d + H_{stp} = z_2 - z_1 \qquad (4.21)$$

$$\text{because } H_{stp} = 0$$

where z_1 is the elevation of the suction datum and z_2 the elevation of the discharge.

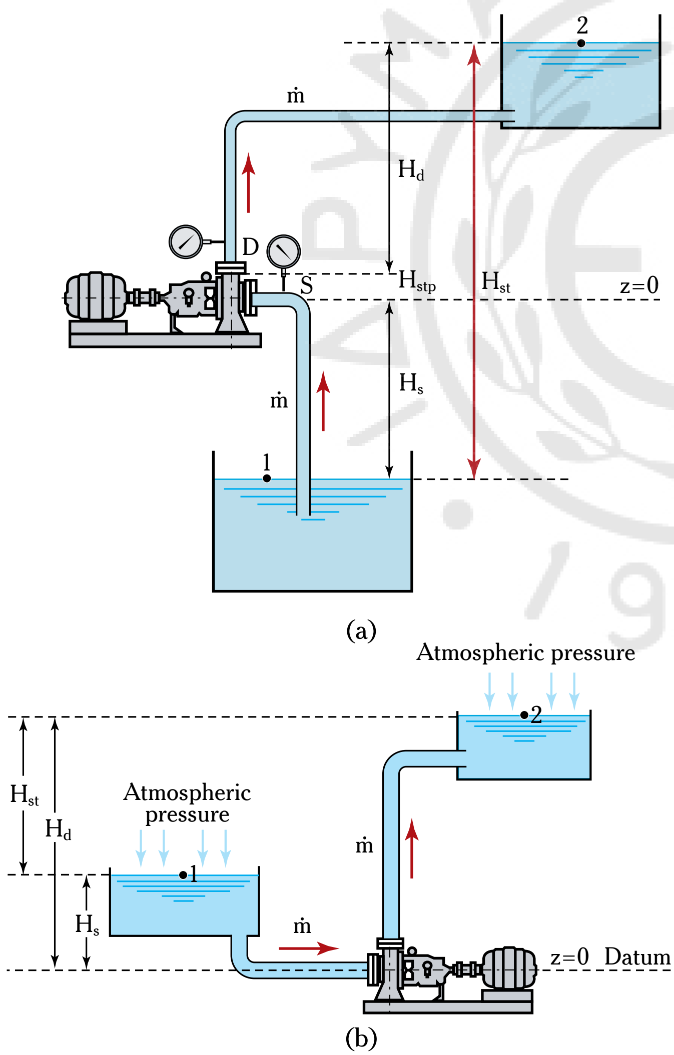


Fig. 4.13
Static heads of a pumping system.

2) The ***pressure head*** or flow work. In a liquid this is p/γ.

Pressure head of the pumping system is the change in pressure between the beginning and end of the system. It equals the change in pressure to the specific gravity of the pumped liquid:

$$H_p = \frac{p_2 - p_1}{\gamma} \qquad (4.22)$$

If the suction and discharge of the system have the same pressure ($p_1 = p_2$), as illustrated in the system of figure 4.14, the pressure head of the pumping system is zero.

And in this case, it is distinguished in suction pressure head $(p_S - p_1)/\gamma$, discharge pressure head $(p_2 - p_D)/\gamma$ and pump pressure head $(p_D - p_S)/\gamma$. The peculiarity is that the pressure head to the suction and discharge are usually negative (since during the flow through the pipes occurs pressure drop: $p_S < p_1$ and $p_2 < p_D$). In contrary, the pump pressure head is too large ($p_S << p_D$), because as we have seen, the energy transmitted from the pump to the liquid at its output has mainly the form of pressure energy. In the diagram of figure 4.14, the pressure variation from the liquid surface in the first tank is illustrated as well as the variations in relation to the atmospheric pressure up to the point 2 on the liquid surface in the second tank. ***The pressure head that the pump provides is called manometric head of the pump*** (or delivered head) and is an important feature. The manometric head of the pump is not included in the energy equation of the pumping system and will be discussed below in the section on the energy heads of the pump.

3) The ***velocity head***, the kinetic energy in a mass of flowing liquid is $(½)\,mv^2$, or $(½)\,(w/g)v^2$. The kinetic energy per unit weight (force) of this liquid is $(½)\,(W v^2/W g)$, or $v^2/2g$. ***Velocity head of the pumping system*** is the change in velocity head between the beginning and end of the system. If $v_1 = v_2$, the system velocity head is zero. The velocity head is small, with the exception of pumping systems that are designed to increase the velocity of liquid in the discharge (e.g., fire fighting pumping systems).

4) The piping losses or ***friction head***, expressing the friction losses. The head losses of the pumping system is the sum of the amount of losses in the suction pipe and the discharge pipe:

$$\Sigma H_f = H_{f1S} + H_{fd2} \qquad (4.23)$$

These energy losses are divided into:

a) The ***idle losses***, caused by the reaction of inertia due to the height of the column of liquid so that the liquid must obtain a certain velocity of movement v. The speed is defined as v_S at the pump inlet, and v_D at the discharge, because it is taken into account that there is difference in the construction of diameters between the suction and discharge pipes.

b) The ***passive losses***, due to friction, turbulence of fluid streamline, constrictions on pipe, cross-sectional enlargements, pipe bends, adjusting devices such as control valves, etc..

The energy losses during the flow inside the pump are related to energy that is not attributed to the pumped liquid, therefore not included in the energy losses of the pumping system, but to the energy losses of the pump (that are not only due to flow).

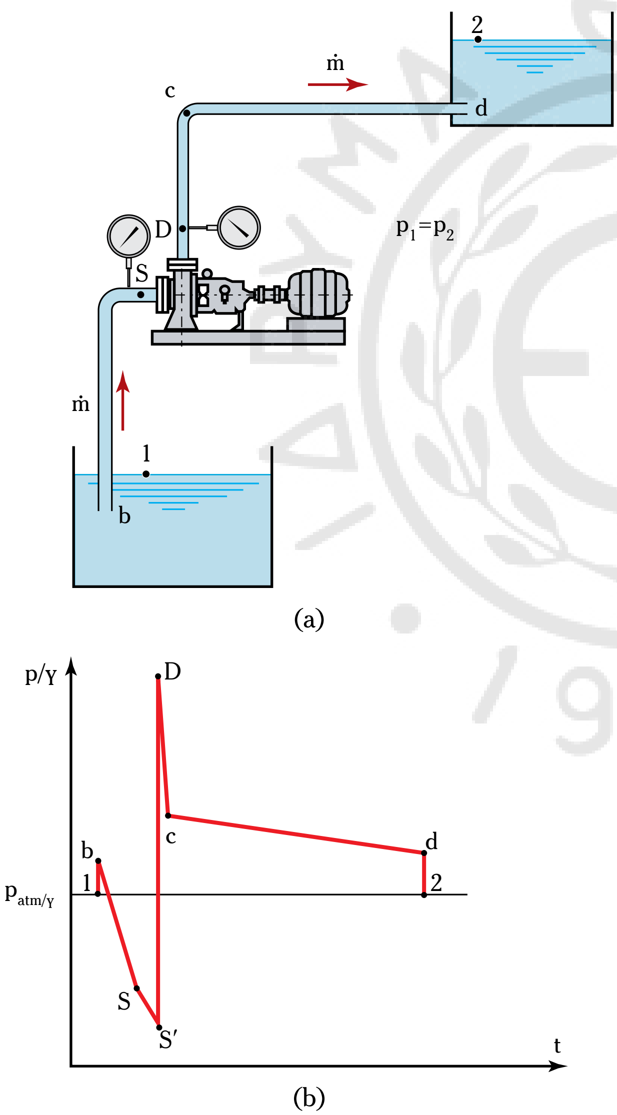


Fig. 4.14

(a) Pressure heads on pumping system, and
(b) the pressure variation diagram of fluid flow at time (t).

5) The ***total head*** of the pumping system, which expresses the energy that the pump gives to the fluid flowing through it. It is the most important head and we have already met it at the energy equation of the pumping system:

$$H=(z_2-z_1)+\frac{p_D-p_A}{\gamma}+\frac{v_D^2-v_S^2}{2\cdot g}+\Sigma h_f$$

According to equation (4.17), the total head is the sum of static head, pressure head, velocity head and the head of the pumping system losses.

2) Energy heads of the pump.

In correspondence with the above energy heads of the pumping system, we also have energy heads of the pump, which are the static head of the pump (H_{stp}), the manometric head of pump (H_{mp}), the velocity head $[(v_D{}^2-v_S{}^2)/2g]$ and the head of losses (Σh_{pf}). Of these, the most important is the manometric head of the pump, since the pump gives energy to the fluid primarily as pressure energy. The static head and velocity head of the pump are very small (often negligible), while the head of losses due to the flow inside the pump comes under total energy losses.

1) ***Manometric head*** of the pump. The mechanical work that a positive displacement pump transmits to the fluid is converted directly into pressure energy. Respectively, the mechanical work that a dynamic pump transmits to the fluid is initially converted to kinetic energy within the pump, the largest percentage of which, as the liquid exits the pump impeller, is converted into pressure energy. In both cases, the energy that increases from the inlet to the outlet of the pump is mainly the pressure energy. (Exception is the reciprocating suction pumps, in old pumping systems that are no longer used, and certain types of specific pumps).

The increase in head of pressure energy from the inlet until the outlet of the pump is called the ***delivered-manometric head*** of the pump and is denoted by H_{mp}.

The manometric head is the difference in fluid pressure energy flowing through the pump between the pump inlet and outlet points. In practice, it can be easily measured by manometers mounted at the inlet and outlet of the pump. Its name is attributed to the manner of its measurement.

In the pumping system shown in figure 4.15, point S corresponds to the pump inlet (suction) and D to its outlet (discharge). According to the definition, the manometric head is given by the equation:

$$H_{mp} = \frac{p_K - p_A}{Y} \qquad (4.24)$$

Applying equation (4.17) between S and D results:

$$H = H_{stp} + \frac{p_D - p_S}{Y} + \frac{v_D^2 - v_S^2}{2 \cdot g} + \Sigma h_f \qquad (4.25)$$

where: $H_{stp} = z_D - z_S$ is the difference in height between the pump inlet and outlet (static head of the pump). In the above equation, the friction losses in the pump are disregarded because they are incorporated to the mechanical work which is consumed by the pump and is not attributed to the fluid ($\Sigma h_{pf} = 0$).

Comparing equations (4.24) and (4.25) results in the relationship between total and manometric head of the pump:

$$H = H_{mp} + \frac{v_D^2 - v_S^2}{2 \cdot g} + H_{stp} \qquad (4.26)$$

Commonly, the diameter of the suction is higher than the depression diameter in order to obtain greater available ***Net Positive Suction Head*** – NPSHa. Accordingly, the velocity in discharge is higher than in the suction. The static head of pump H_{stp} is small

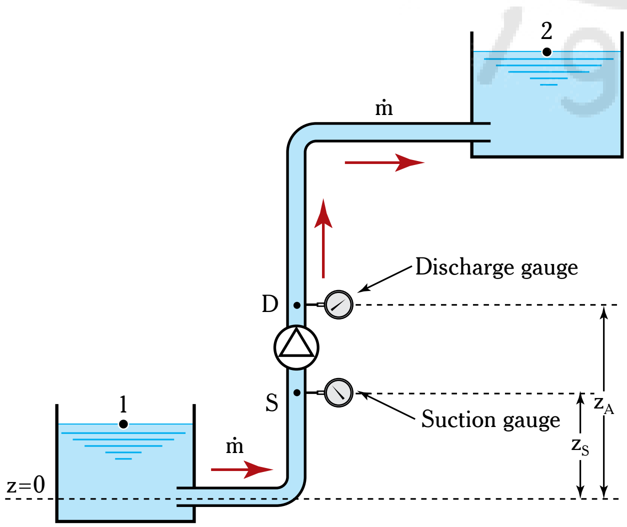


Fig. 4.15
Pump energy heads.

and it is usually not taken into account in calculations. Therefore, the total head of the pump H is slightly higher than the manometric head H_m.

Manometric suction head is the pressure head in the suction S:

$$H_{ms} = \frac{p_S - p_1}{Y} \qquad (4.27)$$

In the pumping systems that are illustrated in figs 4.15 and 4.15, the pressure at point S (p_S) is less than the pressure on the liquid surface in suction tank (p_1), due to the pressure drop in the suction pipe. If the pressure in the tank space is equivalent to the atmospheric pressure ($p_1 = p_{atm}$), the p_S is lower than atmospheric, since $p_S < p_{atm}$ creates a vacuum and the manometric suction head is negative.

Respectively, ***manometric discharge head*** is the amount of relative pressure head in discharge D:

$$H_{md} = \frac{p_D - p_2}{Y} \qquad (4.27a)$$

The manometric suction and discharge heads can be easily measured by placing manometers at points S and D.

From the equations (4.24), (4.26), (4.27) and (4.27a) arises:

$$H_{mp} = H_{md} - H_{ms} \qquad (4.27b)$$

assuming $H_{stp} = 0$ and $v_D = v_S$.

2) Inner head of the pump H_i. The ***inner head*** (or ideal head of the pump) is the amount that the pump would yield to the liquid, if, within the pump, the energy losses during the flow of the liquid or the recirculation of a proportion of liquid are eliminated. The energy losses from the liquid represent the total losses generated by the flow of liquid as it passes through the pump. Namely, it is the theoretically attributable head of the pump due to recirculation of a fluid quantity.

The inner head constitutes a barrier to the motion of the liquid streamline and is expressed in height of liquid column. It is equal to the algebraic sum of the total head H [eq. (4.25)], the height of hydraulic losses Σh_{pf} and between the suction and the discharge of the pump is given as:

$$H_i = H + \Sigma h_{pf} \qquad (4.28)$$

At low speed piston pumps the inner head is omitted because it has a small effect related to the low velocity of the liquid.

3) The ***suction heads*** are particularly critical

energy heads because they ***determine the smooth suction capabilities of a pump***. Such heads significantly affect the operation of the pump and the development of the phenomenon of cavitation. So, if we control the suction heads by studying them during pump installation and by checking them during pump operation, the creation of the phenomenon of cavitation is eliminated and the pump is protected from it.

These are the available and the required Net Positive Suction Heads, as well as the maximum permissible suction Head ($H_{a,\,max}$), where:

– ***Available net positive suction height NPSHa*** is the difference of pressure and velocity head of the liquid at the pump inlet from the pressure level at which the liquid is 'boiling':

$$NPSHa = \left(\frac{p_S}{\gamma} + \frac{v_S^2}{2g}\right) - \frac{p_l}{\gamma} = \frac{p_S - p_l}{\gamma} + \frac{v^2}{2 \cdot g} \qquad (4.29)$$

The NPSH depends on the pumping system.

– ***Required net positive suction head NPSHr*** is the drop of the energy head from the pump's inlet to its interior:

$$NPSHr = \left(\frac{p_S}{\gamma} + \frac{v_S^2}{2g}\right) - \frac{p_{S'}}{\gamma} = \frac{p_S - p_{S'}}{\gamma} + \frac{v_S^2}{2 \cdot g} \qquad (4.29a)$$

where S′ point inside the pump.

The NPSHr depends on the pump.

– ***Maximum permissible suction head $H_{s,max}$*** is the maximum suction energy head that the pump can achieve (without the liquid entering to the pump reaching boiling point due to low pressure):

$$N_{s,max} = H_{sa,max} + \frac{v_S^2}{2 \cdot g} + h_{sf} = \frac{p_1 - p_{vap}}{\gamma} - NPSHr \qquad (4.30)$$

where $H_{sts,max}$ the maximum static suction head of the pumping system.

4.5.2 Volumetric flow rate.

The volumetric pump flow rate or volume flow rate ($\dot{V}$) is the volume of the liquid attributed to the pump discharge port per time unit. It is determined in accordance with the structural and operating characteristics of the pump, such as the pump size, the speed of the rotor that rotates or reciprocates, and the piping system where the pump is connected to.

In accordance with the pump's operation and the pumping system, the volume flow rate is distinguished into:

1) ***Actual flow rate*** ($\dot{V}$), which is the flow rate of the pumping system measured in the discharge pipe. It is noted that in pumping systems with reciprocating pumps, the average (or mean) volume flow rate is different from the instantaneous one, because volume changes (increases and reductions) during the reciprocation of the pistons cause changes in the discharge flow rate. In positive displacement rotary pumps the variation of flow rate is minimal, while in dynamic pumps nonexistent. Therefore, in dynamic pumps operated under permanent flow, the differentiation between instantaneous and mean volume flow is meaningless.

The value of volume flow rate of a pumping system affects directly the head losses, since the total head is associated with flow rate by the equation (4.18) as follows:

$$H = a + b \cdot \dot{V}^2$$

where $a = H_{st} + \frac{p_2 - p_1}{\gamma}$ is the coefficient that equals the sum of the static head and the pressure head, and b is the coefficient that depends on the data of the suction and discharge pipes.

The flow rate of the pumping system depends on the system generally (e.g., pipe diameters, height differences, etc.) but mainly on the pump of the system. So, we can say that it arises as a result of a specific pump in a specific piping system.

2) ***Optimum*** or ***normal volume flow rate*** $\dot{V}_n$, which is the actual flow rate that the pump provides when it operates to the maximum of its efficiency continuously and without being damaged. Although from an energy efficiency point of view the pump must operate with a maximum degree of efficiency, this does not mean that in those conditions we will also have the maximum flow rate. So ultimately, the optimum flow rate is the energy advantageous flow rate and not the maximum one.

3) ***Inner flow rate*** ($\dot{V}_i$), which is the flow rate within the pump. In positive displacement pumps

internal leakages are observed, because a small amount of liquid is recirculated. This results in the internal flow rate being slightly higher than the actual one. In dynamic pumps, the inner flow rate is much higher than the actual one, because the impeller does not make contact with the case, so more quantity of liquid is recirculated.

Hence, it is clear that the internal volume flow rate is equal to the actual one increased by the internal leakages:

$$\dot{V}_i = \dot{V} + \dot{V}_L \tag{4.31}$$

4) ***Theoretical volume flow rate*** ($\dot{V}_{th}$), which is the liquid flow rate that must be discharged from the pump if internal and external leakages to the environment are eliminated:

$$\dot{V}_{th} = \dot{V} + \dot{V}_L + \dot{V}_{env} \tag{4.32}$$

where: $\dot{V}_L$ internal leakages, $\dot{V}_{env}$ the spills of the pump to the environment (e.g., in the engine room where a pump is installed). Generally applies that $\dot{V}_{th} > \dot{V}_i > \dot{V}$.

5) ***Nominal flow rate***[5] ($\dot{V}_N$), which is the volume flow rate that applies for the operation of the pump at rated speed v_N, nominal total head H_N and for pumped liquid in accordance to that specified in the pump's technical characteristics.

4.5.3 Pump power.

The pump does not produce energy, but receives the necessary energy from the prime movers (or pump drivers). The energy is transferred from the prime mover to the pump and from it to the passing fluid. In practice this is impossible to be realized without energy losses due to frictions. Considering that, in terms of the operating costs of the pumping system, the most important parameter is the energy consumption per unit of time, and more specifically the power, it becomes obvious how important the determination of consumed as well as effective power at a pumping system is.

Generally, the transfer of power is illustrated in the diagram of figure 4.16.

Since the pump is the mediator for the power transfer from the drive machine to the pumped liquid, power is divided as follows:

1) The ***pump motor rated power P_m*** is the power (mechanical work per time unit) that is produced by the drive machine and attributed to the pump. The prime mover may be an electric motor or a heat engine. The power transfer to the pump is attained by a mechanical transmission system. Depending on the pumping system and the pump, the power (P_m) which the engine sends to the pump can range from a few to several thousands of kW.

2) The ***pump shaft power P_s*** is the power transferred from the motor to the shaft of the pump. At the power transmission from the drive machine to the shaft of the pump there are some mechanical friction losses. If η_{pm} is the efficiency of the power transmission system, then:

$$P_s = \eta_{pm} \cdot P_m \Rightarrow \eta_{pm} = \frac{P_s}{P_m} \tag{4.33}$$

If the transmission system is single, as often happens in pumps where the motor operates solely for them, and the pump shaft begins from the drive machine, the $\eta_{pm} = 1$, since the shaft power is equal to the motor power. Also, the shaft power represents the energy consumption of the pump.

3) The ***inner hydraulic power*** or drag power P_D is the power which is transferred from the pump and more particularly from the impeller to the liquid that circulates inside dynamic pumps. This power is smaller than the shaft power, because one part of it is consumed to overcome the mechanical friction in

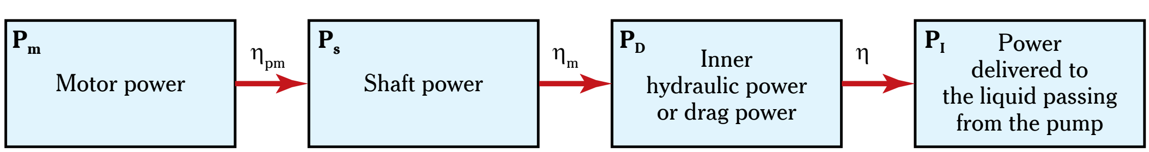


Fig. 4.16
Diagram of power transfer.

[5] ***Nominal flow rate***, ***velocity and total head*** are the theoretical (typical) or structural design values of flow rate, velocity and total head, which have been measured for a given fluid or certain given operating conditions so that they can be taken as reference points when we study a piece of machinery (e.g. a pump).

the bearings and in the sealing system. The internal hydraulic power corresponds to the inner attributable height and to the internal pump flow rate:

$$P_D = P_s - \Sigma P_m = \gamma \cdot \dot{V}_i \cdot H_D \qquad (4.34)$$

where: γ, is the specific weight, $\dot{V}_i$ the internal volume flow rate, H_D the height (in m) and ΣP_m the power losses due to mechanical friction. The power losses due to friction of the fluid inside the pump are given by:

$$\Sigma P_p = \gamma \cdot \dot{V}_i \cdot \Sigma h_{pf} \qquad (4.35)$$

4) The power delivered to the liquid or ***actual power P_I*** is the power which is attributed to the flowing fluid of the pump, since the power is not transferred in total to the passing fluid. A portion of the power is consumed for the quantity of fluid that circulates within the pump and not discharged (internal leakage $\dot{V}_L$), and also for leaks to the environment ($\dot{V}_{env}$). Another portion of the power faces losses due to friction of the fluid inside the pump. The delivered power is given by:

$$P_I = P_D - \Sigma P_L - \Sigma P_p \qquad (4.36)$$

where ΣP_L are the power losses due to leakages and ΣP_p the power losses due to friction of the fluid inside the pump.

The power delivered to the liquid is the beneficial energy result of the pump. It corresponds to the total head H and the actual flow rate $\dot{V}$. Using these characteristic qualities, the delivered power associated with the relation (4.19) is as follows:

$$P_I = \gamma \cdot \dot{V} \cdot H$$

4.5.4 Pump efficiency.

The efficiency of a pump equals the ratio of useful attributable power to the power that is consumed. The greater this ratio is, the better the pump (sometimes it is also called "energy efficiency ratio").

Also, through the operation of the pump, the fluid flow is followed by the development of losses corresponding to the power difference that is provided by the motor to the power that the fluid finally receives. Hence, the work that the pump provides is less than this supplied to the shaft.

The intention by the pump use is to attribute energy height (total head H) to the transferred liquid, so as to achieve a flow rate $\dot{V}$ in the pumping system. For that to occur, the pump should provide power p_I to the transferred fluid.

According to the above observations and depending on the nature of the losses, the efficiency of the pump is divided into:

1) ***Volumetric efficiency*** (η_V), which is the ratio of actual provision of $\dot{V}$ to the internal flow rate losses $\dot{V}_i$ of the pump. The term "volumetric efficiency" provides a measure of losses due to leakage from the clearances between the rotating impeller and the stationary casing, or of imperfect sealing of the valves, the losses from the gland packings, etc. So, the phenomenon of the slip of the liquid from the space of the discharge to the suction or from the interior of the pump to the environment is observed. The volumetric efficiency is given by:

$$\eta_V = \frac{\dot{V}}{\dot{V}_i} \qquad (4.37)$$

or in accordance to the relation (4.31), where $\dot{V}_i = \dot{V} + \dot{V}_L$

$$\eta_V = \frac{\dot{V}}{\dot{V} + \dot{V}_L} < 1 \qquad (4.37a)$$

The volumetric efficiency ranges between 80 and 98%, with higher values corresponding to positive displacement pumps that are in perfect condition, while smaller values correspond to centrifugal pumps, especially when due to wear the clearances between movable and stable parts have increased.

2) ***Hydraulic efficiency*** (η_h), which is related to the construction and installation of the pump. It depends on hydraulic losses developed at the inlet portion, internally from the inlet section to the outlet section, and the outlet portion of the pump. It is the measure of the resistance resulting from the friction of liquid on the inner surfaces of the pump, the acceleration and deceleration of the liquid as well as the changes in flow direction. It is given as:

$$\eta_h = \frac{H}{H_D} \qquad (4.38)$$

and is defined as the ratio of the total head H to the inner (theoretical) head H_{th}. And the inner head consists of the total energy which the fluid receives from the pump and the hydraulic losses. Thus, the relation (4.38), taking into consideration the equation (4.28), becomes:

$$\eta_h = \frac{H}{H + \Sigma h_{pf}} < 1 \qquad (4.38a)$$

wherein Σh_{pf} the height due to flow losses (hydraulic loss) inside the pump.

According to equation (4.38a), the larger the height losses within the pump, the smaller the hydraulic efficiency.

3) ***Mechanical efficiency*** (η_m), which is the ratio of the internal power P_D (i.e., the power transmitted to the liquid flowing into the pump) to the shaft power P_s (which is the energy consumption of the pump). It mainly depends on the friction forces that develop between the moving parts, i.e., losses associated with bearings, seals and other contacting areas within the pump. So, mechanical efficiency is defined as the ratio:

$$\eta_m = \frac{P_D}{P_S} \qquad (4.39)$$

The shaft power P_s which is transmitted to the pump will be converted into effective power P_I to the transferred liquid and into mechanical losses power ΣP_m that is available to overcome the frictional losses of the moving parts of the pump. Thus, according to the equation (4.34), the mechanical efficiency is given as:

$$\eta_m = \frac{P_D}{P_D + \Sigma P_m} < 1 \qquad (4.39a)$$

4) ***Total efficiency*** (η), which is the determinant ratio of quality in energy terms. Therefore it is more simply called "pump efficiency". It is equal to the quotient of hydraulic-delivered to the liquid power P_I to the consumed shaft power from the prime mover P_s, by providing a measure of the total friction losses, and is defined as:

$$\eta = \frac{P_I}{P_S} \qquad (4.40)$$

According to equation (4.19), where: $P_I = \gamma \cdot \dot{V} \cdot H$, from equation (4.40) in combination with equation (4.39) results that:

$$\eta = \frac{P_I}{P_s} = \frac{\gamma \cdot \dot{V} \cdot H}{\dfrac{\gamma \cdot \dot{V}_i \cdot H_\varepsilon}{\eta_m}} = \eta_m = \frac{\gamma \cdot \dot{V} \cdot H}{\gamma \cdot \dot{V}_i \cdot H_D}$$

or

$$\eta = \eta_m \cdot \eta_V \cdot \eta_h \qquad (4.41)$$

That is, the efficiency n is the product of three efficiencies, hydraulic, volumetric and mechanical. For piston pumps it is determined experimentally and remains almost stable at wide height variations, while in dynamic pumps there is a maximum efficiency at the optimum height and fall of yield when it deviates from it. In rotary pumps the hydraulic losses caused by internal frictions and turbulence flow are negligible, so that in practical applications the n_h is taken as equal to one and the relationship of pump efficiency is given as:

$$\eta = \eta_m \cdot \eta_V \qquad (4.42)$$

The total or maximum pump efficiency is one of the most important pump characteristics. It usually ranges from 70% to 90%. However, in combination with the operating conditions (i.e. the requirements of the pumping system), the pump efficiency might be considerably smaller.

The prime mover power is the sum of the shaft power and the power losses due to weakness in converting electrical energy into kinetic energy. Thus, the motor power can be calculated by the ratio of the shaft power to the engine efficiency and is given by:

$$P_m = \frac{P_s}{\eta_{pm}}, \qquad (4.42a)$$

while the axial power, which is the sum of the inner hydraulic power and power losses in transmission of power from the shaft to the transferred liquid, is given by the relationship:

$$P_s = \frac{P_D}{\eta_m} \qquad (4.42b)$$

The power transmission conveyed from the driving motor to the transferred liquid of a pumping system is illustrated in figure 4.17.

4.6 Pump suction.

The term ***pump suction*** is defined as the inlet of the liquid in the pump chamber, which is achieved in various conditions. These conditions refer either to the free liquid surface with respect to the pump, or to the space (volume of cylinder) inside the pump, and to the low pressure which is created by the reciprocation of a piston inside it or the rotation of a rotor.

When the pumping of a liquid is carried out by a tank, which is positioned above the installed pump and on the free surface prevails atmospheric pressure [fig. 4.13(a)], then the liquid flows to the pump. The flow of liquid takes place because of the suction static head, which is the result of the effect of atmospheric pressure on the free surface of the liquid. In this case, the pump provides energy to the fluid flow only for its passage from the rest of the system.

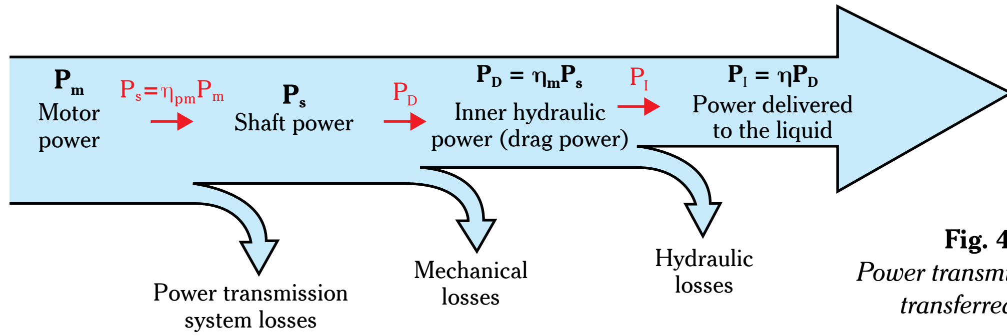


Fig. 4.17
Power transmission to the transferred liquid.

When the tank is lower than the installed pump, at the pump inlet it is necessary to create pressure drop, so that with the effect of atmospheric pressure the liquid will be transferred into the pump [fig. 4.13(b)]. The pressure drop is the desired ***suction vacuum*** of the pump and it has less absolute pressure value than the pressure which prevails on the free surface of the liquid. So, the liquid, due to the difference of pressure prevailing, is forced through the tube from the higher pressure space, where the pressure is atmospheric, to the suction chamber, where the pressure is lower.

The difference between the two pressures may theoretically reach a value of one atmosphere or the current barometric pressure which is exerted on the liquid surface, assuming that the pressure drop created by the pump reaches the perfect vacuum. This is named ***theoretical suction head*** and it reaches up to 10,33 m. The theoretical suction head is almost impossible to be realized, because more factors affect the suction side apart from the height of liquid and the effect of atmospheric pressure. The factors that should be considered in calculating the actual suction of a pump are:

1) The ***Net Positive Suction Head*** or NPSH, which is used to control the suction conditions under which the pump should be operated in order to avoid cavitation (fig. 4.18). The net positive suction head is determined by:

a) The head H_{atm}, which is the absolute pressure (or for open containers the atmospheric pressure) and it measures in water column meters m_{H_2O} or m_{wc}, which acts to the surface of the fluid in the suction side.

b) The vertical distance H_s $(z_2 - z_1)$ from the center of the pump to the free liquid surface that expresses the static suction head and is added algebraically to the head of atmospheric pressure H_{atm} on the liquid surface.

c) The frictional losses in the suction pipe that form the friction head H_f.

d) The velocity head losses H_v due to the fluid flow velocity. These are taken into consideration only at very high flow rates, otherwise they are negligible.

The pump suction can be considered sufficient, when the sum of these losses $H_v + H_f \pm H_s$ is less than H_{atm} that is exerted on the liquid surface on the suction side. These losses must be added to the losses due to evaporation of liquid H_{vap} and the remaining positive suction head required by the pump, to reach the discharge rate for which it was designed. Thus, the sum of the total losses is given by the $H_{atm} \pm H_s - H_f - H_v - H_{vap}$, known as available net positive suction head NPSHa. In systems where the head of velocity head losses (H_v) is negligible, it is omitted.

The available net positive suction height NPSHa refers to the installation (suction side system) and

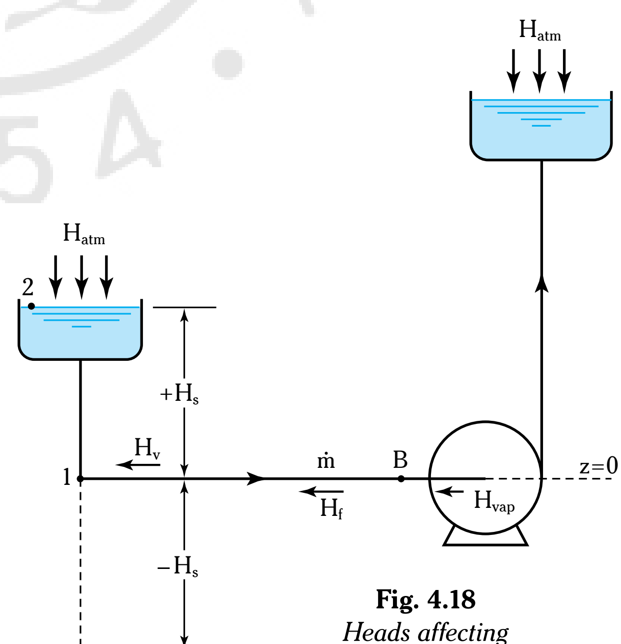


Fig. 4.18
Heads affecting the pump suction.

to calculate it we distinguish two cases. When we have suction lift where the pump is located above the free surface of the suction of liquid (fig. 4.18), the suction static head takes negative value, while when we have positive suction where the pump is located below the liquid surface, static suction head takes positive value.

Pump manufacturers also provide the required net positive suction head NPSHr, which characterizes the pump and is measured in water column meters. It is a function of pump flow and usually increases as the flow rate increases.

The relationship between required (NPSHr) and available net positive suction height (NPSHa) must be such that the available net positive suction head is always greater than the required one (NPSHa > NPSHr). The low value of the available net positive suction head results in cavitation which is created by the bubbling into the liquid during the operation of the pump. Also, it could cause increase in noise level over the prescribed limit, as well as removal of material from the surfaces inside the pump at higher rate than the specified limit of allowable wear.

2) The ***temperature*** of the liquid, since the warmer the liquid is, the more difficult it is for it to be sucked by the pump. This is because under the influence of vacuum in suction evaporation of the liquid is facilitated. Vapors generated occupy space and prevent suction. The vapors production rate is increased as the fluid temperature increases. This phenomenon is particularly important in pumps of certain pumping systems for liquid fuel cargoes, which, even at low temperatures, have high volatility, interrupting the streamline of cargo during suction.

In cases of high temperature of the liquid, the pump is placed as far as possible below the level of liquid to be pumped, in order to ensure suction capability. This happens for example in the condensate pumps of the main steam condenser or in feed water pumps of steam boilers.

3) The ***specific gravity*** of the liquid, which affects inversely the pump suction capability and depends on the weight of the liquid contained in the suction column. Thus, the smaller the specific gravity of the liquid, the easier the suction of the pump is achieved and vice versa.

4) The ***viscosity*** of the liquid. A liquid is characterized as "viscous" when it can hardly be drawn and as "light" when it is easily sucked.

5) The ***resistances in the suction piping***. The fewer the resistances, the easier the suction of the pump is realized.

The resistances depend on:

a) The ***diameter of the tube***. The larger the diameter, the more the resistance decreases.

b) The ***internal surface texture (roughness)*** of the pipes. When they are smooth, there are no friction forces between the liquid and the internal surface of the pipe.

c) The ***direction of the pipes***. When they are straight without many variations in diameter, the frictions by turbulence of liquid are reduced.

d) The ***interferences in the pipe*** by regulating equipment. The less interferences there are, the smoother the fluid flows.

6) The sealing of the suction pipe and the entire pump mechanism. With the adequate sealing of pipes, valves and pistons in piston pumps, better suction of liquid is achieved. The same applies to rotating pumps, where the clearances of the rotor to the housing must be as small as possible.

Also, the suction capability of a pump is influenced by its type. The suction is affected negatively by the large number of strokes in piston pumps, while in rotary pumps by revolutions per minute, if they are many. Also the suction capability is affected by the clearance between the piston and the cover, making the suction difficult when the clearance is large.

Practically, the suction height reaches only 7 m in ideal conditions, due to losses from frictions in the suction pipe, and the limits placed on the construction of a pump. Additionally, any increase in water temperature over 15°C will cause undesired effects by influencing the pressure in suction through the creation of vapors.

Hence, when the water temperature is above 75°C, the suction head must be positive or, if this is impossible, the suction pipe must be short near the suction point, without curves which cause flow changes. Furthermore it must have smooth interior surfaces and low flow rate, less than 1 m/s.

In an effort to reduce losses affecting the suction head of piston pumps, aeration is performed by suitably positioned air vent valves on the pump. By these valves, the air within the pump casing or the air pockets (that are formed in portions of the piping system and affect the flow) are eliminated. At pump starting these spaces must initially be filled with liquid.

The centrifugal pumps, which do not have valves and pistons that interrupt the flow of fluid to create the conditions for the removal of air, are frequently positioned below the liquid level suction. In centrifugal pumps that are positioned higher than the level of liquid suction, there are priming pumps installed in the suction pipe (devices for initial suction that remove the air from the pump casing), or arrangements to fill the suction pipe with liquid.

4.7 Pump discharge.

The term "discharge" describes the extraction of fluid from the pump to the piping system that the pump is installed in. Theoretically, the discharge of a pump could be performed in unlimited height, since the liquid is incompressible and the pump operation is based either on the displacement of the liquid (as happens in positive displacement pumps), or on the conversion of the kinetic condition and the change of the kinetic energy into static pressure (as happens in dynamic pumps).

However, in practice, a part of the energy supplied to the pump is dissipated in the environment as heat from the frictions of the moving parts. The rest will be converted into increase in pressure and fluid flow rate.

By increasing the pressure generated, one portion will be lost on loss from frictional resistance to the discharge network, another portion as static head of the system and another as pressure on the free surface of the liquid in the vessel of the discharge area. The head of losses due to resistance in velocity, as also happens in the suction side, is negligible and so it is omitted.

Thus, to achieve the desired discharge head of the pump, the losses described above are treated by:

1) special features in pump construction,

2) a study of the characteristics of the piping system, and

3) a calculation of the total losses due to frictions.

According to the construction characteristics and capacity of pumps:

1) discharge at great heights is achieved by positive displacement pumps, rotary or piston, and multistage centrifugal pumps, while

2) for discharge at lower height and increase in the speed of the liquid, centrifugal pumps are used.

4.8 Effect of physical properties of liquid to the pump.

The most important physical properties of fluids that affect (in energy or functional terms) the pumping system and particularly the pump, are as follows:

1) Density, Specific Weight and Specific Gravity.

The density of a fluid is the mass per unit of volume, usually designated by the symbol ρ and is given as $\rho = m/V$.

The specific weight of a fluid is its weight per volume unit and is usually designated by the symbol γ. It is related to density as $\gamma = \rho \cdot g$, where g is gravity.

The specific gravity is a dimensionless unit defined as the ratio of the density of a substance to the density of water (at a specified temperature) and can be expressed as $SG = \rho_{substance} / \rho_{H_2O}$.

The pressure head (p/γ) depends on the specific weight. The other terms of the total head (i.e. the dynamic and kinetic energy heads) are independent of the specific weight of the liquid. More important is its effect on the output of the delivered power P_I. As seen from equation (4.19), the delivered power is proportional to the specific weight.

2) Compressibility.

All liquids are practically incompressible. This does not affect the energy of the pumping system. However, it creates serious problems in the sudden change in their kinetic condition within the pipes (water hammer). So special attention should be paid during pump operation so that at transitional conditions (starting and stopping) operation is as smooth as possible. The problem is particularly important in reciprocating pumps, where it is treated by using air accumulators.

3) Vapour Pressure.

Fluids will evaporate unless prevented from doing so by external pressure. The vapor pressure affects directly the net positive suction head available (NPSHa) and the maximum suction height Ha max, which can be achieved by the pumping system. The more volatile a liquid is (i.e. the greater the vapor pressure for a given temperature), the lower the suction head to which the pump can operate efficiently, and the greater the risk of cavitation.

4) Solubility of gases.

The high solubility of certain gases in liquids, especially in water, may cause problems in the pumping system, both energy problems (decrease of pump efficiency) and functional ones (cavitation). These problems due to the solubility of gases in liquids are reduced with increasing temperature and pressure drop (Henry's law). Possible results of the reduction of solubility are the formation of bubbles, the creation of undesirable air pockets in transfer pipes and, primarily, in the introduction of the liquid from the suction pipe to the pump. In this critical area (the suction to the pump), the maximum pressure drop is observed, hence the minimum solubility of the gases.

5) Corrosive properties of liquids.

Pumped fluids are in constant contact with the internal surfaces of pipes and pumps. The corrosive environment which is generated by many of these fluids threatens the pumps with serious damage, initially reducing their performance, and often leading to complete scrapping of parts or even whole pumps. To address the corrosion caused by hazardous liquids, pumps made of special materials are used. Therefore, for acidic solutions with pH from 0 to 4, resistant metals are used (e.g., stainless steel, chromium or alloy of steel chromium-nickel with molybdenum, titanium, chromium or siliceous iron, lead hardening with antimony), or ceramic materials and plastics (with temperature restrictions: PVC to 60°C, polyethylene to 120°C). For liquids with pH 4 to 6, various brass alloys are used. For pH 6 to 9, iron is used. For pH 9 to 14, cast or monel (Ni-Cu alloy) is used. It is evident that only the parts that come in contact with the corrosive liquid are made (or coated) of durable materials.

6) Viscosity.

The viscosity of the liquid affects the friction coefficient, and hence the head of loss. Apart from this, liquids with high viscosity create problems in dynamic pumps. Therefore, the pumping of viscous liquids is usually performed by static rotary pumps.

7) Temperature.

As mentioned, in relation to the solubility of the gases, the increase of temperature during pumping leads to a decrease in the solubility of the gases and increase of vapor pressure. Consequently, conditions of gas generation are created, and the risk of cavitation is increased.

CHAPTER FIVE
Flow in dynamic pumps

General introduction.

In the previous chapter the role of pumps was studied from an energy and efficiency perspective within a pumping system, which, as mentioned, consists of three subsystems: the suction (suction tank and pipe), the pump, and the discharge (discharge pipe and possibly discharge tank). In this study, the pump was considered as a device that delivers the energy required for the operation of the system. Although a description of these devices has already been provided (Ch. 3 and 4), so that in general there is the picture of the capabilities and limitations of the pumps, the question remains whether a particular pump placed in the pumping system can respond to the energy that is required for fluid transfer through a piping system.

In fact, the pump not only provides the energy to the system but also, in combination with the subsystems of suction and discharge, creates the final result of pumping operation. This, in some cases, may be readily assessed. If, for example, in a pumping system an energy sufficient reciprocating pump is used, the flow rate of the system is determined exclusively by it, since it will scarcely be influenced by the energy load of the system. The same happens if the pump is of the rotary static type. In fact, the flow rate could be regulated directly by changing the speed.

With dynamic pumps, the situation is more complex and is presented in this chapter. The flow rate here depends on both the pump and the pumping system. And of course, what happens to the flow is extended to losses of head (proportional to the square of supply), to the delivered head, to the power, etc. Therefore, when a dynamic pump is studied in a pumping system, the knowledge of the mode in which the system influences the pump and the pump influences the system is required. It is necessary therefore for the flow of fluid through the dynamic pumps to be examined, focusing on centrifugal pumps, which are widely used. Note that, similar phenomena are observed in rotodynamic pumps.

5.1 Flow in a centrifugal pump.

The study of liquid flow inside the pump is an essential condition for the design and the understanding of the operation of dynamic pumps. Due to the complexity of the liquid flow, the study is performed in two levels:

1) Study of theoretical flow conditions.
During the study of the theoretical conditions, it is considered that the flow renders pump efficiency equal to 100%.

2) Study of various losses of the pump.
These are losses of head and power, due to hydraulic frictions and internal leakages in the pump. The complexity of the flow inside centrifugal pumps, in comparison to the flow in a pipe which seems very simple, makes the study impossible without the aid of the ***theory of similarity and experimentation***.

For the study of the flow inside the centrifugal pump, the paths through which the flow of liquid inside the pump is developed have to be clarified. According to figure 5.1 there are three paths: the

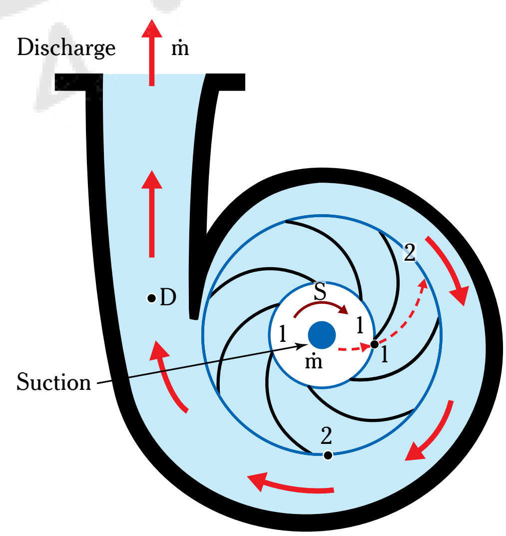


Fig. 5.1
Flow of liquid through a centrifugal pump.

first path S-1, where the liquid enters the suction in the center of the ***impeller S*** and, dispersed radially, reaches the ***base of the vanes which is the region*** (1) with a radius r_1 (fig. 5.2). This path is short and without energy significance. However, it is of great importance as regards further pressure drop in the pump inlet and the risk of creating cavitation conditions.

In the ***second path 1-2***, the liquid, driven by the blades and sliding there due to the centrifugal force, moves through the ducts forming the consecutively vanes, from the inner circumference (1) ***with radius r_1 towards the outer circumference of the impeller*** (2) with radius r_2. During this path the pump attaches energy to the liquid and increases significantly its kinetic energy and the pressure energy. It is the most interesting path for the design and operation of the pump, since throughout its duration the energy is transferred from the shaft of the pump to the liquid. For this reason it will be discussed further in detail below.

At the ***third path 2-D***, the liquid, leaving the impeller at high speed, enters the volute of the casing trough and moves pivotally towards the ***discharge D***. The increase of volute cross-section and the hopper outlet results in the conversion of kinetic energy into pressure.

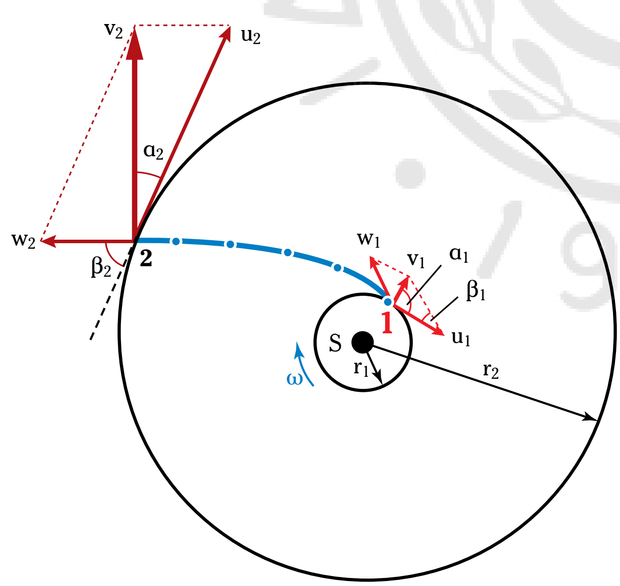


Fig. 5.2
Flow of liquid in impeller.

– *The flow of the liquid in the impeller - Triangles of speed.*

For the energy study of fluid flow from the base to the circumference of the impeller, we initially have to admit that the flow is ideal. This specifically means that:

1) The frictions are not taken into account (it is considered that they do not exist).

2) It is considered that there is an infinite number of vanes with nonentity-thickness and infinitesimal distance between them.

This ideal condition is called ***perfect guidance***. A first consequence of this case is that the flow is considered one-dimensional[1] (since the thickness of the liquid between two blades is infinitesimal). A second is that there is an exact symmetry and thereby, at all points that are at distance r, far from the pivot axis (regional locations), the liquid has equal magnitude of velocity vector. In the study of fluid flow, the laws of incompressible liquids are used (the continuity equation and the Bernoulli equation), and the laws of the rotary motion (as they are supplemented below). The movement of an elementary liquid mass, as pushed by the blade while sliding on it, is illustrated in figure 5.2.

At point 1 (base of the blade) the elementary mass of the liquid rotates under the thrust of the blade with ***angular velocity "ω"*** and simultaneously slides along the blade under the effect of centrifugal force. The ***tangential component*** of the velocity (due to the rotational movement) is u_1, and the tangential component on the blade velocity (the sliding velocity on the blade) is w_1, which is represented by the normal component of the velocity vector. The ***resultant velocity vector*** is $\dot{v}_1$ and equals with:

$$\dot{v}_1 = \dot{u}_1 + \dot{w}_1 \tag{5.1}$$

It is observed that the a_1 between vectors of velocities v_1 and u_1 is approximately right angle (the appropriate design of the base of the blade is important). This means that it can be considered that the resultant velocity v_1 has radial direction.

At point 2 (at the end of the blade) the tangential velocity component is u_2, and w_2 is the tangential to the blade, so that the resultant velocity vector v_2 is:

---

[1] One-dimensional flow means that the motion takes place in one direction. In general, one-dimensional flow consists of a simple velocity component in one specific coordinate direction.

$$\dot{v}_2 = \dot{u}_2 + \dot{w}_2 \tag{5.2}$$

The velocity is given by the equation as follows:

$$v_2 = \sqrt{w_2^2 + u_2^2 - 2 \cdot w_2 \cdot u_2 \cdot \cos\beta_2} \tag{5.2a}$$

Regarding the velocities, the following applies:

1) The tangential component of the speed in accordance with the laws of circular motion is proportional to the radius:

$$u_1 = \omega \cdot r_1 \tag{5.3a}$$

$$u_2 = \omega \cdot r_2 \tag{5.3b}$$

Since the radius r_2 is much greater than r_1, the tangential component u_2 is also much greater than u_1. So the relations (5.3a) and (5.3b) indicate that:

$$\frac{u_2}{u_1} = \frac{r_2}{r_1} \tag{5.3c}$$

2) The slide on the blade is performed by the influence of centrifugal force. If the perpendicular to the flow elementary area dA remained constant, the magnitude of tangential component w would increase significantly from point 1 to point 2. But, as the radius increases, the area A of the cross section is increased accordingly. This increase, according to the equation of continuity, tends to reduce the magnitude of the tangential w and compensates the tendency of its increase due to the centripetal force.

The result of the changes in these two velocity components, as the liquid moves from point 1 to point 2, is that the tangential component u is greater in relation to the tangential component w. This means that the angle α_2 between v_2 and u_2 velocity vectors is much smaller than the corresponding angle a_1 at point 1 (fig. 5.2).

The angle β_2, formed by the tangent to the blade velocity w_2 with the tangent to the circumference of the impeller at point 2, constitutes an important design angle of the impeller. This is because on this angle depends the direction of component velocity v_2, by which the elementary mass of the liquid leaves the vanes of the impeller. Note that the direction of the component w_2 is not radial ($\beta_2 \neq 90°$). More specifically: $\beta_2 < 90°$.

The triangles that are created during the synthesis of the vectors u and w are known as ***velocities triangles***. The velocities triangle where the elementary mass of liquid, taking the necessary energy, is leaving the impeller is illustrated in figure 5.3 at point 2.

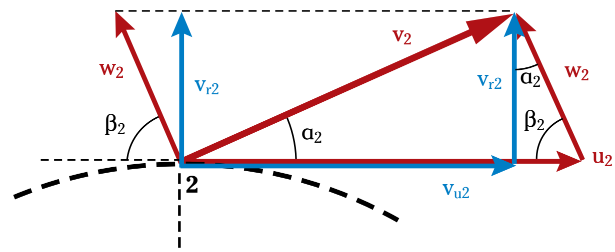


Fig. 5.3
Velocity triangle as the fluid is leaving the impeller.

The resultant velocity vector v_2 can now be analyzed in two new components: The ***radial component*** v_{r2} and the ***tangential component*** v_{u2}. From the right triangle that is formed by the new components (fig. 5.3), occurs:

$$v_{r2} = v_2 \cdot \sin\alpha_2 \tag{5.4a}$$

$$v_{u2} = v_2 \cdot \cos\alpha_2 \tag{5.4b}$$

The angular momentum[2] L_2 of elementary mass m in point 2 is given by:

$$L_2 = m \cdot r_2 \cdot v_2 \cos\alpha_2 \tag{5.4c}$$

Corresponding velocity triangle we also have in point 1 (fig. 5.4), for which it arises:

$$v_{r1} = v_1 \cdot \sin\alpha_1 \tag{5.5a}$$

$$v_{u1} = v_1 \cdot \cos\alpha_1 \tag{5.5b}$$

$$L_1 = m \cdot r_1 \cdot v_1 \cdot \cos\alpha_1 \tag{5.5c}$$

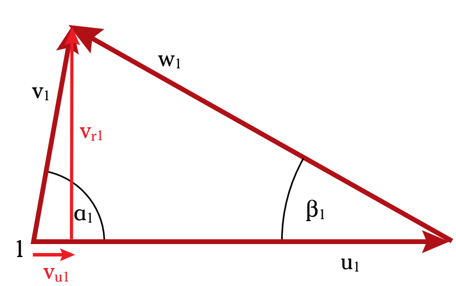


Fig. 5.4
Velocity triangle as the fluid is entering the impeller.

[2] See, *Φυσική* ("Physics") by Antonios Vroulos and Stefanos Karnavas, Athens: Eugenides Foundation publications, 2012.

In the case where $\alpha_1 = 90^o$:

$$v_{r1} = v_1 \text{ and } v_{u1} = 0.$$

Because the absolute symmetry of the system has been set as an assumption, what applies on the elementary mass in point 2, applies for the total mass that is located on the circumference of the impeller. Therefore, at the equations (5.4c) and (5.5c), the mass m is the mass that passes through the periphery of the impeller and is:

$$\frac{m}{t} = \dot{m} = \rho \cdot \dot{V}$$

where: $\dot{m}$ is the mass flow rate and ρ the density of the fluid.

According to Newton's second law for rotation, the change of angular momentum is equal to the external torque (M) multiplied by time (t):

$$M \cdot t = L_2 - L_1 = m \cdot (r_2 \cdot v_2 \cdot \cos\alpha_2 - r_1 \cdot v_1 \cdot \cos\alpha_1) \Rightarrow$$

$$\Rightarrow M = \dot{m} \cdot (r_2 \cdot v_2 \cdot \cos\alpha_2 - r_1 \cdot v_1 \cdot \cos\alpha_1) \Rightarrow$$

$$\Rightarrow M = \rho \cdot \dot{V} \cdot (r_2 \cdot v_2 \cdot \cos\alpha_2 - r_1 \cdot v_1 \cdot \cos\alpha_1) \qquad (5.6)$$

Equation (5.6) gives the required torque, so that the liquid would exit the impeller with velocity v_2 and flow rate $\dot{V}_2$.

According to the rotation laws, power is equal to the torque multiplied by the angular velocity of the rotating object as follows:

$$P = M \cdot \omega \Rightarrow$$

$$\Rightarrow P = \rho \cdot \dot{V} \cdot \omega \cdot (r_2 \cdot v_2 \cdot \cos\alpha_2 - r_1 \cdot v_1 \cdot \cos\alpha_1) \quad (5.7)$$

In case of radial input at the impeller, the angle α_1 is right angle and equation (5.7) takes the form:

$$P = \rho \cdot \dot{V} \cdot \omega \cdot r_2 \cdot v_2 \cdot \cos\alpha_2 \qquad (5.7a)$$

Equations (5.7) and (5.7a) give the (theoretical) power delivered to the liquid as it passes through the impeller.

The work received by the weight unit of liquid (theoretically delivered head) is associated to the corresponding power in the equation:

$P = \gamma \cdot \dot{V} \cdot H$ and with equation (5.7) we have:

$$H = \frac{\omega}{g} \cdot (r_2 \cdot v_2 \cdot \cos\alpha_2 - r_1 \cdot v_1 \cdot \cos\alpha_1) \quad (5.8)$$

and for $\alpha_1 = 90^o$:

$$H = \frac{\omega}{g} \cdot r_2 \cdot v_2 \cdot \cos\alpha_2 \qquad (5.8a)$$

The theoretically delivered ***total head-volume flow rate relationship***, is calculated as follows: The liquid passes through the impeller circumference area A_2 with velocity v_2. Each elementary circumference area is vertical to the radius of the impeller. Therefore, the vertical velocity component (on the area A_2) is the radial component v_{r2} and the flow rate $\dot{V}$ is given by:

$$\dot{V} = A_2 \cdot v_{r2} \Rightarrow v_{r2} = \frac{\dot{V}}{A_2} \qquad (5.8b)$$

From the velocities triangle of figure 5.3 is given:

$$v_{u2} = v_2 \cdot \cos\alpha_2 \qquad (5.8c)$$

$$\tan\beta_2 = \frac{v_{r2}}{u_{r2} - v_{r2}} \Rightarrow v_{u2} = v_2 \cdot \cos\alpha_2 = u_2 - \frac{v_{r2}}{\tan\beta_2}$$

Considering that $u_2 = \omega \cdot r_2$ and $\dot{V} = A_2 \cdot v_{r2}$ substituting in equation (5.8a) indicates that:

$$H = \frac{\omega^2 \cdot r_2^2}{g} - \frac{\omega \cdot r_2}{g \cdot A_2 \cdot \tan\beta_2} \cdot \dot{V} \qquad (5.9)$$

At constant angular velocity ω, the above equation takes the following form:

$$H = A - B \cdot \dot{V} \text{ (linear equation)}, \qquad (5.9a)$$

$$\text{where:} \quad A = \frac{\omega^2 \cdot r_2^2}{g}, \; B = \frac{\omega \cdot r_2}{g \cdot A_2 \cdot \tan\beta_2} \qquad (5.9b)$$

The slope of line $H - \dot{V}$ depends on the sign of the coefficient B, therefore the sign of the tangent of angle β_2 results that:

1) If angle β_2 is acute ($\beta_2 < 90^o \Rightarrow \tan\beta_2 > 0$), the line has a negative slope: The total head delivered decreases when the flow rate increases.

2) If angle β_2 is right ($\beta_2 = 90^o \Rightarrow 1/\tan\beta_2 = 0$), the line is horizontal: The theoretical total delivered head is independent of the flow rate.

3) If angle β_2 is obtuse ($\beta_2 > 90^o \Rightarrow \tan\beta_2 < 0$), the line has a positive slope: The total delivered head decreases with the increase of the flow rate.

As for the direction of rotation, the first circumstance corresponds to curved blades, the second to flat blades and the third to hollow blades. When the line $H - \dot{V}$ is horizontal or has a positive slope, instability of the flow in the system can be created, so almost always, ***the first shape of blades is selected (curved in the direction of rotation)***. The angle β_2 in real pumps ranges between 15° and 35°. ***The theoretical total head according to equations (5.9) and (5.9a) has a maximum value when***

the flow rate tends to zero ($H_{max} = \omega^2 \cdot r_2{}^2 / g$) ***and linearly decreases by increasing the flow rate.*** Both the maximum value and reduction rate depend on the rotation speed and the design characteristics of the impeller (r_2, A_2, β_2). The graph of equation (5.9) in a $H - \dot{V}$ diagram is illustrated in figure 5.5.

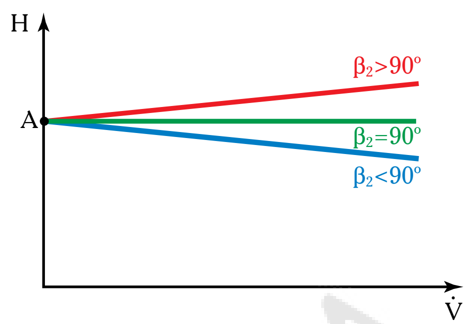


Fig. 5.5
Variation of theoretical head (H) to the flow rate ($\dot{V}$).

Similarly results the relation of theoretical delivered power to the flow rate from relations:

$$P = \gamma \cdot \dot{V} \cdot H \quad \text{and} \tag{5.9}$$

then: $$P = \omega_2 \cdot r_2^2 \cdot \rho \cdot \dot{V} - \frac{\omega \cdot r_2}{A_2 \cdot \tan\beta_2} \cdot \rho \cdot \dot{V} \tag{5.10}$$

Applying the Bernoulli equation between inlet and outlet of the impeller, we obtain the relationship between the theoretical head and the variations in pressure head, as well as the head velocity of the liquid (the difference in height and the energy losses between the inlet and the outlet of the impeller are considered negligible):

$$H = +\frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} \tag{5.11}$$

In the liquid paths from suction at the inlet of the impeller (S-1) and from the outlet of the impeller to the discharge (2-D), no additional energy value is attributed to the liquid. Applying the Bernoulli equation to these flow paths (assuming that frictions are zero and considering that the height differences are negligible) respectively results in the equations:

$$\frac{p_S - p_1}{\gamma} + \frac{v_1^2 - v_S^2}{2 \cdot g} \tag{5.11a}$$

$$\frac{p_D - p_2}{\gamma} + \frac{v_2^2 - v_D^2}{2 \cdot g} \tag{5.11b}$$

Equation (5.11a) shows that any increase in the velocity from the suction to the inlet of the impeller, leads to pressure drop (which affects the net positive suction head required - NPSHr of the pump).

Equation (5.11b) indicates that the velocity head acquired by the liquid as it is passing through the impeller is converted to a pressure head in discharge.

The above are related to the ideal circumstances of flow without frictions and an impeller with infinite number of zero-thickness vanes (assumption of perfect guidance of flow by the impeller vanes). This theoretical condition is far from the actual operating condition. But it forms the base for pump calculations. For the transition from theoretical to the actual condition, the ***theory of similarity***[3] and the experimental process are utilized. Since in the theoretical condition there is no difference between internal and given head ($H_{th} = H_D = H$), in practice the difference is critical and is one of the determinants of the quality of the pump. Also important is the diversification of power from the inlet of fluid into the pump up to its discharge.

In the next paragraphs the energy losses will be identified that lead from the theoretical to the actual flow into the pump.

5.2 Head loss or pressure drop.

The head calculated by the equation of given head-flow rate [eq. (5.9)] is the theoretical height that the pump would yield to the liquid flowing within, in ideal flow conditions. In practice, the delivered head of the pump is reduced compared to the theoretical one, due to the recirculation flow of liquid, friction and shock. This reduction is depicted in figure 5.6, in which the first line corresponding to the theoretical head is a graphical presentation of equation (5.9) for a centrifugal pump with curved blades ($\beta_2 < 90°$). However, there are parameters that reduce the theoretical head and give the total delivered head. These are discussed in the following paragraphs.

5.2.1 Flow recirculation on impeller vanes (recirculation losses).

The liquid leaves the vane in point 2 of figure 5.8

[3] For the theory of similarity (similitude), see *Μηχανική των Ρευστών* ("Fluid Mechanics") by Nikolaos Pantzalis, Athens: Eugenides Foundation publications, 2008.

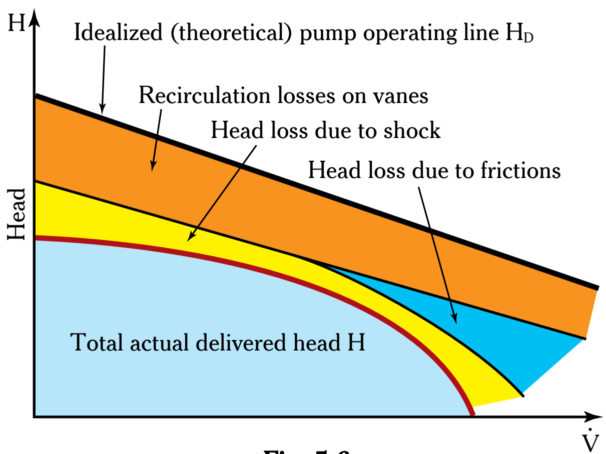


Fig. 5.6
Head loss.

under angle β_2' which in practice is smaller than β_2. This is because the assumption of perfect guidance of flow by the impeller vanes does not apply, since: The number of vanes is finite, their thickness different from zero, and the distance between two adjacent vanes significant and increasing from the center to the circumference. The elementary mass of the liquid situated in front of the rotating vane receives the thrust, and has higher pressure than the elementary mass situated behind it. Therefore, the equally spaced from the axis elementary masses of the liquid have different pressures. As the pressure is increased axially, a fluid flow circulation on the vanes occurs. This phenomenon is illustrated in figure 5.7(a). The red lines are isobaric, and the greater thickness corresponds to greater pressure. This causes a circular flow from the high to the low pressure. The result to the tangential velocities on the points with the same radius is illustrated in figure 5.7(b), (since the tangential component of velocity, $v = \omega \cdot r$, is unaffected).

Due to the liquid flow on the vanes, the output velocity of the impeller v_2' is smaller than the theoretical v_2 (fig. 5.8). But v_2' is the resultant of w_2' and u_2' which, since it depends only on the radius and the angular velocity in figure 5.8, is equated with u_2, and is given by:

$$u_2' = \omega \cdot r_2 = u_2$$

Hence, as illustrated in the modified triangle of velocities in figure 5.8, the angle β_2' is smaller than β_2.

The result of liquid flow between the vanes is the reduction of the inner head which is found from equation (5.8). It is also noted that the flow circulation does not create energy losses due to frictions, therefore does not affect the efficiency of the pump. According to equation (5.7), the required power P is reduced at a percentage corresponding to that of the reduction of the head.

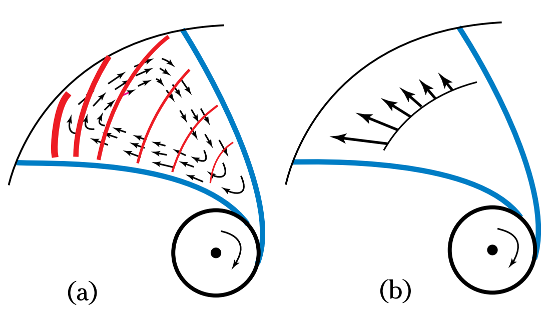


Fig. 5.7
Liquid flow between vanes.

5.2.2 Frictions on the impeller.

Part of the total head is consumed to tackle the unavoidable frictions generated by the flow of liquid through the impeller (which were zero in the previous analysis). Consequently, this part is head loss and is not attributed to the liquid passing through the pump.

Applying the Bernoulli equation between points 1 and 2 (fig. 5.2) (inlet and outlet points on the impeller) results:

$$H_D = +\frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} + h_{pf} \qquad (5.12)$$

where h_{pf} are the head losses due to hydraulic frictions within the impeller.

The flow through the passages and grooves of the pump presents friction losses proportionate to the square of the speed, and therefore, to the square of flow rate. Thus, the head loss of frictions is very important for high flow rates (fig. 5.6), and has the effect of reducing the pump performance.

5.2.3 Shock loss.

In pumps with volute casing (i.e. with spiral circular section casing progressively increasing the velocity of the liquid), the liquid that leaves the vane in the direction of the vector v_2 is introduced suddenly in a streamline that moves circumferentially between the housing and the impeller, with the result of shock induction. Increasing the angle v_2 which is the result of fluid flow on the blades (fig. 5.8), the problem exacerbates. This happens because into the surround-

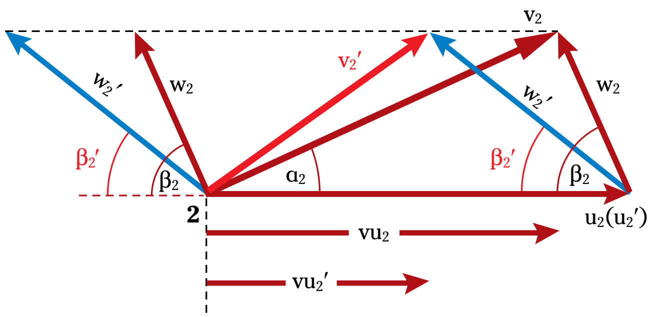


Fig. 5.8
Modification of triangle of velocities due to liquid flow on vanes.

ing volute-duct of the impeller, the fluid moves at the tangential to the impeller circumference. The sudden change in motion direction creates intense turbulence, resulting in significant energy loss, expressed in both loss head and power. These losses are called shock losses (or ***hammering losses***) (h_H).

The head that remains after the reduction due to the fluid flow to the blades, and due to the removal of the head losses (h_{pf}) within the pump, and the losses (h_H) caused by the shock upon exiting the impeller, is the ***actual total head H***. The inner theoretical height H_D [eq. (5.12)] is greater than the height that the pump delivers to the liquid flowing from it. The difference $H_D - H$ is the energy head required for the inevitable ***re-circulation*** of a proportion of the liquid, which does not leave the pump. This ensures the hydraulic completeness of the pump, namely the equal distribution of fluid pressure within it. It is caused by the difference between the internal flow rate of liquid $\dot{V}_i$ into the impeller (which is taken into account to derive the theoretical head-benefit relationship) and the actual flow rate $\dot{V}$.

In the pump design an effort is made to minimize the losses. In this effort, the number of blades, their shape, the design of the blades, the distance between the blades and housing, etc., play a decisive role. Of particular importance is the reduction of the shock losses for large high-speed pumps. These pumps are designed with ***diffusers*** which are a series of stable vanes positioned between the circumference of the impeller and the housing. These vanes direct the liquid which is leaving the impeller, changing the direction of its velocity smoothly (centrifugal pumps with diffusers, par. 3.3).

The curves that illustrate the actual or total head in function to the flow rate are the major characteristic curves of pumps, as will be seen in subsequent paragraphs.

5.2.4 Power loss.

The loss that appears in practice and in the theoretical power which the centrifugal pump transmits to the flowing liquid is equivalent to the previous energy losses. But here, the study of losses should start from the shaft power P_s, which is provided by the drive machine to the pump. In the theoretical considerations of paragraph 5.1, it was considered that the total shaft power reaches the liquid that is transferred within the pump, i.e. $\eta_m = 1$. In practice, this is not the same, but applies $\eta_m < 1$).

The power loss categories encountered are the following:

1) Bearing losses.

Regardless of the phenomena accompanying the flow inside the pump, a part of the shaft power P_s is consumed to tackle the pump's unavoidable mechanical frictions. It is the power consumption required to overcome the mechanical frictions on the bearings and seals, but also to treat disk losses. The total head is not affected by these power losses.

2) Disk losses.

The liquid that has been pumped but is not discharged from the pump flows between the outer surface of the (rotating) impeller and the (stationary) inner surface of the casing. Continuous forced movement of this fluid between the stationary housing and the rotating disk of the impeller creates major friction, and therefore, power loss. The power loss due to friction reduces the mechanical efficiency.

The rest of the power P_D (***inner hydraulic power***) is transferred from the impeller to the liquid. During the flow through the impeller, part of the inner power is consumed at different hydraulic losses. The remaining is the power ultimately delivered to the pumped liquid (P_l). The power losses in the fluid flow from the inlet to the outlet of the pump can be classified into the following categories:

1) ***Friction losses.*** This is the liquid frictions that flow inside the impeller which lead, apart from head loss, to power loss too (mechanical energy conversion into heat). The friction losses increase significantly with increasing flow rate.

2) ***Shock losses.*** These losses due to shock are generated as the fluid exits from the impeller.

3) ***Leakage losses.*** The liquid which exits the impeller and continues to the duct of the shell has

much higher pressure than the pressure in the suction of the impeller. To prevent movement of the high pressure fluid to the suction, a sealing ring (wear ring) is placed at the inlet of the impeller. However, since the impeller is rotated, the seal is not complete. A small amount of fluid inevitably is leaking to the suction and re-pumped, resulting in power loss.

Recall that, the flow circulation between the impeller vanes, although it reduces the theoretical total head, does not result in power consumption.

Figure 5.9 shows graphically the above losses. It is observed that a significant part of the shaft power is consumed to overcome them. The $\dot{V}_n$ is the ***optimum flow rate*** because in this point we have the best relationship between delivered and shaft power, i.e. the maximum efficiency. The curves of the shaft power in function with the flow rate are another group of performance curves of the pump.

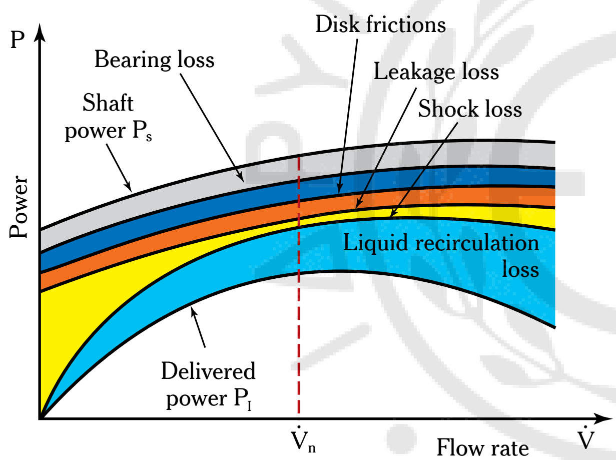


Fig. 5.9
Power loss.

5.3 Pump efficiency.

In the ideal pumping conditions of paragraph 5.1, the energy losses were absent and consequently all degrees of efficiency were 100%: $\eta_m = \eta_V = \eta_h = \eta = 1$. But in practice, there are mechanical losses ($\eta_m < 1$), internal leaks ($\eta_V < 1$), head loss of ($\eta < 1$). All this causes the consumption of a significant proportion of the shaft power and therefore a total efficiency less than one ($\eta < 1$).

The total efficiency in accordance to its definition [par. 4.5.4, eq. (4.40)], equals to the ratio of delivered power to the shaft power: $\eta = P_I / P_s$.

According to the diagram in figure 5.9, it is found that for constant angular velocity when the flow rate tends to zero, the delivered power also tends to zero, but the shaft power remains different from zero. This condition occurs when the discharge valve closes, and results in the flow rate being zero. But in that case the mechanical losses remain, since inside the pump a quantity of liquid is continuously circulated ($\dot{V}_i \neq 0$), resulting in a very significant hydraulic friction. The shaft power is then consumed to tackle the (mechanical and hydraulic) frictions and since the liquid does not enter or exit the pump in order to eliminate the heat generated by friction, the pump would overheat. In conjunction to the flow rate, the delivered power becomes zero, and, hence, the efficiency of the pump becomes zero too.

By increasing the supply (gradually opening the discharge valve), a slight increase of the shaft power is observed, while the power delivered increases much more rapidly. Hence, as the flow rate increases, the efficiency increases.

Alongside the increase in flow rate, power loss is observed due to frictions of liquid. Indeed, the rate of increase of the latter is very rapid (fig. 5.9). However there is a flow rate that depends on the pump design characteristics, in which the maximum efficiency occurs. This flow rate is the ***optimum or normal capacity of the pump*** $\dot{V}_n$. For a flow rate more than normal, the pump efficiency decreases.

The relation of efficiency to the flow rate is illustrated in the diagram of figure 5.10. The diagrams of this form are characteristic of the quality and capacity of the pump.

– *Observations on impeller calculations.*

The flow through the impeller in centrifugal (and

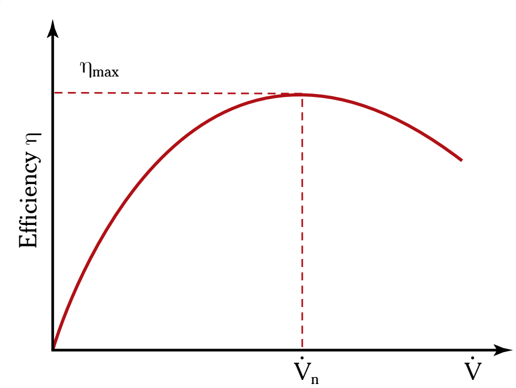


Fig. 5.10
Relation of efficiency and flow rate in a centrifugal pump.

generally in dynamic) pumps (par. 5.1) is the essential phase of the power transmission to the pumped fluid. In the calculating process new parameters are added that are associated with the impeller. Characteristic parameters of the impeller are firstly the geometric dimensions, i.e. the radius $\boldsymbol{r_1}$ and $\boldsymbol{r_2}$ and width $\boldsymbol{b}$. Also, the design elements of the impeller vanes are the angle $\boldsymbol{\beta_1}$ and, especially, the termination angle $\boldsymbol{\beta_2}$. Finally, we have the number of vanes and their thickness, the velocity of the liquid in the impeller, which in accordance to the preceding paragraphs, presents important characteristics. New, important physical qualities for the calculations are the angular velocity $\boldsymbol{\omega}$ and the rotation frequency of the impeller $\boldsymbol{n}$ (where $\boldsymbol{\omega} = 2\pi \cdot n$).

Fundamentals for the calculation of the flow in the impeller continue to be the Bernoulli principle and the continuity equation (expressing for incompressible fluids the law of conservation of energy and mass, respectively). To these are added the particular equations that were presented in the above analysis. Significant assistance in calculations offer also the velocity triangles. The Bernoulli equation is usually applied between points 1 and 2 (inlet and outlet of impeller, fig. 5.1) and the following equation results:

$$H_D = +\frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} + h_{pf}$$

The ***continuity equation*** for liquid flowing through the impeller is used in its general form:

$$\dot{V}_i = A_i \cdot v_{ri} \qquad (5.13a)$$

where: A_i is the surface through which the liquid passes, and v_{ri} is the velocity component (vertical to the flow, thus the radial component). At the outlet of the impeller, the area A_2 is equal to the length of the circumference multiplied by the width of the impeller: $A_2 = 2\pi \cdot r_2 \cdot b$. Correspondingly, the inlet surface is equal to $A_1 = 2\pi \cdot r_1 \cdot b$. Therefore, the continuity equation takes the form:

$$\dot{V}_i = 2\pi \cdot r_1 \cdot b_1 \cdot v_{r1} = 2\pi \cdot r_2 \cdot b_2 \cdot v_{r2} \qquad (5.13)$$

This equation is the flow rate of liquid in the inlet and outlet relative to the radial component of velocity. From equation (5.13) arises that if $b_1 = b_2$, the radial component of the input velocity is substantially greater than the radial component in the outlet (periphery) of the impeller:

$$r_1 \cdot v_{r1} = r_2 \cdot v_{r2} \qquad (5.13b)$$

It is recalled that if $\alpha_1 = 90^o$, then from equations (5.5a) and (5.4a) resulting:

$$v_{r1} = v_1 \quad \text{and} \quad v_{r2} = v_2 \cdot \sin \alpha_2$$

From the velocity triangles (figs 5.3 and 5.4) other relations useful in calculations are also exported. To understand and computationally utilize velocities triangles, it must be taken into account that the velocity v is sometimes analyzed as normal component w and tangential component u, while at other times as radial components v_r and tangential direction v_u:

$$\dot{v}_i = \dot{w}_i + \dot{u}_i = \dot{v}_{ri} + \vec{v}_{ui} \qquad (5.14)$$

Also from the velocities triangles, arise the trigonometric relationships between the vectors of velocity and its components. At the outlet (2) the equation applies:

$$v_2^2 = w_2^2 + u_2^2 - 2 \cdot w_2 \cdot u_2 \cdot \cos\beta_2 \qquad (5.14a)$$

and at input (1), if $\alpha_1 \approx 90^o$, then:

$$v_1 = v_{r1} \qquad (5.14b)$$

To connect the velocity to other flow dimensions, the equation (5.13) is exploited, that connects the radial component of velocity v_r with the flow rate as well as the relationship of tangential component u of the angular velocity ω:

$$u_i = \omega \cdot r_i = 2\pi \cdot n \cdot r_i \qquad (5.15)$$

5.4 Dimensional analysis and pump similarity.

According to paragraph 5.1, the theoretical flow within the impeller is studied by exploiting the laws of physics. For the transition of the theoretical to the actual flow conditions of the pumps, a range of phenomena must be considered (frictions, minor flows between rotor and casing gaps, leakages, etc.), which lead to loss of head and power. These phenomena and the corresponding losses are presented in paragraphs 5.2.2 and 5.2.3. But their complexity is such that makes it impossible to calculate the axial flow conditions by exploitation of the laws of physics, so the use of the experimental process is essential.

In order for reliable and generalized conclusions to emerge from this experimental process, a ***dimensional analysis*** of the problems and an application of the theory of similarity (or the use of affinity laws) to the pumps must precede.

The reference to the dimensional analysis and to the affinity[4] laws of the pumps aims to present, below, the basic parameters and findings that are important for understanding the flow in pumps and the flow charts provided by the manufacturers.

1) Dimensional analysis of flow in the pump.

Considering the theoretical study of the flow in the impeller of a centrifugal pump (par. 5.1) and the geometric characteristics of the impeller, we showed that the theoretical head which the pump attaches to the liquid is given by equation (5.9), and may be written according to the following equation:

$$g \cdot H = (2\pi \cdot n \cdot r_2)^2 - \frac{n}{b \cdot \tan\beta_2} \cdot \dot{V} \qquad (5.16)$$

In equation (5.16) the fact that the product of the gravitational acceleration to the total head ($g \cdot H$) corresponds to the theoretical energy delivered to fluid mass unit is taken into account. The use of the product allows reducing the variables in the dimensional analysis of the pump by one, since the acceleration of gravity is considered a variable. The theoretical energy delivered per mass unit is a function of the following physical quantities: the volume flow rate V, the rotational speed of the pump n, and the geometric characteristics of the impeller (radius r, width b, angle β), which are parameters on which depends the actual energy delivered. Additionally, the existence of head losses described in Chapter 4, and the dynamic viscosity μ (or the ***kinematic viscosity***[5] $\boldsymbol{v} = \mu / \rho$), the relative roughness of surfaces ε, and the density of the liquid ρ are introduced as important parameters (variables). Theoretical and experimental examination concludes that the power is a function of the following key parameters:

$$g \cdot H = f\,(\dot{V}, n, d, \rho, \mu, \varepsilon, l, \beta) \qquad (5.17)$$

where $\dot{V}$ is the flow rate of the pump, n the number of vanes, ρ the density of the liquid, ε the roughness of the surfaces, β the angle of the vanes and μ the dynamic viscosity.

From this relation, with the application of dimensional analysis and customization of the physical qualities that are related to the flow of a dynamic pump, taking into account the Buckingham π theorem, result the dimensionless numbers of table 5.1 for the pump.

Table 5.1: *Dimensionless numbers for a pump.*

Dimensionless number	***Specific name***
$N_h = \dfrac{g \cdot H}{n^2 \cdot d^2}$	Total head vs. diameter and speed number
$N_V = \dfrac{\dot{V}}{n \cdot d^3}$	Flow vs diameter and speed number
$N_P = \dfrac{P}{\rho \cdot n^3 \cdot d^5}$	Power number
$N_{Re} = \dfrac{n \cdot d^2 \cdot \rho}{\mu} = \dfrac{n \cdot d^2}{\nu}$	Reynolds number
$N_r = \dfrac{\varepsilon}{d}$	Relative roughness

These findings are used in the development of pump similarity laws that derive from a dimensionless analysis of three important parameters that describe pump performance (flow, total head and power). Then this greatly facilitates experimental research, while giving important conclusions that can be exploited independently.

2) Similarity or affinity of pumps.

The precondition for full compatibility between the model and the prototype, on which the study of similarity is based, is the equality of all the values of the dimensionless parameters that the dimensional

[4] Because the word "affinity" denotes an inherent similarity between things, the affinity laws are often referred to as the pump similarity laws.

[5] The term kinematic viscosity refers to one of the properties of fluids, and, in particular, the resistance present in their flow. The measurement unit of kinematic viscosity is the cSt (centistoke). Kinematic viscosity v is equal to the ratio of dynamic (absolute) viscosity μ to density ρ. Dynamic viscosity is a measure of internal resistance to flow. Dynamic (absolute) viscosity is the tangential force per unit area required to move one horizontal plane with respect to the other at unit of velocity when maintained a unit of distance apart by the fluid.

analysis has revealed. Consequently, two pumps are similar when the corresponding dimensionless numbers in table 5.1 are mutually equal. Namely, the requirements of complete similarity are:

$$N_p \equiv \frac{P}{\rho \cdot n^3 \cdot d^5} = \frac{P_m}{\rho_m \cdot n_m^3 \cdot d_m^5} \qquad (5.18a)^6$$

$$N_h \equiv \frac{g \cdot H}{n^2 \cdot d^2} = \frac{g \cdot H_m}{n_m^2 \cdot d_m^2} \qquad (5.18b)$$

$$N_{\dot{V}} \equiv \frac{\dot{V}}{n \cdot d^3} = \frac{\dot{V}_m}{n_m \cdot d_m^3} \qquad (5.18c)$$

$$N_{Re} \equiv \frac{n \cdot d^2}{\nu} = \frac{n_m \cdot d_m^2}{\nu_m} \qquad (5.18d)$$

$$N_r \equiv \frac{\varepsilon}{d} = \frac{\varepsilon_m}{d_m} \qquad (5.18e)$$

$$\frac{l}{d} = \frac{l_m}{d_m} \qquad (5.18f)$$

Therefore, if to the model and the prototype the equations apply (5.18 a-f), their behavior will, *mutatis mutandis*, be exactly the same. However, it is extremely difficult, even impossible, to achieve complete similarity between the prototype and the model. For this reason, in an effort to form a model of behaviour that is as close as possible, the following individual similarity requirements are posed, in succession.

1) ***Geometric similarity.*** A model and a prototype are geometrically similar if in both pumps each of the lengths in all three spatial coordinates is proportional. This ratio is defined as ***scale of length*** or simply scale, and is usually denoted by LL.

The geometric similarity requirements, which are the prerequisite of physical similarity, are expressed by equations (5.18e) and (5.18f). Of these, easily obtained for the corresponding linear dimensions of pumps is the equation:

$$\frac{d_m}{d} = \frac{\varepsilon_m}{\varepsilon} = \frac{l_m}{l} = LL \qquad (5.19)$$

It is recalled that, when there is geometric similarity, the corresponding angles of model and prototype are equal. For the angles of vanes, applies:

$$\frac{\alpha_m}{\alpha} = \frac{\beta_m}{\beta} = 1 \qquad (5.19a)$$

[6] The symbol ≡ means equals by definition.

Regarding the corresponding areas and volumes of model and prototype, the following relationships apply:

$$\frac{S_m}{S} = LL^2 \qquad (5.19b)$$

$$\frac{V_m}{V} = LL^3 \qquad (5.19c)$$

2) ***Kinematic similarity.*** According to the definition of kinematic similarity, ***two pumps are kinematically similar if the corresponding elementary masses*** (i.e. masses with similar characteristics) ***of the pumped liquid are in corresponding positions in corresponding times***. That is, if the geometric similarity introduces the area ratio, the kinematic similarity introduces the ratio of the transfer time of the elementary masses, since it relates to the ratios of velocity and acceleration, or the flow rate of pumped liquid. From equation (5.18c) arises the dominant requirement of kinematic similarity, which is expressed by the relationship:

$$\frac{\dot{V}}{n \cdot d^3} = \frac{\dot{V}_m}{n_m \cdot d_m^3} \qquad (5.20)$$

where is indicated that the liquid flow rate in the model and in the prototype must be equal.

Because the flow in the pump is turbulent, the achievement of kinematic similarity is combined with the requirements of dynamic similarity.

3) ***Dynamic similarity.*** Prerequisite for dynamic similarity is the geometric and kinematic similarity. The dynamic similarity introduces the ratio of forces: ***Two pumps are dynamically similar when they have ratio of linear dimensions (geometric similarity), ratio of time scales (kinematic similarity) and ratio of forces.*** During liquid pumping by a pump, inertia forces, gravitational, and frictional-viscosity forces are encountered. Dynamic similarity requirements are derived from equations (5.18b) and (5.18d).

$$\frac{g \cdot H}{n^2 \cdot d^2} = \frac{g \cdot H_m}{n_m^2 \cdot d_m^2} \quad \text{or} \quad \frac{1}{Fr} \qquad (5.21)$$

$$\frac{n \cdot d^2}{\nu} = \frac{n_m \cdot d_m^2}{\nu_m} \quad \text{or} \quad Re \qquad (5.22)$$

where Re is the Reynolds number and the Fr the Froudè number.

Note that the relation (5.18a) combines the kinematic and dynamic similarity requirements of the pumps.

Despite the degree of difficulty entailed in creating an accurate model of a pump, as already mentioned, it is relatively easy to achieve geometric similarity (LL). But it is difficult and often impossible (or disadvantageous) to use a liquid with corresponding kinematic viscosity, since the conditions of kinematic similarity are affected if the rotation speed changes.

The most reliable model of a pump is the pump itself. As paradoxical as it may sound, the obvious pronouncement that ***a pump is identical with itself with scale LL = 1***, provides findings that result from the application of previous relationships that are extremely useful. More specifically, by changing the rotational speed from n_1 to n_2, since the geometric dimensions are unchanged, the flow rate, the total head and the required power are varied. According to similarity equations (5.18c), (5.18b) and (5.18a), arise the following relations:

$$\frac{\dot{V}_1}{n_1 \cdot d^3} = \frac{\dot{V}_2}{n_2 \cdot d^3} \Rightarrow \frac{\dot{V}_1}{\dot{V}_2} = \frac{n_1}{n_2} \tag{5.23}$$

$$\frac{g \cdot H_1}{n_1^2 \cdot d^2} = \frac{g \cdot H_2}{n_2^2 \cdot d^2} \Rightarrow \frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{5.24}$$

$$\frac{P_1}{\rho \cdot n_1^3 \cdot d^5} = \frac{P_2}{\rho \cdot n_2^3 \cdot d^5} \Rightarrow \frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{5.25}$$

These refer to the case where the pump speed is changed, and the pipeline remains unchanged. The study in case of modifications in the pipeline is completely different.

5.5 Specific speed.

5.5.1 Definition.

The experimental process, as well as experience, have shown that the geometry of a rotodynamic pump primarily affects three interconnected physical qualities:

1) the angular velocity ω (or rotation frequency: $\omega = 2 \cdot \pi \cdot n$),

2) the flow rate $\dot{V}$, and

3) the total head of the pump H.

Therefore, a dimensionless number which only contains these three qualities is particularly useful. This number can be obtained by combining some of the dimensionless numbers in table 5.1. More specifically, dimensionless numbers N_h and N_V are selected, containing the above qualities and diameter of impeller d which is eliminated as follows:

$$N_V = \frac{\dot{V}}{n \cdot d^3} \Rightarrow d^3 = \frac{\dot{V}}{n \cdot N_V}$$

$$N_h = \frac{g \cdot H}{n^2 \cdot d^2} \Rightarrow d = \frac{(g \cdot H)^{0.5}}{n \cdot N_h^{0.5}} \Rightarrow$$

$$\Rightarrow d^3 = \frac{(g \cdot H)^{1.5}}{n^3 \cdot N_h^{1.5}}$$

By equating the two relations, arises that:

$$\frac{\dot{V}}{n \cdot N_V} = \frac{(g \cdot H)^{1.5}}{n^3 \cdot N_h^{1.5}} \Rightarrow n^2 \cdot \frac{\dot{V}}{(g \cdot H)^{1.5}} = \frac{N_V}{N_h^{1.5}} \Rightarrow$$

$$\Rightarrow \frac{N_Q^{0.5}}{N_h^{0.75}} = n \cdot \frac{\dot{V}^{0.5}}{(g \cdot H)^{0.75}} \tag{5.26}$$

Substituting rotational speed n with angular velocity ω, where $\omega = 2\pi \cdot n$, the following dimensionless number is obtained:

$$2\pi \cdot \frac{N_V^{0.5}}{N_h^{0.75}} = \omega \cdot \frac{\dot{V}^{0.5}}{(g \cdot H)^{0.75}} \tag{5.27}$$

The dimensionless number of equation (5.27) is an important characteristic of dynamic pumps and is defined as ***Universal specific speed*** Ω_s. The specific speed associates the angular velocity, the flow rate and total head of the pump and is determined by the relationship:

$$\Omega_s = \omega \cdot \frac{\dot{V}^{0.5}}{(g \cdot H)^{0.75}} = 2\pi \cdot n \cdot \frac{\dot{V}^{0.5}}{(g \cdot H)^{0.75}} \tag{5.28}$$

where: ω is the angular velocity of the impeller (1/s), n is the rotational speed (or frequency of rotation, rps), $\dot{V}$ the flow rate (m^3/s), H the total head of the pump (m), and g is the acceleration of gravity ($g = 9\,807\ m/s^2$).

If in a pump with specific speed Ω_s the impeller speed changes from n_1 to n_2, but the pumping system remains the same, the specific speed is not changed:

$$\Omega_{s1} = 2\pi \cdot n_1 \cdot \frac{\dot{V}_1^{0.5}}{H_1^{0.75}} \quad \text{or} \quad \Omega_{s2} = 2\pi \cdot n_2 \cdot \frac{\dot{V}_2^{0.5}}{H_2^{0.75}}$$

by simplifying arises:

$$\frac{\Omega_{s1}}{\Omega_{s2}} = \frac{n_1}{n_2} \cdot \left(\frac{\dot{V}_1}{\dot{V}_2}\right)^{0.5} \cdot \left(\frac{H_2}{H_1}\right)^{0.75}$$

Given the affinity laws for the flow rate and total head [eq. (5.23) and (5.24)], it follows that:

$$\frac{\Omega_{s1}}{\Omega_{s2}} = \frac{n_1}{n_2} \cdot \left(\frac{n_1}{n_2}\right)^{0.5} \cdot \left[\left(\frac{n_2}{n_1}\right)\right]^{0.75}$$

$$\Rightarrow \frac{\Omega_{s1}}{\Omega_{s2}} = 1 \Rightarrow \Omega_{s1} = \Omega_{s2}$$

In contrast, ***if the rotational speed of the pump is kept constant and the flow rate changes***, for example by using a valve on the discharge pipe, then the total head and ***the specific speed*** are ***altered***, as defined by the equation of the specific speed Ω_s [eq. (5.28)].

The Ω_s is defined at the ***Best Efficiency Point*** – BEP of the pump, namely a flow rate at which the pump has the maximum efficiency, or otherwise, the normal flow rate $\dot{V}_n$, then:

$$\Omega_s = 2\pi \cdot n \cdot \frac{\dot{V}_n^{0.5}}{(g \cdot H_n)^{0.75}} \qquad (5.28a)$$

From equation (5.28) is found that ***the specific speed of the pump is independent of the pumped liquid properties***.

In ***multi-stage pumps*** the specific speed refers to one stage, so for its calculation the total head of one stage is used.

Note that, ***the specific speed does not represent a rotational speed, but is a dimensionless number***. However, it has been proved extremely important, because it is ***the main standard parameter for the classification, design and comparison of dynamic pumps***.

Also, due to the fact that specific speed Ω_s is independent of the measuring system, it is defined as ***Universal Specific Speed.***

5.5.2 The specific speed in industrial practice.

Unfortunately, pump manufacturers, by 'simplifying' specific speed, and using commercial units for the measurements, have moved away from the above theoretical basis for the specific speed as dimensionless number. Although at first glance this divergence seems to be practical, it is actually difficult to understand the definition of specific speed, and incompatibility has been created, since the values depend on volumetric flow.

In the USA, the volumetric flow unit was commonly the gpm (gallons US per min). So, they set the values of flow rate in gpm, the rotational speed values in rpm, and the head values in ft. But these units, although more practical, are not compatible. Moreover, because the gravity acceleration is constant, they eliminated it from the definition type of the specific speed. For the same reason they also deleted the 2π. Thus arose:

$$N_S(US) = n(rpm) \cdot \frac{\left[\dot{V}(gpm)\right]^{0.5}}{\left[H(ft)\right]^{0.75}} \qquad (5.29a)$$

In Great Britain, a corresponding type was used, and the only difference is the flow rate units (imperial gallon per minute).

In other places in Europe, where the metric systems prevailed, the practical unit of measurement of the rotational speed was the rpm, while for flow rate the m^3/h. And here they deleted from the formula the g and 2π:

$$N_S(Metric) = n(rpm) \cdot \frac{\left[\dot{V}(m^3/h)\right]^{0.5}}{\left[H(m)\right]^{0.75}} \qquad (5.29b)$$

Therefore, the term that prevailed as "specific speed" was a variation of universal specific speed (dimensionless number Ω_s) (par. 5.5.1) in the form:

$$N_S = n \cdot \frac{(\dot{V})^{0.5}}{(H)^{0.75}} \quad \text{or} \quad N_s = \frac{n \cdot \sqrt{\dot{V}}}{(H)^{\frac{3}{4}}} \qquad (5.29)$$

In multi-stage pumps the relation (5.29) is given as:

$$N_s = \frac{n \cdot \sqrt{\dot{V}}}{\left(\frac{H}{i}\right)^{\frac{3}{4}}} \qquad (5.29c)$$

where ***i*** is the number of stages, because the total head increases on each impeller at H / i.

In a pump with double entry impeller, it is given as:

$$N_s = \frac{n \cdot \sqrt{\frac{\dot{V}}{2}}}{H^{\frac{3}{4}}} \qquad (5.29d)$$

since each side of the impeller provides the half of total flow $\dot{V}$. This specific speed is not dimensionless and given as:

$$[N_s] = [L]^{0.75} \cdot [T]^{-1.5}$$

The dimensions resulting do not have physical meaning and therefore the N_S is referred as number without units.

In order for special speed to obtain physical meaning, by exploiting the calculations of equation (5.29), the following empirical definition is given: ***Specific speed, or N_s, is defined as the speed of an ideal pump geometrically similar to the actual pump, which, when running at this speed, will raise one unit of volume, in one unit of time through one unit of head.*** It is an index of impeller design that describes the relationship between the amount of head generated by the rotation of an impeller relative to the amount of flow produced by the impeller.

The ***Hydraulic Institute***, a foundation of the largest pump manufacturers in North America which cooperates with the respective European organization ***Europump***, defines the specific speed as ***the speed at which an impeller, and consequently a pump, geometrically similar to the one under consideration, would run if it were of such a size as to deliver one unit of flow rate (i.e., 1 m³/h for EU or 1 gpm for USA) at one unit of total head (i.e., 1 m for EU, or 1 ft for USA)***.

On the basis of previous definitions of the specific speed and in an attempt to harmonize with the SI, an expression of specific speed is added in SI units, where: rotational speed is in rps, flow rate in m^3/s, head in m. Basically the following specific speeds are encountered, which differ in their values:

1) Universal specific speed: dimensionless number, Ω_s.

2) Specific speed US: (rpm, gpm, ft), N_S (US).

3) Specific speed Metric: (rpm, m^3 / h, m), N_S (M).

4) Specific speed SI: (rps, m^3 / s, m), N_S (SI).

The specific speed conversion rates are given in the following table 5.2.

By separating the dimensionless number Ω_s from the empirically determined specific speed Ns, where primarily values will be used that are produced by using SI units, the relationship between Ω_s and N_S is given as follows:

$$\Omega_s = 2 \cdot \pi \cdot N_S(SI)/g^{0.75} \Rightarrow$$
$$\Rightarrow \Omega_s = 1.13 \cdot N_S(SI) \qquad (5.30)$$

According to the Hydraulic Institute empirical definition of specific speed, if a pump's impeller rotational speed is equal to its specific speed, the pump to its optimum operating point will yield a head equal to the unit of length, and volume flow rate equal to the unit of flow. In SI:

If $n = N_S$, $H_S = 1\,m$ and $\dot{V}_S = 1\,m^3/s$, the delivered power of the pump would be:

$$P_I = \gamma \cdot \dot{V} \cdot H \Rightarrow P_I = \gamma \qquad (5.30a)$$

To express the specific speed of the pump as a function of the rotational speed, the head, and the power (N_{SP}), by replacing the flow rate in equation (5.29) it is found that:

$$N_S = n \cdot \frac{[P/(\gamma \cdot H)]^{0.5}}{(H)^{0.75}} = n \cdot \frac{P^{0.5}}{\gamma^{0.5} \cdot (H)^{1.25}}$$
$$N_{SP} = N_S \cdot \gamma^{0.5} = n \cdot \frac{P^{0.5}}{(H)^{1.25}} \qquad (5.31)$$

The number N_{SP} is defined as ***dynamic specific speed***. As observed in the definition relation (5.31), it is a variant of specific speed N_s, and is also defined as kinematic specific speed.

The dynamic specific speed is defined empirically in a corresponding manner to the specific speed (the parameter of flow rate is substituted by that of the delivered power), thus:

Dynamic specific speed N_{SP} of a pump is defined as the speed at which a geometrically similar impeller, and consequently a pump, must rotate to produce total head equal to one unit of length and the required power to be equal to the one (or given) unit of power.

Table 5.2: *Specific speed NS conversion rates.*

Conversion to →	Ω_s	N_S(SI)	N_S(M)	N_S(US)
N_{SU}	1	0.882	3 176	2 734
N_S(SI) (m, m^3/s, rps)	1,13	1	3 600	3 100
N_S(M) (m, m^3/h, rpm)	$3.15 \cdot 10^{-4}$	$2.78 \cdot 10^{-4}$	1	0.861
N_S(US) (ft, gpm, rpm)	$3.66 \cdot 10^{-4}$	$3.23 \cdot 10^{-4}$	1.162	1

for example N_S(US) = 0,861 N_S(M)
Ω_s = 1,13 N_S(SI)

It is observed that if the speed and specific weight of the pumped liquid are known, we can easily calculate the dynamic specific speed by the equation:

$$N_{SP} = N_S \cdot \sqrt{Y} \qquad (5.31a)$$

For pumped liquid water, the numerical relationship kinematic and dynamic speed in SI is:

$$N_{SP}\,(SI) = 3.13 \cdot N_S\,(SI) \qquad (5.31b)$$

The specific dynamic speed is considered as a net number although it has dimensions (of which, however, does not result physical meaning).

– ***Specific speed and pump classification.***

Specific speed is one of the main characteristic parameters of a dynamic pump, since it is related to the pump type and the shape of the impeller, and is also independent of the size of the pump and the pumped fluid.

This classification of dynamic pumps by their specific speed is illustrated in figure 5.11. It is observed that the radial flow pumps (centrifugal) have specific Ω_s speeds from about 0.17 to 1.5 [(or 500 – 4 000 N_S (US)]. The mixed flow pumps, from 1,5 to 3.2 (4 000 – 9 000, U.S.) and the axial flow pumps (propeller) greater than 3.2 (from 9 000 U.S.).

The specific speed of a dynamic pump refers to optimum operating conditions, i.e., to the maximum pump efficiency. Respectively, the degree of efficiency is affected by the specific speed of the pump (fig. 5.12).

Consequently, the economic operation of a centrifugal pump makes it necessary to avoid very low specific speed values. But this results in limitations to the capacity of the pump, since it could not respond to big heads. The increased specific speed adversely affects the permissible suction head (it decreases).

In mixed and axial flow pumps excessive values of specific speed are avoided, because, in addition to the reduced head, the efficiency is also reduced.

Specific speed affects the design of the impeller, the efficiency, the permissible head and flow rate. Therefore, it also significantly determines the pump's performance curves (Ch. 6).

Despite the fact that the definition of specific speed Ω_s was based on the similarity of dynamic pumps, it can also be used in positive displacement pumps. Considering the rotational speed of the rotor (or for reciprocating pumps, the crankshaft), which is less than that of dynamic pumps, it is logical to have less specific speed. It should be recalled that positive displacement pumps give relatively smaller flow rate and greater heads. The specific speed regions of some positive displacement pumps are illustrated in figure 5.13.

5.6 Cavitation.

An important limitation regarding each pumping system, regardless of the type of pump, is related to its suction. There, the liquid usually reaches a substantially lower pressure than that which prevails on the liquid surface in the suction tank. So, the possibility, due to evaporation, for vapors of liquid

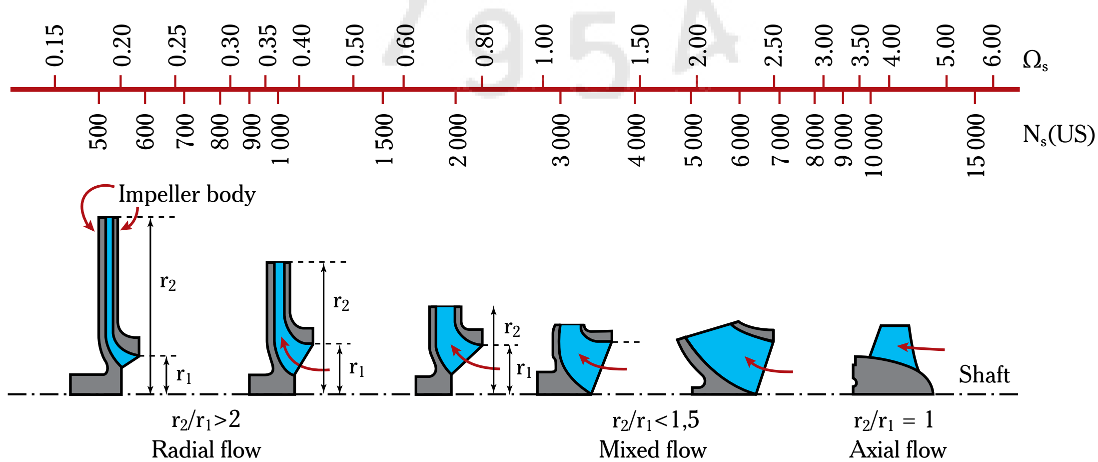


Fig. 5.11
Classification of impellers in dynamic pumps under the criterion of specific speed. (ANSI / HI 1.1-2-2.000).

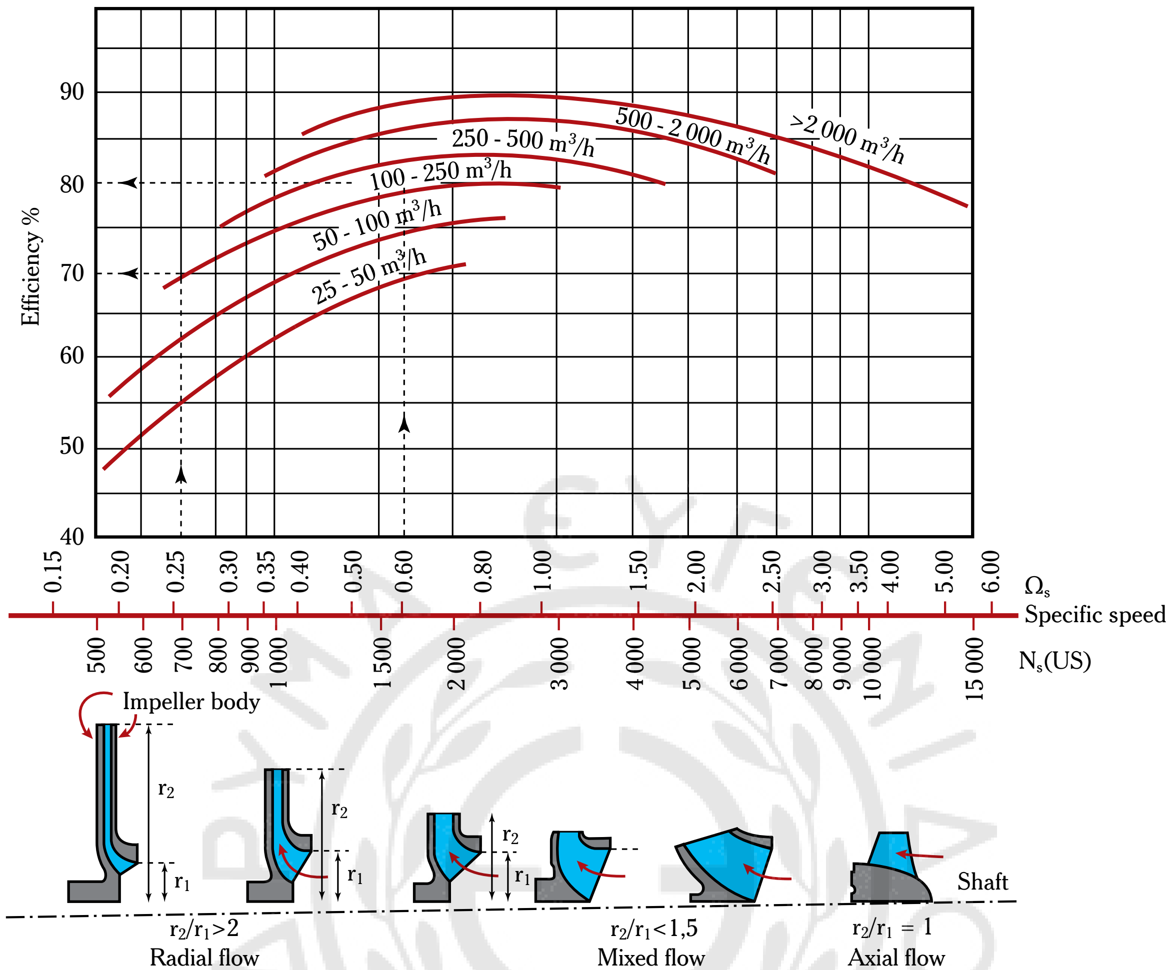


Fig. 5.12
Efficiency of centrifugal pumps vs specific speed, size, and shape.

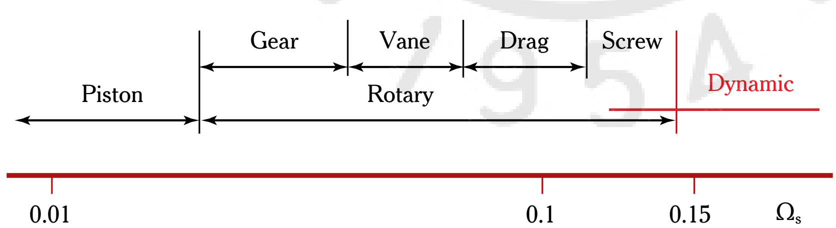


Fig. 5.13
Specific speeds of a positive displacement pump.

to be created within the suction of the pump is increased. Moreover these vapors are presented in the form of bubbles. At rotodynamic pumps these vapor pockets (bubbles) may have exceptionally undesirable effects. Also, as the liquid enters into the eye of the impeller, the pressure is reduced even more, and the vapor pockets are reinforced. Then, as the pump operates, they are transferred in areas of higher pressure where, by their sudden collapse, a significant amount of pressure is released. This pressure results in the erosion of the solid surface of the impeller. This phenomenon is defined as ***cavitation*** and leads to the rapid wear of the pump impeller.

5.6.1 Suction of dynamic pumps.

As discussed in Chapter 4, from the suction tank to the pump inlet a significant pressure drop occurs. Because in the suction tank, usually, atmospheric pressure prevails, the pressure drop in the suction pipe "approaches" the region of liquid boiling point, which is expressed by the tendency of liquid to evaporate during pumping in the particular temperature. When the fluid reaches the pump inlet, it must have energy height greater than the energy height at which it boils. The difference of these energy heights is defined as ***Net Positive Suction Head available (NPSHa)***, which was described in Chapter 4. Specifically, the energy level of the liquid in the pump inlet is:

$$H_s = z_s + \frac{p_s}{Y} + \frac{v_s^2}{2 \cdot g} \tag{5.32a}$$

To start boiling, the pressure p_v on the liquid surface should drop, so as to become equal to its evaporation tendency. So, the energy height at which boiling occurs is:

$$H_{vap} = z_s + \frac{p_{vap}}{Y} \tag{5.32b}$$

Therefore, the Net Positive Suction Head available is:

$$NPSHa = H_A - H_{vap} \quad \text{or}$$

$$NPSHa = \left(\frac{p_A}{Y} + \frac{v_s^2}{2 \cdot g}\right) - \frac{p_{vap}}{Y} = \frac{p_A - p_{vap}}{Y} + \frac{v_s^2}{2 \cdot g} \tag{5.32}$$

The Net Positive Suction Head available is a characteristic of the suction system and is independent of the pump.

As the liquid from point A (flange) enters into the pump and is driven to the entrance of the impeller, it faces many frictions (fig. 5.14). Therefore, in accordance with the Bernoulli equation, it presents a further pressure drop. Then, as it starts rotating, its speed is increased significantly and at the back of the vanes the pressure decreases further. After that, as the liquid is removed from the center of the rotating axis of the impeller (or of the propeller for axial flow pumps), the pressure increases (fig. 5.14). So, there is a maximum pressure drop, from the inlet of the pump until the minimum pressure region at the entrance of the impeller (or, more correctly, the back of the vanes and near the inlet). This pressure drop depends entirely on the design features of the pump and the suction velocity of the liquid, therefore, on the pump flow rate. Because the flow rate of a dynamic pump depends on the rotational speed (where the pipeline is not changed), the pressure drop depends on the pump design and the rotational speed.

The decline in pressure from the suction until the minimum pressure region within the impeller is defined as Net Positive Suction Head required (NPSHr) and is an important feature of the pump.

The exact expression NPSHr is obtained by applying the Bernoulli equation between the suction A and the minimum pressure region inside the pump A′:

$$\left(\frac{p_A}{Y} + \frac{v_A^2}{2g}\right) - \left(\frac{p_{A'}}{Y} + \frac{v_{A'}^2}{2g}\right) = h_{AIA'F} \Rightarrow$$

$$\text{or} \quad \left(\frac{p_A}{Y} + \frac{v_A^2}{2g}\right) - \frac{p_{A'}}{Y} \equiv NPSHr = h_{AA'f} + \frac{v_{A'}^2}{2 \cdot g} \tag{5.33}$$

Although the equation (5.33) qualitatively explains the phenomenon of pressure drop, it is computationally unprofitable. This is due to the inability to calculate the amount of head losses $h_{AA'}$ and because it is impossible to identify the precise position of the minimum pressure point A′ (it is known only that it is located in the inlet area of the impeller and the rear side of the vanes), or the velocity of liquid at this area. ***The Net Positive Suction Head required is calculated experimentally and is given by the manufacturer together with the pump.***

The experimental calculation of the pump's NPSHr is realized by placing the pump in a closed pumping system identical to that of figure 5.15. A vacuum pump gradually reduces the pressure on

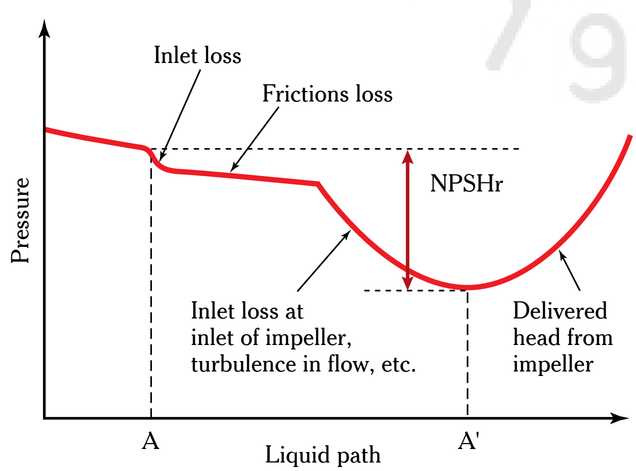


Fig. 5.14
Pressure drop at the pump inlet. A point on the inlet flange and A' point within the pump.

the free surface of liquid into a closed suction tank, therefore also of the NPSHa of the system. Once the first vapor bubbles begin to be created, the drop of total head occurs. When the drop is equal to 3% of the total height[7], the experimental procedure is stopped and the NPSH is calculated. This is considered equal to the pump's NPSHr, for which the manufacturer guarantees operation of the pump without cavitation.

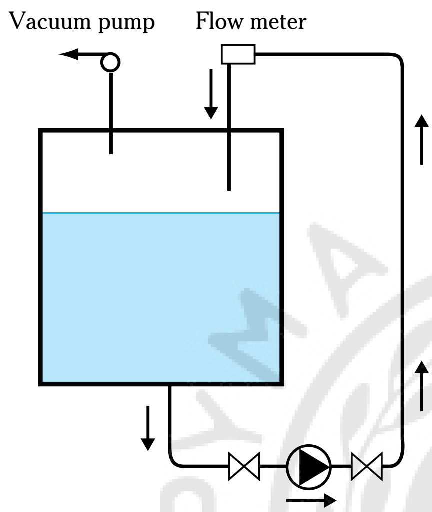


Fig. 5.15
Experimental determination of NPSHr.

To avoid the phenomenon of cavitation during the operation of a rotodynamic pump, a prerequisite is the net positive suction head available (NPSHa) to be greater than required (NPSHr):

$$\text{NPSHa} > \text{NPSHr} \tag{5.34}$$

5.6.2 Pump cavitation.

The ship's engineers must rigorously observe the NPSHa>NPSHr prerequisite. Moreover, when designing a pumping system, it must be ensured that it has sufficient net positive suction head available (if necessary, the pump might be located lower than the suction tank). When selecting a pump, the ship's engineers must ensure that the required net positive suction head is less than that of the pumping system. If this condition does not apply, then, inevitably, vapors will be produced in the liquid and, simultaneously, cavitation occurs:

$$\text{NPSHa} < \text{NPSHr} \Rightarrow \text{CAVITATION} \tag{5.35}$$

It is noted that, ***in order to avoid the phenomenon of cavitation, the available NPSHa should be kept as high as possible and in any case greater than the pump required NPSHr***, which the manufacturer provides us. The parameters that increase NPSHa are listed in table 5.6.

Table 5.6: *Changes on available NPSHa parameters.*

Parameter	*Impact on NPSHa*
1. Height loss (hs) on suction pipeline.	Increasing the height of loss reduces NPSHa.
2. Suction static head (H_{sts}).	Increasing the static suction head decreases NPSHa.
3. Pressure into suction tank (p_1).	Reducing the pressure in the suction tank reduces NPSHa. If at the suction tank prevails the atmospheric pressure, the change in atmospheric pressure affects the appearance of the phenomenon of cavitation.
4. Temperature T.	Increasing the temperature increases the evaporation tendency, thus reducing the NPSHa.

A considerable difference between the available and required NPSH, at least 15% greater NPSHa ***is recommended***. (The Hydraulic Institute, depending on the suction conditions, proposes that the ratio of available to the required NPSH ranges between 1.1 and 2.5).

$$1.1 \leq \frac{\text{NPSHa}}{\text{NPSHr}} \leq 2.5$$

The creation of gas bubbles in regions with very low pressure is shown in figure 5.16.

Regions of the impeller where low pressure is presented and are therefore susceptible to cavitation are demonstrated in figure 5.17 in red. These are primarily the entrance areas to the impeller (eye) and the areas at the rear of the vanes, about 2/3 of their length (although in cases of intense cavitation

[7] The drop of the total head by 3% is considered excessive by many designers, since the cavitation has already begun.

they can be extended up to the edges of the vanes).

The moment of collapse of a bubble of water vapors is illustrated in figure 5.18. It is a peculiar "explosion", known as implosion, where the bubble explodes inwardly. Once the bubble collapses, or more correctly implodes, it is converted into a strong ***microjet (microscopic liquid stream)*** impinging forcibly and vertically on the vanes surface (calculations raise the microjet velocity near the speed of sound, up to 400 m/s). Simultaneously, the fluid surrounding the bubble abruptly invades the volume which was previously occupied by vapors which liquefied (the saturated steam depending on temperature has a volume of 2 000 to 60 000 greater than the saturated liquid), so as to create a ***shock wave***.

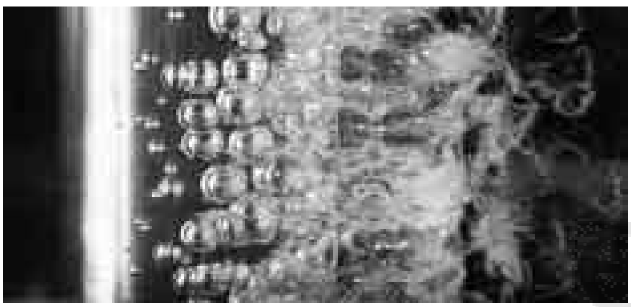


Fig. 5.16
Bubbles creation due to evaporation in low pressure regions.

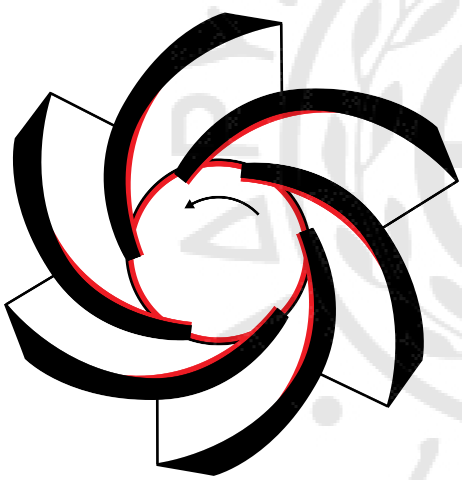


Fig. 5.17
Regions of impeller susceptible to cavitation (in red).

The most important consequence of cavitation, regards the ***erosion*** of surfaces with which gas bubbles come into contact, in other words the surfaces of the impeller vane (fig. 5.19). In order to understand the problem, we must take under consideration that the gas bubbles creation and implosion process is rapid and is repeated thousands of times on a relatively short surface distance.

Experimental research shows that the creation and collapsing process of a vapor pocket barely lasts a few milliseconds (about 3 to 4 ms). Therefore, the problem of surface erosion is not created by one bubble only but hundreds of thousands of gas bubbles which collapse "hitting" the surface with continuous microjet and shockwaves. The local pressure which is developed is big. As the speed is zeroed at the impact point of microjet, kinetic energy is converted into pressure energy. This consecutive energy release at the impact point increases the temperature of the metal. The local temperature which increases at the area of continuous strikes reaches to a few thousand degrees. The fatigue of metal is severe and its wear unavoidable. Thus, a strong cavitation process will cause damage to the vanes which will be visible in a few days, while in marginal cavitation conditions, the erosion of the vanes will only be perceived after a few months or even years. Cavitation

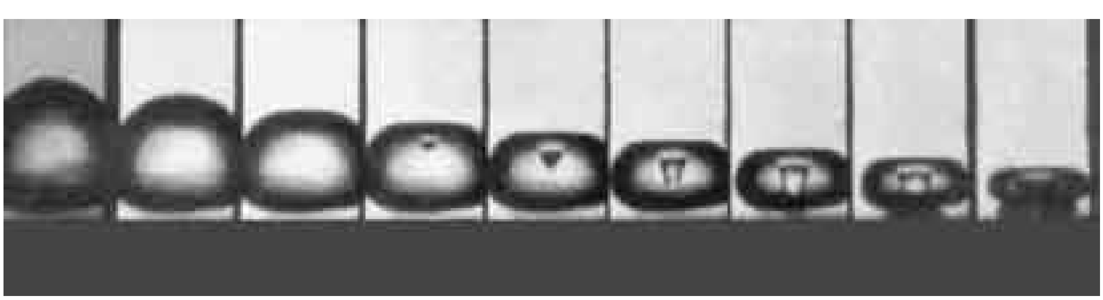


Fig. 5.18
Bubble implosion.

has a particular form, since it regards specific areas of metal surfaces, and is defined as ***cavitation-corrosion*** (fig. 5.20).

The endurance of metal to cavitation-corrosion (or mechanical corrosion) depends on its chemical synthesis and the surface smoothness. The following is a classification of certain metal alloys, in order of increasing endurance: Lead – Cast iron – Brass – Aluminum – Stainless Steel.

The ***noise*** transmitted during the bubble collapse and the pump ***vibration*** caused by the shockwaves are indicative of the creation of cavitation. Both phenomena are characteristic and allow a visual diagnosis of cavitation.

The above description relates to the main form of cavitation (or classic cavitation). There is however a ***secondary cavitation***, which is due to the recirculation of an amount of liquid at the impeller entrance (eye). In fact, in certain situations cavitation can appear even when the energy levels do not allow the development of classic cavitation as it is described in previous paragraph.

5.6.3 Special suction speed.

The high velocity of the liquid in the suction pipeline increases the amount of losses and thereby reduces the available NPSHa of the system, facilitating the phenomenon of cavitation. However, how easy or difficult it is for the phenomenon to appear, depends on the required NPSHr of the pump, which is directly related to the geometry of the impeller inlet.

An answer to the problem of forming a criterion that could indicate that a pump does not easily create cavitation conditions was first developed by L. Moody and D. Thoma (1922) by inserting the cavitation coefficient. This was defined as ***cavitation coefficient σ***, which is the ratio of net positive suc-

(a)

(b)

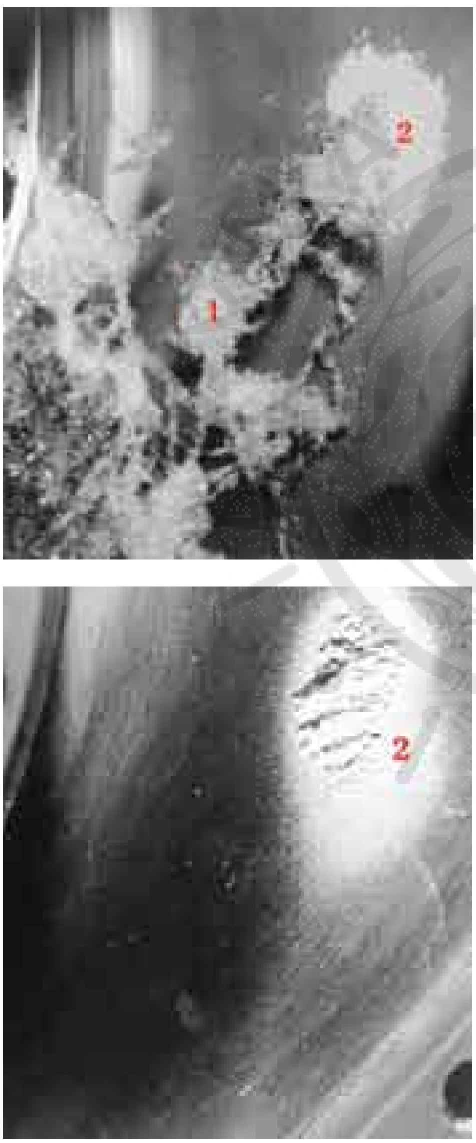


Fig. 5.19

Cavitation on impeller of a centrifugal pump, (a) creation of vapor bubbles (point 1) and collapse (point 2), and (b) cavitation wear in area 2.

Fig. 5.20

Cavitation of impeller.

tion head required to the total head of the pump:

$$\sigma = \frac{NPSHr}{H} \tag{5.36}$$

However, the cavitation coefficient proved an insufficient criterion for pump design and selection. More important was proved the criterion of ***suction specific speed of the pump*** (the meaning of which was introduced by engineer Karassik, of Worthington Co.). This is defined as the specific speed, but instead of using the total head of the pump, the Required Net Positive Suction Head (NPSHr) is used. As a dimensionless number, resulting from dimensional analysis, it is defined as ***universal suction specific speed*** – S_s :

$$N_{SSU} = 2\pi \cdot n \cdot \frac{\dot{V}_n^{0.5}}{(g \cdot NPSHr)^{0.75}} \tag{5.37}$$

However, the specific suction speed, as is the case with the specific speed, is in practice simplified and takes dimensions as follows:

$$N_{SS} = n \cdot \frac{\dot{V}_n^{0.5}}{NPSHr^{0.75}} \tag{5.38}$$

Thus arises the empirical definition of ***specific suction speed –N_{SS}–*** which is defined as the speed of an ideal pump geometrically similar to the actual pump, which when running at this speed will raise a unit of net positive suction head required, in a unit of time through a unit of head.

Specific suction speed is closely related to Specific Speed (N_s) (par. 5.5). It is an indicator of the net positive suction head required (NPSHr) for given values of capacity and also provides an assessment of a pump's susceptibility to internal recirculation.

As N_s describes an impeller design relative to head and flow, ***N_{ss} describes an impeller design relative to head and NPSHr***. Therefore, larger entry section to the vanes and smoother inlet angle improve the suction specific speed, reducing the risk of cavitation.

Usually, the specific suction speeds of pumps range between 2.2 up 4.4 (N_{SS}) or 6 000 up to 12 000 (US). Pumps with greater values have a wide impeller eye and are susceptible to cavitation due to the recirculation of the liquid, while pumps with lower values have a smaller impeller eye and are susceptible to classical cavitation.

In particular, the assessment of the tendency to cavitation of a pump system is carried out if the calculated specific suction speed available (N_{SSA}) of the pumping system is compared to the N_{SS} of the pump. The specific suction speed available is obtained from equations corresponding to equations (5.37) and (5.38) as follows:

$$S_{SA} = 2\pi \cdot n \cdot \frac{\dot{V}_n^{0.5}}{(g \cdot NPSHa)^{0.75}} \tag{5.37a}$$

$$N_{SSA} = n \cdot \frac{\dot{V}^{0.5}}{NPSHa^{0.75}} \tag{5.38a}$$

It should apply that:

$$N_{SSA} > N_{SS} \tag{5.38b}$$

Experience has shown, and this is also a recommendation of the Hydraulic Institute, that a pump must be operated with available suction specific speed equal to or less than 3.1 (N_{SSA}) or 8 500 (US). In pumps with high rotational speeds or very high flow, we encounter specific suction speed 4 (N_{SSA}) or bigger.

The graph in figure 5.21, resulting from experience and laboratory testing, illustrates the flow rate as a percentage of optimum flow at which the pump could operate according to the specific suction speed.

A solution to the limitations that are posed by the specific suction speed is the use of an inducer. An inducer is an axial impeller with a low number of vanes, which is arranged immediately upstream of the actual centrifugal pump impeller and rotates at the same rotational speed as the pump impeller.

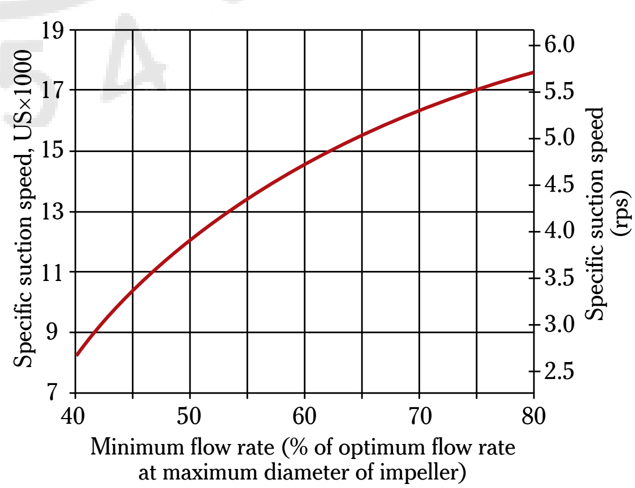


Fig. 5.21
Relationship between the minimum allowed flow rate and specific suction speed.

By this, the flow is normalized and cavitation is prevented. In such cases, the specific suction speed can reach much higher values (S_s = 7.5 or greater).

5.7 Air in the pump.

Besides evaporation (boiling) of the pumped liquid, there is also another significant cause of the creation of bubbles within the dynamic pumps, which is not associated to the vapor pressure but to the dissolved gases in the pumped liquid. According to Henry's Law, the solubility of gases in liquids decreases when the pressure is decreased (or the temperature is increased). Thus, when the pumped liquid has significant amounts of dissolved gases, solubility is low at points of decreased pressure (the pump entrance, the impeller entrance, areas rear of vanes). If the concistency of the liquid in gas is bigger than the solubility limit, the excessive amount of gases will be expelled by forming bubbles. Contrary to the vapors bubbles, gases bubbles do not implode. As they are drawn to areas of higher pressure, once again the solubility is increasing and quantities of gases are, once more, redissolved in the liquid. The size of the bubble is gradually and smoothly decreased until it totally disappears. The only result of this phenomenon is the disturbances to the flow which entail minor head losses (the solubility energy is overwhelmingly less than the evaporation energy). The phenomenon of the creation of bubbles due to the decrease of solubility is often referred to as ***gaseous cavitation***, contrary to cavitation of vapors or ***vaporous cavitation*** (par. 5.6.2).

The problem of having air or gases present within the dynamic pumps, constitutes a generic problem with significant consequences on their function. The most common sources of gases in the pumped liquid are as follows:

1) Influx of air with the pumped liquid. Especially when the suction tank is at atmospheric pressure and the static suction head is greater or the pressure is close to zero (vacuum conditions), at the pump suction there is a vacuum and this results in possible entry of air from some hole or crack into the pump (suction is not airtight).

2) Insufficient air removal from the pump casing during the starting operation of the pump, resulting in the hindering of the initial suction during the starting of pumping.

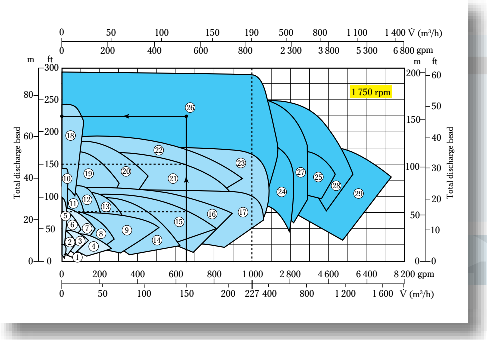
3) Significant amounts of dissolved air or other gases in combination with low pressure and high temperatures.

4) Continuous chemical processes which produce gases during pumping. These types of gas bubbles can appear in water that contains organic enzymes which might be in decomposition.

5) Entry of air in the area of low suction pressure of the impeller through a hole or slot (crack) of the housing or from the sealing points of the pump. The small amount of air in these operational conditions can possibly gather at the suction center at the impeller (eye), and since it separates the liquid flow, it can cause stopping of pumping.

When a mixture of air and liquid enters into the impeller of a centrifugal pump, the centrifugal acceleration of liquid is significantly greater than that of the gas. If the amount of gas is small, the gas diffuses in the liquid in the form of small bubbles which are drifted by the liquid and pumped with it. However, if the quantity of gases is significant, the liquid moves towards the area faster and the gases are trapped at the center of the impeller and create an air pocket. If the amount of gas is large, separation occurs resulting in the disruption of the flow, and stopping of the pumping.

The capability of rotodynamic pumps to manage incoming air relates to its geometry. More specifically, as regards the specific speed of the pump, the higher it is, the larger the amount of air that can be managed. This relates to the growing centrifugal effect as well as to the wide area of the impeller inlet. Pumps with high specific speed and wide area of impeller inlet are the combined and, mainly, the axial flow pumps (fig. 5.11). Here, the centrifugal action is limited, therefore the separation is more difficult. The presence of gases within dynamic pumps has adverse effects on its total height and its efficiency.



CHAPTER SIX
Characteristic curves

6.1 Characteristic performance curves of dynamic pumps.

The flow of liquid in dynamic pumps is quite complicated. Respectively, the relation among the pump's characteristic parameters is complicated as well, when an effort is made to express it with mathematical accuracy.

There are, for example, conditions under which the pump has to function in order to attribute great height and relatively small flow rate, or the opposite. But, under what criteria must this ***choice*** be made? Furthermore, after the previous question is answered, one must investigate the most appropriate ***location*** of the pump on a pumping system from a hydrodynamic aspect. Finally, the pumps that will be selected and installed will possibly have to face various loads (heads). So, how should their ***handling*** be in order to avoid unwanted damages?

The ***characteristic curves of pumps*** specify and depict particularly the abilities and limits of each pump and compose essential data for the estimation and operation of any kind of pumping system. Dynamic pumps' characteristic curves provide an important tool for design engineers, operators and maintenance personnel to arrive at an optimum energy design solution, or, to establish opportunities to curtail energy outgoings, as part of the operation of the pumping system.

As shown in paragraph 5.1, only in the case of the ideal friction can the relationship of characteristic qualities of the pump ($\dot{V}$, H, P, n) be (theoretically) expressed with equations. Specifically, the ***relationship of the theoretical head*** to the ***pump flow rate*** is given by the equation:

$$H_{th} = \frac{\omega^2 \cdot r_2^2}{g} - \frac{\omega \cdot r_2}{g \cdot A_2 \cdot \tan\beta_2} \cdot \dot{V}_{th} \qquad (6.1)$$

or more simply: $H_{th} = A - B \cdot \dot{V}_{th}$ (6.1a)

where: $A = \dfrac{\omega^2 \cdot r_2^2}{g}$, $B = \dfrac{\omega \cdot r_2}{g \cdot A_2 \cdot \tan\beta_2}$ (6.1b)

Accordingly, the ***relationship of theoretical power to the flow rate*** is given as:

$$P_{th} = \omega^2 \cdot r_2^2 \cdot \rho \cdot \dot{V}_{th} - \frac{\omega \cdot r_2}{A_2 \cdot \tan\beta_2} \cdot \rho \cdot \dot{V}_{th}^{\,2} \qquad (6.2)$$

or $P_{th} = \gamma \cdot A \cdot \dot{V}_{th} - \gamma \cdot B \cdot \dot{V}_{th}^2$ (6.2a)

However, the ***actual physical qualities*** are completely different. There are significant ***head losses*** (due to recirculation of flow, frictions and shocks), as well as ***power losses*** (due to frictions, shocks, leaks, disk losses and bearings). Also, the theoretical flow rate varies slightly from the inner and considerably from the actual flow rate.

But, due to the complexity of the flow inside the pump, the approach of real relationships can only be performed by dimensional analysis and experiment.

The dimensional analysis (par. 5.4) gave the following functions between the actual qualities $\dot{V}$, H, F:

$$\frac{g \cdot H}{n^2 \cdot d^2} = f\left(\frac{\dot{V}}{n \cdot d^3}, \frac{n \cdot d^2}{v}, \frac{\varepsilon}{d}, \frac{l}{d}, \beta\right) \qquad (6.3)$$

$$\frac{P}{\rho \cdot n^3 \cdot d^5} = f\left(\frac{\dot{V}}{n \cdot d^3}, \frac{n \cdot d^2}{v}, \frac{\varepsilon}{d}, \frac{l}{d}, \beta\right) \qquad (6.4)$$

The experimental process and the appropriate design, manufacture and testing of a pump are the subsequent steps. The design of the impeller is based on the specific speed Ω_s (or the N_s), which is a combination of two basic dimensionless parameters of the function (6.3):

$$\Omega_s = \frac{N_Q^{0.5}}{H_h^{0.75}}$$

where $N_h = \dfrac{g \cdot H}{n^2 \cdot d^2}$, $N_V = \dfrac{\dot{V}}{n \cdot d^3}$

The final relationships between head-flow rate, power-flow rate, efficiency-flow rate as well as net positive suction head-flow rate, are given in a form of diagrams by the manufacturer. The curves which are illustrated there are defined as characteristic curves of the pump (or ***pump characteristic performance curves***). It is noted that identical pumps or pumps with the same specific speed will have qualitatively similar diagrams, but, since this depends on the size of the pumps and their rotational speed, they will greatly differ regarding the values of qualities.

A dynamic pump is designed and constructed for reaching a particular range of flow rate and head. In this region it has the optimum correlation of values between beneficial effect and consumption, hence it has the best performance. Although the manufacturer recommends the pump's operation to be at this region, often it operates in other flow rates and heads too, because of the needs of the pumping system.

6.1.1 The curve $H = f_1(\dot{V})$ or throttling curve $H-\dot{V}$ of the pump (head vs flow rate).

According to dimensional analysis, the relationship between head and –actual– flow rate (actual capacity) is given by the function (6.3). For a given pump, where the roughness, the proportion of lengths and the angle β are specified, the variable parameters of the function are the following:

1) The volume flow rate $\dot{V}$.

2) The head H (that is attributed from the pump).

3) The rotational speed n.

4) The diameter of the impeller D (note the possibility that within the pump's casing can be placed impellers of various diameters).

5) The viscosity of the pumped fluid.

In order to reduce the variables, water is considered the pumped liquid. While the viscosity in the head-flow rate relationship is not so important, any potentially major change should be taken into consideration. The observation of the pump in rotational speed (n) and impeller diameter (D), reduces the variables to two: The total (or attributed) head and the flow rate.

Consequently, for n constant speed and D specific diameter of the impeller, the total pump head H depends on the flow rate $\dot{V}$:

For n and D constant, then:

$$H = f_1(\dot{V}) \tag{6.5}$$

The relation (6.5) is expressed by a diagram (not by an equation) where the ordinate axis is the theoretical head and the axis of abscissas the flow rate $\dot{V}$. The form illustrated is a curve and is called ***$H-\dot{V}$ or throttling curve*** (fig. 5.6). More particularly, the shape of the curve $H-\dot{V}$ depends on the specific speed of the pump N_S, the number of vanes and the shape of the casing.

There are two types of curves $H-\dot{V}$:

1) ***The stable curves $H-\dot{V}$:*** in them, when the supply is increased, the total head is reduced.

In figure 6.1a stable curves are 1 -1a and 1 - 1b. It is easily found that the maximum head is presented at zero point of flow rate:

$$\text{When} \quad \dot{V} \to 0 \quad \text{then} \quad H \to H_{max}$$

In stable curves, each head value H corresponds to one and only one flow rate value $\dot{V}$.

2) ***Unstable curves $H-\dot{V}$*** (curve 2 of figure 6.1): In this case, which occurs in certain centrifugal pumps, the maximum head H_{max} is presented for $\dot{V} \neq 0$ flow rate. For flow from 0 to $\dot{V}$ the total head is increased, while, when there is further increase, the flow rate of the head amount decreases. Hence, there is a range of head H, where two flow rates correspond, the area between points 3, 4. The pump manufacturers aim at stable curves $H-\dot{V}$ and generally avoid the curves of unstable form, because some instability is created and, essentially, unpredictable behavior in case of connection and operation of pumps in parallel.

Generally, stable curves $H-\dot{V}$ are shown in pumps with high specific speeds. Thus, the mixed and axial flow pumps have stable curves. It is more difficult to achieve a stable curve in radial flow pumps. The smaller the specific speed, the greater

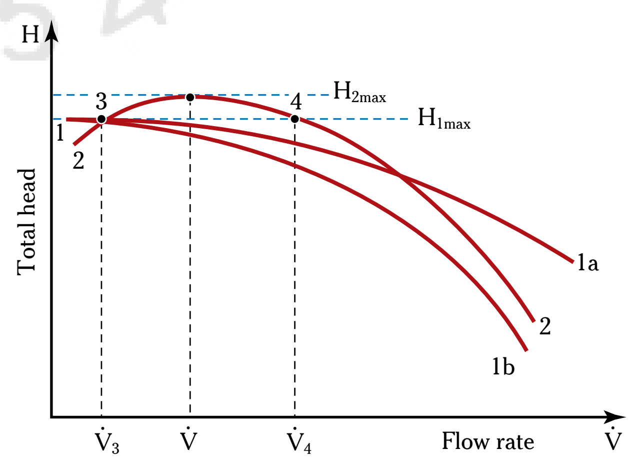


Fig. 6.1
Curves $H-\dot{V}$ of dynamic pumps.

the chance of an unstable curve. Reducing the number of vanes and the angle (β) on the outlet of the impeller are two important design methods that contribute to yielding stable curves.

The characteristic curves $H-\dot{V}$, according to their slope, are also divided to ***smooth*** and ***abrupt fall curves***. Generally, the larger the specific N_S speed, the greater the fall of the curve.

The curve of head-flow rate is the major curve of the pump and is therefore often called ***main characteristic curve***.

6.1.2 The curve $P_s = f_2(\dot{V})$ or $P_s - \dot{V}$ (shaft power vs flow rate).

For the relationship of power-flow rate, the dimensional analysis yielded the function (6.4). It is observed that the same parameters are used, which that also influence and relationship head vs flow rate. The only additional parameter is the density of the liquid. If we assume water as the pumped liquid, constant speed n, and specific diameter impeller D, then with n and D stable, arises:

$$P_s = f_2(\dot{V}) \qquad (6.6)$$

The relation (6.6) is a chart relative to ordinates axis the shaft power (P_s) and abscissa the flow rate $\dot{V}$. The types of power vs flow rate curves (one of which is encountered in fig. 5.9) are shown qualitatively in figure 6.2. We observe that in radial flow pumps, which have a small specific speed (curves 1

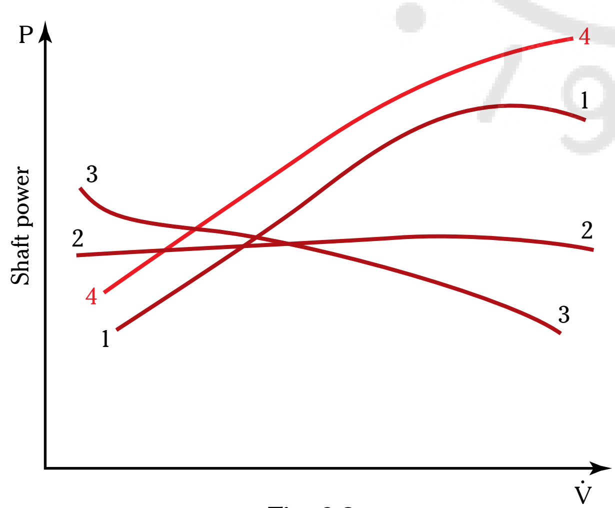


Fig. 6.2

Performance curves $P_s - \dot{V}$ of dynamic pumps:
1, 4, radial flow, 2 mixed flow, and 3 axial flow.
1, 2, 3 non-overload, 4 overload.

and 4 in fig. 6.2), the power consumption generally increases with increasing flow rate, a possible small exception being the power in high capacities. Conversely, in axial flow pumps (curve 3 in the figure) the shaft power decreases with increasing flow rate. Finally, mixed flow pumps show small changes in power with respect to the flow rate (curve 2 in the diagram).

The $P_s - \dot{V}$ curves are divided into ***overloading*** and ***non overloading*** curves. The first correspond to unstable curves $H-\dot{V}$, thus refer to certain radial-type pumps. In those, the shaft power continuously increases with increasing flow rate and there is a possibility to create overloading problems in the pump drive motor. Non-overloading curves correspond to stable curves $H-\dot{V}$, and increasing of flow rate is compensated by a significant reduction of the height, which results in reduction of the shaft power. The mixed and axial flow pumps (as well as several radial flow pumps) have non-overloading $P-\dot{V}$ curves. Curves 1, 2 and 3 in figure 6.2 are non-overloading curves, whereas 4 is an overloading curve.

6.1.3 The curve $\eta = f_3(\dot{V})$ or $\eta - \dot{V}$, efficiency vs flow rate and the Best Efficiency Point (BEP).

According to section 6.1, the delivered power to the transferred liquid is less than the shaft power that is consumed by the pump.

More specifically, a part of the shaft power is consumed to face the mechanical frictions (bearings and disk losses), another to face recirculation of liquid into the pump (shock losses and leakages) and another part to control hydraulic frictions (friction losses). Since the rotational speed and the diameter of the impeller are given, the losses due to mechanical frictions are practically independent of the flow rate, the losses due to the liquid recirculation within the pump assume large values at low flow rates, while friction losses assume large values at high flow rates (fig. 5.9). Therefore the pump efficiency ($\eta = P_I / P_s$) is directly dependent on the flow rate, and if n and D are constant, then:

$$\eta = f_3(\dot{V}) \qquad (6.7)$$

The function $\eta = f_3(\dot{V})$ is expressed by a diagram which is called ***pump characteristic perfor-***

mance curve of efficiency vs flow rate or simply ***$\eta - \dot{V}$ curve***. This diagram always has the form of figure 6.3.

The slope of the curve (how sharply it increases from 0% to n_{max} and especially how abruptly it descends for flow rate $\dot{V}>\dot{V}_n$), as well as the Best Efficiency Point, depend on the specific speed of the pump.

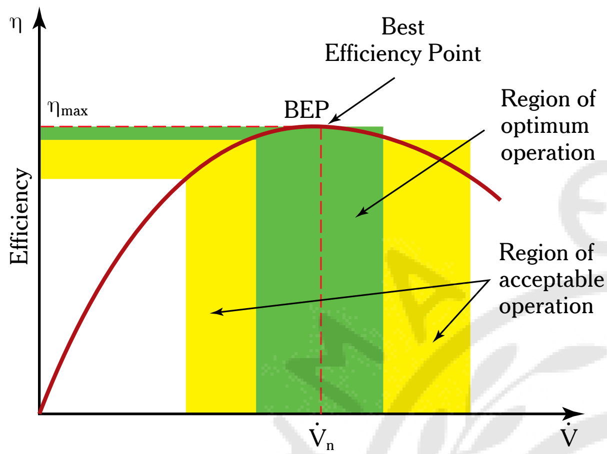


Fig. 6.3
Curve $\eta - \dot{V}$ of dynamic pumps and Best Efficiency Point.

Manufacturers, as expected, design and suggest each pump to operate at maximum flow rate, i.e. at the point of $\eta - \dot{V}$ curve corresponding to the n_{max}. This point is defined as ***Best Efficiency Point (BEP)***. The flow corresponding to BEP is defined as ***optimum*** or ***normal supply*** ($\dot{V}_n$) of the pump. Correspondingly, the height is called "normal delivery head" (H), and the power is called "normal power" (P_n).

It is observed that for slightly smaller or larger value of flow rate, a small reduction of efficiency is caused. This area is recommended as suitable for the operation of the pump and is called ***optimum operating range***. The pump operation in this region has high yield, less noise and smooth operation (vibration reduction). Typically, ***the optimum operating region is defined as the region in which the efficiency deviates to 15% of its maximum value***. Also, there is a larger area, for even smaller or greater efficiency (the yellow color in fig. 6.3), in which even if the efficiency is decreased sufficiently, the operating conditions are acceptable. Beyond this region the pump operation must be avoided because it is uneconomical, and there is a risk of serious failure.

Recall that the specific speed of the pump is calculated for the optimum point of operation.

6.1.4 NPSHr = $f_4(\dot{V})$ or NPSHr – $\dot{V}$ curve.

Last but quite important relationship that defines, in conjunction with the previous ones, the selection, placement and use of dynamic pumps, is the relationship of required net positive suction head (NPSHr) vs flow rate ($\dot{V}$). As mentioned (par. 5.6.1), the NPSHr is given by the computational unusable equation (5.33):

$$\mathrm{NPSHr} = h_{AA'f} + \frac{v_{A'}^2}{2 \cdot g}$$

But both $h_{AA'f}$ losses and the velocity $v_{A'}$ are proportional to the square of flow. Therefore, the net positive suction head required is small for small flows and sharply increases for large flows (with its increase, the cavitation risk increases too). Because both the losses and the location of point A′ depend on the chosen impeller inlet geometry, the above relationship is expressed graphically, for n and D constant, in the following function:

$$\mathrm{NPSHr} = f_4(\dot{V}) \tag{6.8}$$

The form of the NPSHr – $\dot{V}$ curve is illustrated in figure 6.4.

It is observed that in high flow rate the NPSHr increases rapidly and therefore the risk of cavitation is imminent. This area should be avoided during pump operation. Designers of pumps [by an appropriate configuration of the inlet to the pump and to the impeller, (par. 5.6.3)] ensure to obtain as small NPSHr for the BEP as possible, and generally for the area of optimum operation of the pump.

Sometimes the curves between $H - \dot{V}$ and $P - \dot{V}$ do not refer to the absolute values of these quali-

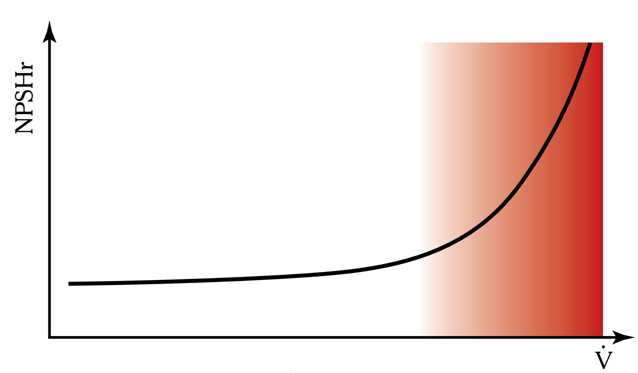


Fig. 6.4
NPSHr – $\dot{V}$ curve of dynamic pumps.

ties. They refer to relative values of the respective qualities that correspond to the maximum efficiency: H/H_n, P/P_n, $\dot{V}/\dot{V}_n$ (%).

6.1.5 Composite diagrams of curves for dynamic pumps.

The purpose of pump characteristic curves diagrams with specific geometric features and rotational speed is to provide an overall picture of the pump, monitor its operation and give the ability for calculations on the pumping system to which it is located. So, it would make no sense if for each pump the qualities are illustrated in four charts independently, since the four curves have the flow rate $\dot{V}$ as a common abscissa. Hence, the possibility of fitting the four curves in one diagram[1] is provided by using the same abscissa and different ordinates axis for each curve.

Thus is obtained the composite diagram of figure 6.5 (in which each curve corresponds to the same vertical axis values). In this diagram some significant findings for the pump may be observed. Based on the previous ***four distinct regions of flow (or capacity)*** and their respective values, are illustrated:

1) The ***optimum*** operating region bounded by the symbol III, around the BEP.

2) The ***acceptable*** operating region bounded by II_a and II_b, around the optimum region.

3) The area of low flow I, where the prolonged use of the pump must be avoided, since apart from uneconomical operation, there are risks of significant failures due to vibrations and overheating of the pump, because the small flow rate can not abduct heat.

4) Finally, the region IV, where the operation of the pump should be avoided. This is because there is an imminent risk of cavitation, due to high NPSHr values. On pumps with overloading power curve the risk of motor overload is also immediate, while the axial thrust takes high values.

However, even this graph, which provides extremely qualitative data, becomes hard to use if all

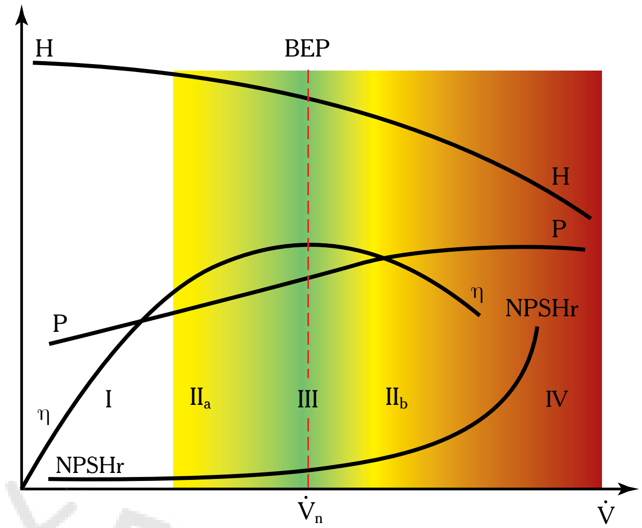


Fig. 6.5
Operating characteristic curves of a pump.

four different parameters of H, P, n, NPSHr are placed on one ordinate axis with specific values for each one. So, it is preferable to maintain the flow rate on the horizontal axis and the other curves of parameters to be placed one below the other.

ISO proposes the structure of curves to be similar to that illustrated in figure 6.6, and that there should be a parallel recording of the data of the pump, the fluid and the tests. The diagram of the figure refers to a particular pump and its description is not only qualitative but also quantitative (has values). For example, in the diagram is observed how this pump, when it operates at the BEP, gives flow rate $\dot{V} = 210\,m^3/h$, total head $H = 89\,m$, power consumption $P_s = 60\,kW$ and has efficiency ≈ 85%.

From this diagram the specific speed of the pump can be calculated. Thus, if the pump operates at 1 500 rpm, the specific speed (which is always calculated to the BEP) will be:

$$\Omega_s = 2\pi \cdot n \cdot \frac{\dot{V}_n^{0.5}}{(g \cdot H_n)^{0.75}} \Rightarrow \Omega_s = 0.236$$

Namely, it is a typical centrifugal pump, and its type can be determined by the forms of the curves (particularly the $P-\dot{V}$ curve).

It is noted that the equations encountered in the

[1] The graphs always give approximate values. Double computation of delivered power should be performed to confirm the values that we read in the diagrams. Also, the manufacturer diagrams are given for water as the pumped liquid. Therefore, when it is replaced with another liquid, the power of the diagram should be multiplied by the relative density of the liquid: $P_s = \gamma \cdot \dot{V} \cdot H/\eta$, $P_{pl} = \gamma_l \cdot \dot{V} \cdot H/\eta \Rightarrow P_{pl} = \rho_{rel} \cdot P_s$ wherein P_s the shaft pump power, P_{pl}, the shaft power for other liquid, $\gamma = \rho \cdot g$ the specific weight of water, and γ_l the specific weight of liquid different to water.

previous chapters also apply to the values obtained from the diagrams of characteristic curves. Thus, the delivered power to the fluid (P_I), for water as transferred liquid, is given:

$$P_I = \gamma \cdot \dot{V} \cdot H \quad \text{and} \quad P_I = \eta \cdot P_s$$

Such values should attribute approximately the same magnitude of delivered power, in each of the above two equations application[2], namely:

$$P_I = \gamma \cdot \dot{V} \cdot H \quad \text{or} \quad P_I = \rho \cdot g \cdot \dot{V} \cdot H$$

or $$P_I = 1\,000 \frac{kg_f}{m^3} \cdot 10 \frac{m}{s} \cdot \frac{210}{3\,600} \frac{m^3}{s} \cdot 89\,m$$

or $$P_I = 51.9\,kW$$

Example 1.

A centrifugal pump has the characteristic curves according to figure 6.6 and is used to pump liquid with relevant density ρ_l 0.9 at a flow rate 150 m^3/h.

Calculate the pump total head, the shaft and delivered power, the efficiency, and the net positive suction head required.

Solution:

Data: $\rho_l = 0.9$, $V = 150\ m^3/h$, diagrams of figure 6.6 and 6.7.

Requested: H, P_s P_I, η, NPSHr.

According to the diagram of figure 6.7 the intersection points of the known flow rate 150 m^3/h on the curves $H - \dot{V}$, $P - \dot{V}$, $\eta - \dot{V}$ and NPSHr $- \dot{V}$ are found. Then, reading the corresponding values on the scales of parameters, is given that:

$$H = 99\,m$$

$$P_s = 50\,kW$$

$$\eta = 81.5\%$$

$$NPSHr = 5\,m$$

Because the pumped fluid is not water, the value

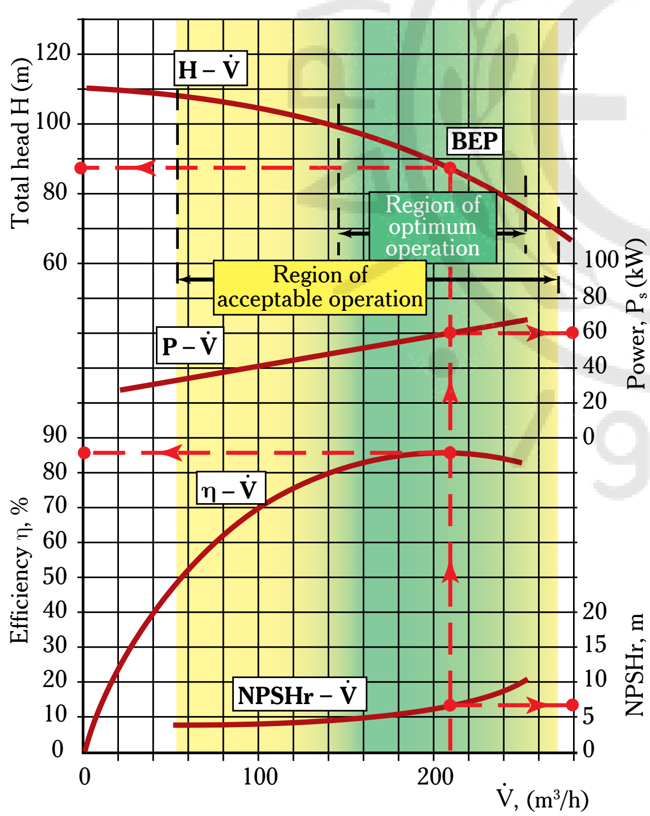


Fig. 6.6
Characteristic pump curves by ISO.

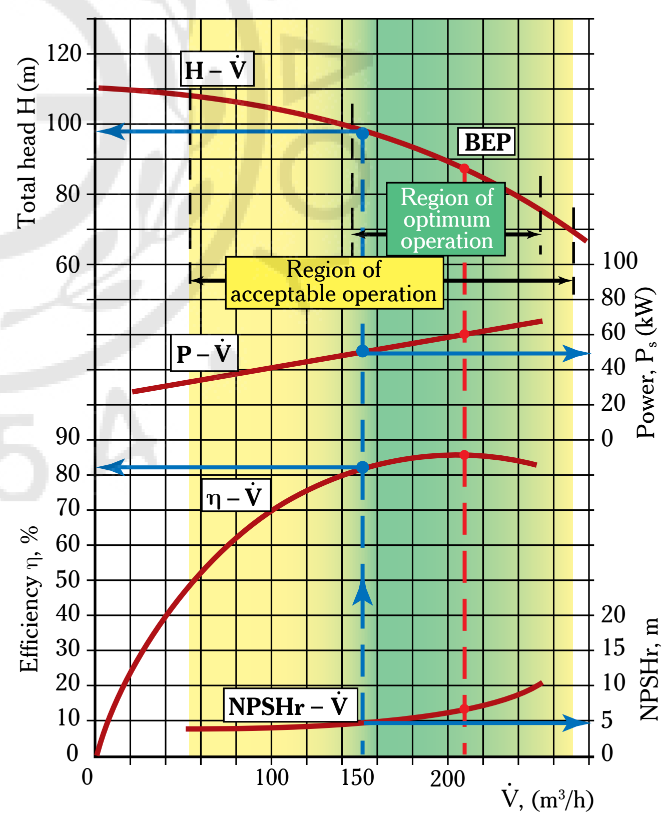


Fig. 6.7
Characteristic performance curves.

[2] $P_I = \eta \cdot P_s \Rightarrow P_I = 0.855 \cdot 60\,kW \Rightarrow P_I = 51.3\,kW$. (Also, 1 J/h = 0.0002777777778W).

of the shaft power delivered to the pumped liquid (P_l) should be corrected as follows:

$$P_l = \rho_l \cdot P_s$$

or $$P_l = \rho_l \cdot P_s = 0.9 \cdot 50\,\mathrm{kW} = 45\,\mathrm{kW}$$

The delivered power is not shown in the diagram of figure 6.7, but is calculated from the relation:

$$P_{ll} = \rho_l \cdot \gamma_{H_2O} \cdot \dot{V} \cdot H$$

where: ρ_l is the fluid density, γ the specific weight of water in N/m^3 (the specific gravity of water in SI is 9 806.65 N/m^3), $\dot{V}$ the flow rate in m^3/s, and H the total head of the pump in m. From the above comparison arises that:

$$P_{ll} = 0.9 \cdot 9\,806.65 \frac{N}{m^3} \cdot \frac{150}{3\,600} \frac{m^3}{s} \cdot 99\,m$$

or $$P_{ll} = 36\,407\,W$$

or $$P_{ll} = 36.47\,kW$$

6.2 Complete characteristic curves.

In the above analysis, the speed n and the diameter of the impeller D (for NPSHr the D of inlet) are considered constant. However, frequently, because the impeller can be replaced with an impeller of different diameter, a pump is able to operate at different rotational speeds. Thus, the functions that arise for the head, power, efficiency and NPSHr are:

$$H = f_1(\dot{V}, n, D)$$

$$P = f_2(\dot{V}, n, D)$$

$$\eta = f_3(\dot{V}, n, D)$$

$$NPSHr = f_4(\dot{V}, n, D_{inlet})$$

It is observed that each of these has four variables, consequently their graphical representation is impossible. If one of the two new variables is maintained constant, for example the diameter ***D = constant***, then the resulting functions are:

$$H = f_1(\dot{V}, n)$$

$$P = f_2(\dot{V}, n)$$

$$\eta = f_3(\dot{V}, n)$$

$$NPSHr = f_4(\dot{V}, n)$$

These functions of three variables can now be reflected in three-dimensional space with the flow rate V as base coordinate, the rotational speed (n) on the graph, and for each function the corresponding parameter. So, instead of one characteristic curve, what is taken for each function is a characteristic surface area from which the ***characteristic surface areas of pumps*** result. These are not practical since we cannot distinguish accurately the parameter values.

It is much more practical to add on the two-dimensional diagrams that have already been presented, the other curves as well, which correspond to other values of rotational speed (or diameter if the rotational speed is regarded as constant). This results in the very useful and practical ***complete characteristic curves of pumps***.

A complete characteristic curve of a pump with constant diameter of impeller and variable rotational speed is shown in the qualitative graph of figure 6.8. As expected, the delivered height and the consumed power increase by increasing the rotational speed:

$$n_1 < n_2 < n_3 < n_4 < n_5$$

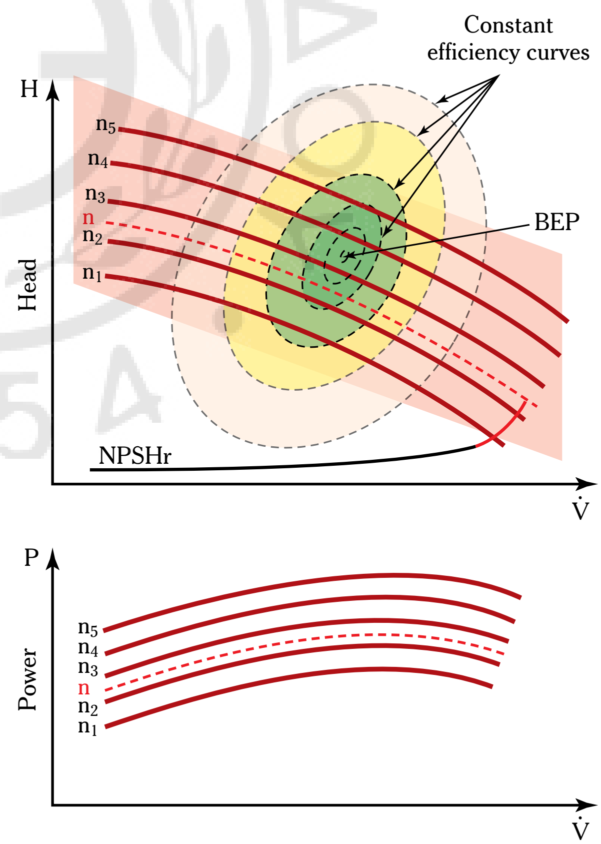


Fig. 6.8
Complete characteristic pump curves at various rotational speeds (D constant).

By observing the sequential performance levels we find that deflection from the BEP results in the operation of the pump becoming more uneconomical.

When the pump is operated at intermediate rotational speeds (n), for example $n_2 < n < n_3$, the approximate curves of head and power between the curves corresponding to the values n_2 and n_3 are taken into consideration.

Note that, at each rotational speed, the corresponding curve has its own optimum operation point.

For the transition from the simple to the complete characteristic curves, the contribution of the theory of similarity is extremely important. When changing the rotational speed or scale of lengths (LL), the pump which is taken is similar to the preceding one. According to the similarity of the pumps [par. 5.4(2)], it is known that:

$$\frac{\dot{V}_1}{n_1 \cdot D_1^3} = \frac{\dot{V}_2}{n_2 \cdot D_2^3} \Rightarrow \frac{\dot{V}_1}{\dot{V}_2} = \left(\frac{n_1}{n_2}\right) \cdot \left(\frac{D_1}{D_2}\right)^3 \tag{6.9a}$$

$$\frac{H_1}{n_1^2 \cdot D_1^2} = \frac{H_2}{n_2^2 \cdot D_2^2} \Rightarrow \frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \cdot \left(\frac{D_1}{D_2}\right)^2 \tag{6.9b}$$

$$\frac{P_1}{n_1^3 \cdot D_1^5} = \frac{P_2}{n_2^3 \cdot D_2^5} \Rightarrow \frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \cdot \left(\frac{D_1}{D_2}\right)^5 \tag{6.9c}$$

where: $D_1 / D_2 = LL$, the scale of geometric similarity.

In the cases where the diameter remains constant ($LL = 1$) and the rotational speed is altered, the similarity relations are becoming:

$$\frac{\dot{V}_1}{\dot{V}_2} = \frac{n_1}{n_2} \tag{6.10a}$$

$$\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{6.10b}$$

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{6.10c}$$

Respectively, if the scale of lengths is altered, for constant speed results:

$$\frac{\dot{V}_1}{\dot{V}_2} = \left(\frac{D_1}{D_2}\right)^3 \tag{6.11a}$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \tag{6.11b}$$

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5 \tag{6.11c}$$

The equations (6.11a, b, c) apply when changing proportionally all the dimensions of the pump (geometric similarity), not only the diameter. Therefore, they only apply approximately in cases where only the impeller is altered, adjusting the new diameter to the same eye of the impeller and leaving intact the other geometric characteristics of the pump.

This excellent feature that manufacturers provide to the pumps (defined as *back pullout pumps*) constitutes an important advantage and it is unfortunate that some pump users utilize it only for pump maintenance reasons. However, in those cases, the manufacturer will give the complete characteristic curves for constant rotational speeds and different impeller diameters. Such a (qualitative) diagram is shown in figure 6.9.

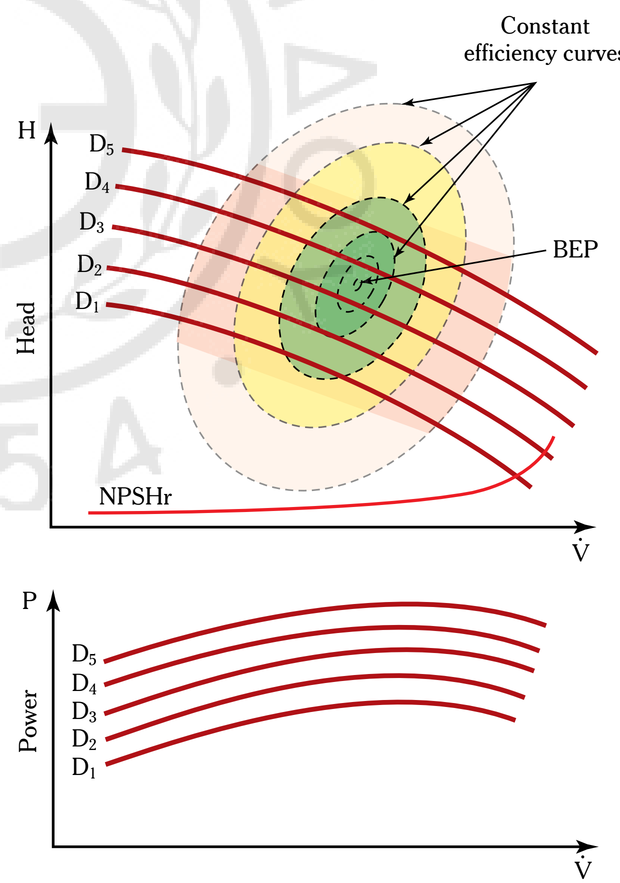


Fig. 6.9
Complete characteristic curves of a pump with different impeller diameters and constant rotational speed.

Concerning the NPSHr curve of this diagram, it is specified that it is the same for all diameters since the entry section to the impeller is not changed.

If the operating characteristics of the pump must be calculated when working at different rotational speeds, the respective parameters for the specific speed are calculated from the graph. Then, to achieve the reduction of the parameters to the speed that the pump is operating, equations (6.10a, b, c) are used.

Particular specifications for testing as well as presenting the results in charts have been published by the standardization organizations. More recent is the ISO 9906 standard (2012): Rotodynamic pumps – Hydraulic performance acceptance tests. Nevertheless there are still differences, depending on the pump type, manufacturer, and the units that are used.

The diagrams of three different manufacturers of dynamic pumps are given in figures 6.10, 6.11 and 6.12. Figure 6.10 relates to a centrifugal pump rotating at 1 500 rpm, where the diameter of the impeller can be changed. In this, it is observed that in the head - flow rate chart, apart from the efficiency curves, the power and NPSHr curves are also incorporated.

The diagram in figure 6.11 relates to a centrifugal pump with stable impeller diameter which operates at different rotational speeds (US units, form as per Hydraulic Institute). The curves of figure 6.12 refer to a mixed flow pump which rotates at 1 480 rpm and follows the specifications of ISO 9906.

The characteristic pump curves that are supplied by the manufacturer relate to smooth operation. If this is disturbed, the characteristic curves are differentiated. So, if, for example, the concentration of the pumped liquid in gases increases, the characteristic $H - \dot{V}$ decreases (smaller attributed height for the same flow rate) with similar effects to the other curves. As will be seen in paragraph 6.4.4, the curve $H - \dot{V}$ decreases sharply at the cavitation region.

Example 2.

A pump with the characteristic curves of figure 6.10 and impeller diameter 315 mm, gives flow rate of water 180 m^3/h. Calculate the head (H), the

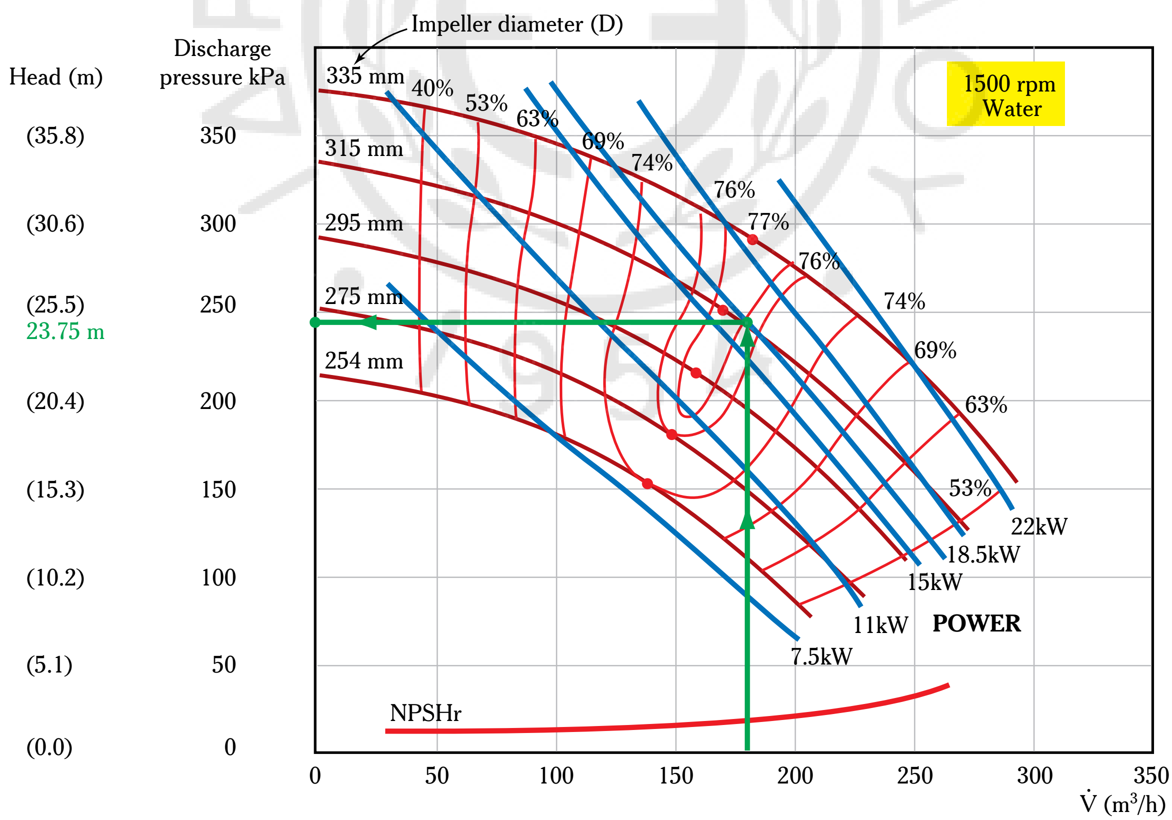


Fig. 6.10
Typical centrifugal pump curves (Armstrong Company).

shaft power (P_s), the power output (P_I) and its efficiency (η), if it is operated (a) at 1 500 rpm and (b) at 1 700 rpm, $g = 9.81\ m/s^2$.

Data: $\dot{V} = 180\ m^3/h$, the specific weight of water: $\gamma_{H_2O} = 9\,810\ N/m^3$, the diameter $D = 315$ mm, and the curves of figure 6.13.

Requested: H, P_I, P_S, n for a) $n_1 = 1\,500$ rpm and b) $n_2 = 1\,700$ rpm.

Solution:

a) From figure 6.13 wherein $n_1 = 1\,500$ rpm, $D \equiv 315$ mm and $\dot{V} = 180\ m^3/h = 0.05\ m^3/s$, is obtained the discharge pressure $p_d = 233$ kPa. Then, because applies

$$p_d = \rho \cdot g \cdot H \Rightarrow H = \frac{P_d}{\rho \cdot g}$$

or $$H = \frac{p_d}{\gamma} = \frac{233 \cdot 10^3\ Pa}{9\,810\ N/m^3} \text{ or } H = 23{,}75\ m$$

Also, from the same chart is obtained:

$$P_s = 16\ kW, \quad \eta = 0.752 \quad \text{and}$$

$$NPSHr \cdot \gamma = 20\ kPa \Rightarrow NPSHr = \frac{20\ kPa}{\gamma}$$

$$NPSHr = \frac{20 \cdot 10^3 Pa}{9\,810\ N/m^3} = 2.04\ m$$

because applies the relationship

$$\eta = \frac{P_I}{P_s} \Rightarrow P_I = \eta \cdot P_s$$

or $$P_I = 0.752 \cdot 16\ kW = 12\ kW$$

(Also is applied:

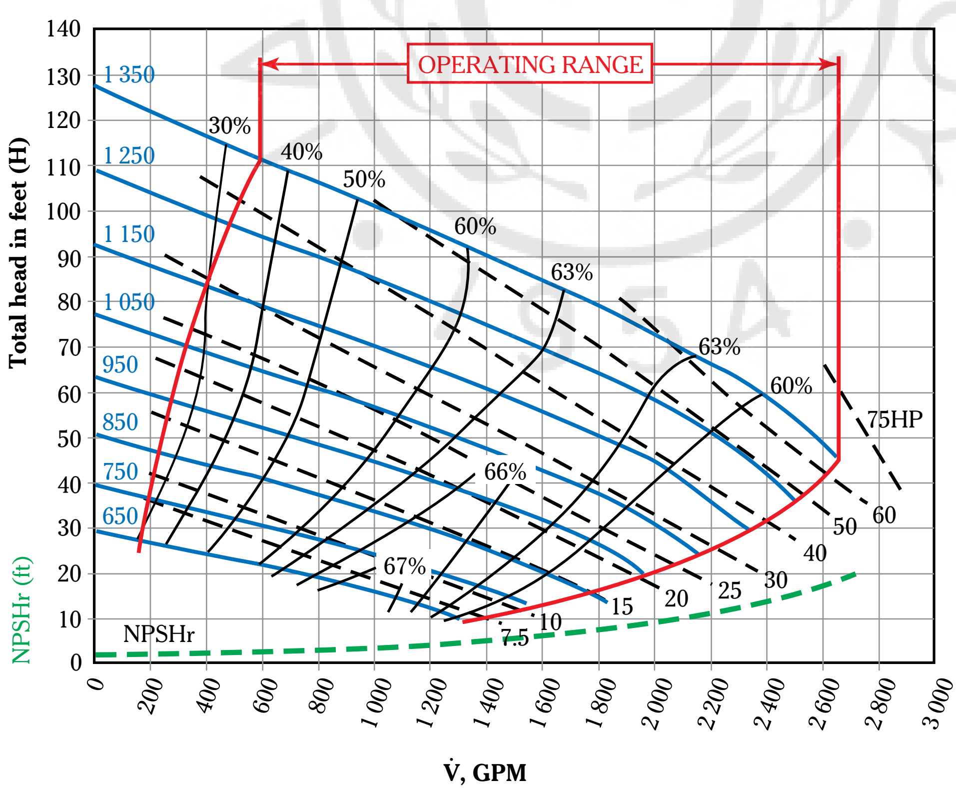
$$P_I = \gamma \cdot \dot{V} \cdot H =$$
$$= 9\,810\ N/m^3 \cdot 180/3\,600\ m^3/s \cdot 23.75\ m =$$
$$= 11.7\ kW \text{ verification of the result).}$$

Since the values of head and flow rate have more accuracy, they are selected because greater accuracy in calculation of the delivered power is achieved.

b) For the calculation of the respective parameters in the pump at $n_2 = 1\,700$ rpm the similarity equations (6.10a) are used. So applies that:

$$\frac{\dot{V}_1}{\dot{V}_2} = \frac{n_1}{n_2} \Rightarrow \dot{V}_2 = \dot{V}_1 \cdot \frac{n_2}{n_1} = 180\ m^3/h \frac{1\,700\ rpm}{1\,500\ rpm}$$

or $$\dot{V}_2 = 204\ m^3/h$$



Diameter:
Suction: 8"
Discharge: 8"
Impeller: 14.75"

Rotational speed	Suction head
rpm	feet
650	9
750	12
850	15
950	17
1 050	20
1 150	21
1 250	22
1 350	23

Fig. 6.11
Characteristic curves of a pump with constant impeller diameter.

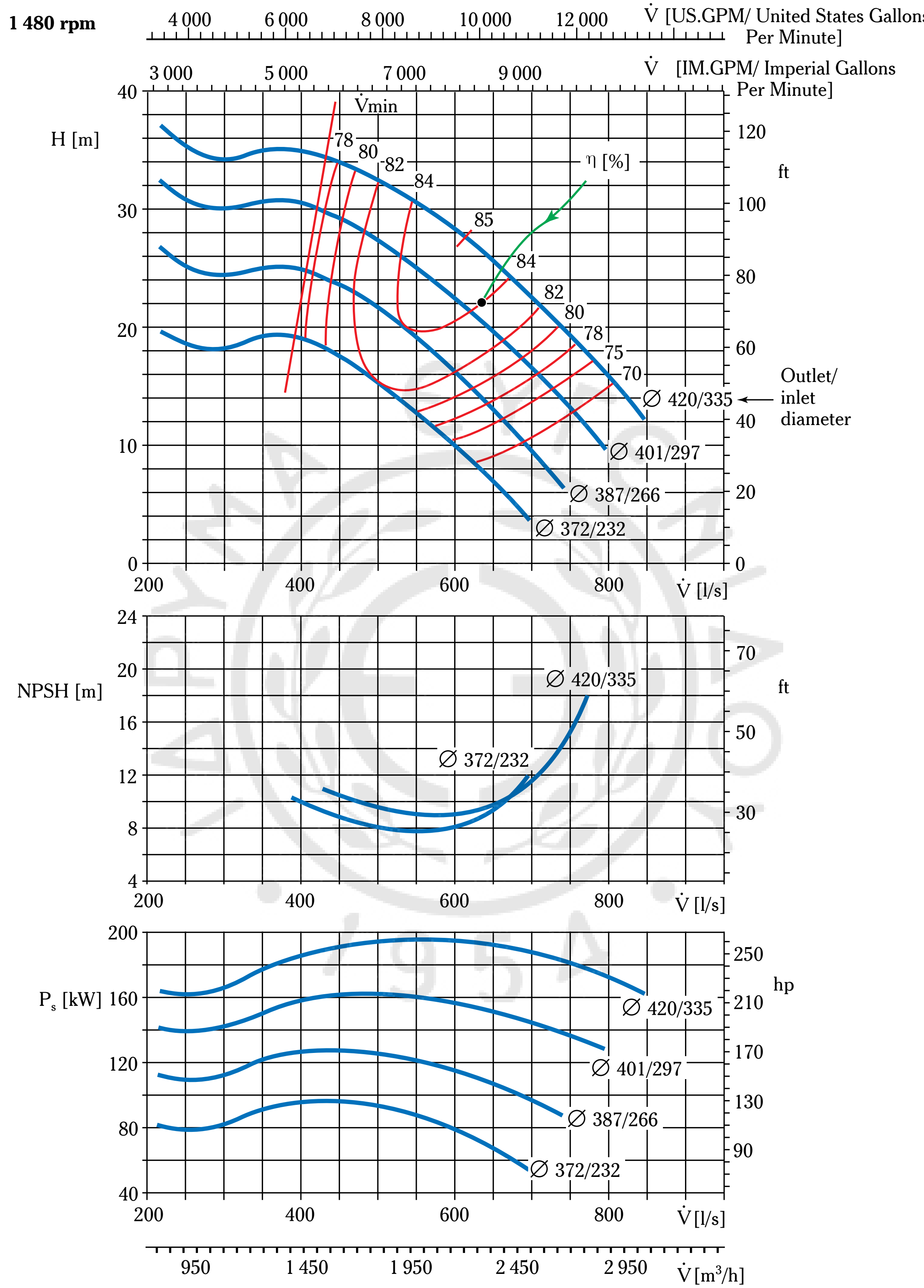


Fig. 6.12

Performance characteristic curves of mixed flow pump (centrifugal pumps of KSB co.).

From the diagram of figure 6.13 wherein for $\dot{V} = 204\,m^3/h$ and $D = 315$ mm is obtained:

$$H \cdot \gamma = P_d \quad \text{or} \quad H \cdot \gamma = 218\,kP_a$$

or
$$H = \frac{218 \cdot 10^3 P_a}{9\,810\,N/m^3} = 22.2\,m$$

Also from the same diagram is obtained:

$$P_s = 16.75\,kW, \quad \eta_2 = 74.3\% \quad \text{and}$$

$$NPSHr \cdot \gamma = 25\,kPa$$

or
$$NPSHr = \frac{25 \cdot 10^3 Pa}{9\,810\,N/m^3} = 2.5\,m$$

$$P_I = \gamma \cdot \dot{V} \cdot H =$$
$$= 9\,810\,N/m^3 \cdot \frac{204}{3\,600}\,m^3/s \cdot 22.2\,m = 12.4\,kW$$

(Because applies: $\eta = \dfrac{P_I}{P_s}$

or
$$P_s = \frac{P_I}{\eta_2} = \frac{12{,}4\,kW}{0.743} = 16.67\,kW$$

verification of previous result of P_I taken from diagram).

The exercise can be solved using the similarity relations (6.10b) and (6.10c).

– *Pump Selection Diagrams.*

Each pump has a dynamic optimum and a wider acceptable operating region (par. 6.1.5). These regions are indicated in the diagrams of figure 6.8, 6.9 and 6.11. Pump manufacturers give special attention to covering the widest possible range of needs on heads and flow rates. In order to facilitate customers in choosing the right pump for their needs, they provide head-flow rate diagrams that address a type of pumps, with a very wide range of prices. Such a diagram cannot of course be used for calculations. Its use is confined to selecting the appropriate pump. This will be accompanied by the characteristic curves. A pump selection diagram is shown in figure 6.14.

Also, the diagram may have the form of that illustrated in figure 6.15, wherein the pump operating range is displayed, so that, according to the type of its constructional characteristics, the most appropriate pump for the system requirements is selected.

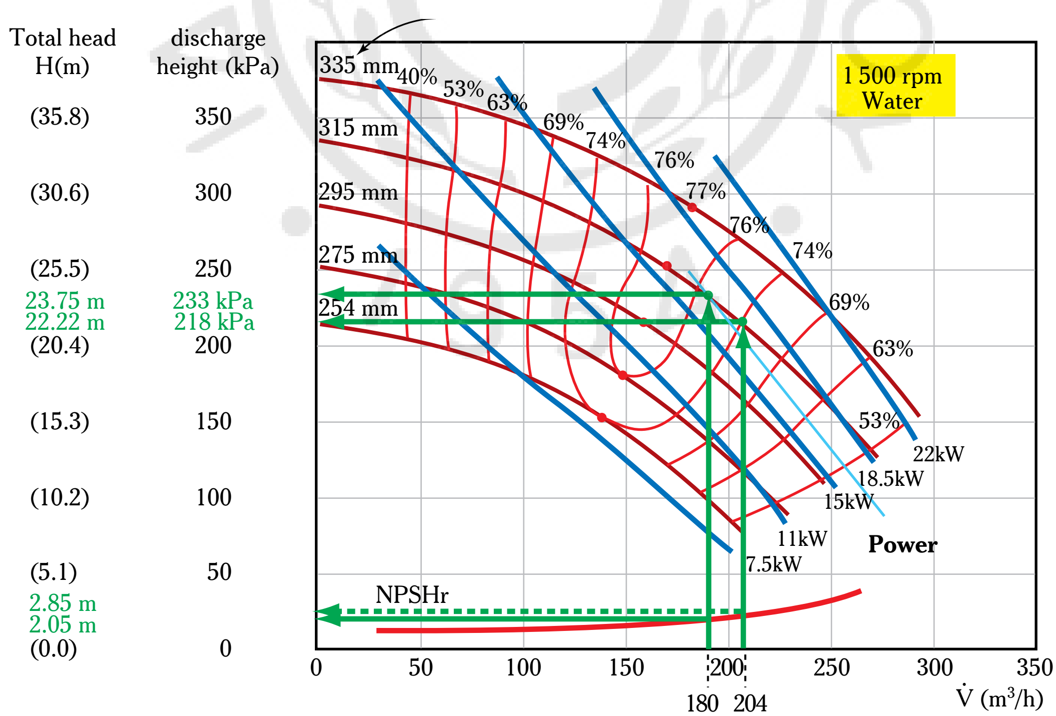


Fig. 6.13
Characteristic curves of a pump.

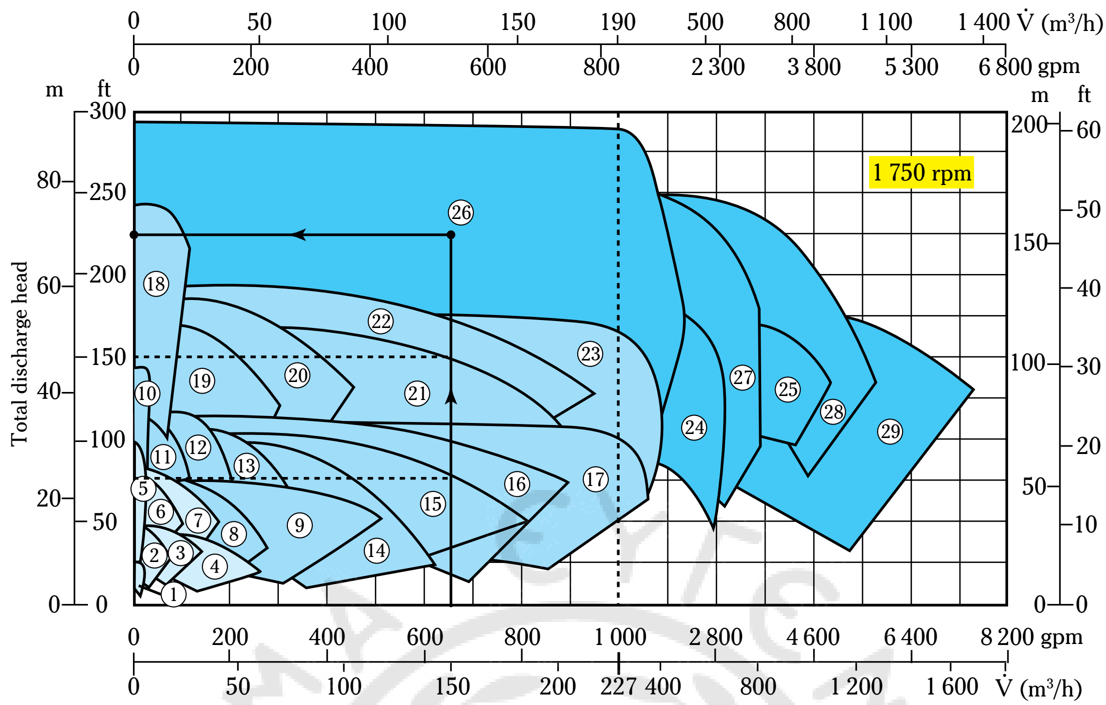


Fig. 6.14

Pump selection diagrams.

Pump model			kV300	kV350	kV400	kV450-3	kV450-4	kV500-2
Flow rate $\dot{V}$		m^3/h	1 000	1 500	2 000 2 500	3 000 3 500	4 000 4 500 5 000	5 500 6 000
Total head (H)		m	120	130	135	140	145	150
Rotational speed (n)		rpm	1 750	1 810	1 540 1 580	1 370 1 410	1 170 1 180 1 190	1 090 1 100
Shaft power (P_s)		kW	420	660	890 1 110	1 370 1 600	1 890 2 120 2 360	2 680 2 930
Impeller inlet diameter	Suction	mm	350	400	450	600	600	700
	Discharge	mm	300	350	400	450	450	500
Weight		kg_f	1 030	1 300	1 400	2 400	3 000	3 800

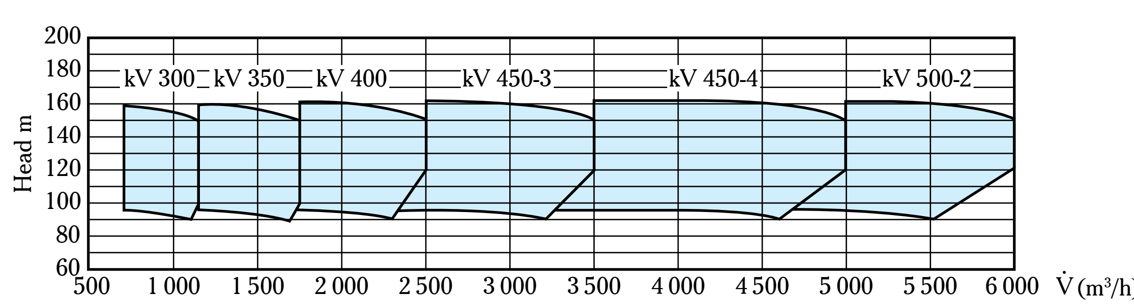


Fig. 6.15

Different models and range of pump operation.

6.3 Characteristic curves of positive displacement pumps.

A double acting reciprocating pump, as presented in Chapter 2, operating at constant speed gives actual flow $\dot{V}$, which can be calculated from the geometric characteristics of the cylinder-piston system and the number of crankshaft rotations.

According to equations (2.7) and (2.8) applies:

$$\dot{V} = \eta_V \cdot \frac{\pi \cdot (2D^2 - d^2)}{4} \cdot s \cdot n$$

If the internal leakages are inconsiderable, the flow rate is constant. Therefore the capacity depends on the geometrical characteristics of the pump and on the rotations of the crankshaft and not on height that the pump faces. This applies with sufficient accuracy if the height is relatively small.

However, when the height which the pump yields is increased, the internal leakages are inevitable, the volumetric efficiency η_V decreases, and therefore the flow rate is reduced.

The rotary positive displacement pumps have a similar behavior, with a slightly larger bending of curve $H-\dot{V}$. Therefore the relationship of height-flow rate can qualitatively be displayed for all positive displacement pumps as shown in diagram of figure 6.16. Typical charts of characteristic curves of power and flow rate with respect to the speed of a gear pump are shown in figure 6.17.

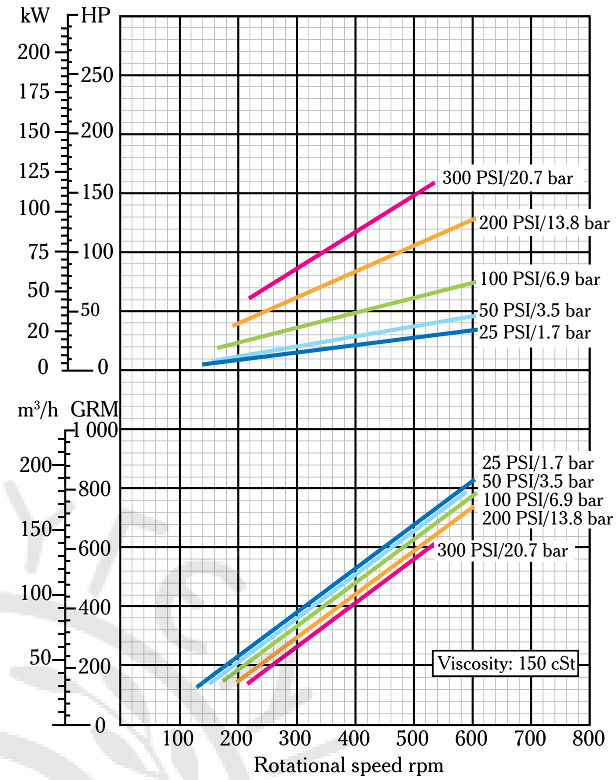


Fig. 6.17

Performance curves for a gear pump showing the pressure change as a function of speed, flow rate and power.

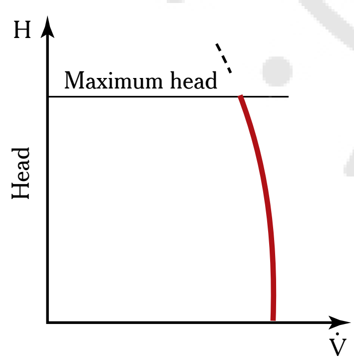


Fig. 6.16

Typical curve $H-\dot{V}$ of positive displacement pump.

6.4 Performance curves of a pumping system.

The characteristic curves of the pumps, in particular the curve $H-\dot{V}$, provide the capability of pumps. A specific pump, depending on the conditions, will operate at a point which lies on the curves that characterize it. Since a pump with a certain diameter of the impeller operates at a specific number of rotations, its operating point will depend on the system where it is positioned.

6.4.1 Pumping system curve.

The pump is always placed in a pumping system and provides the system with the required energy for pumping the liquid. Thus it has to face the hydraulic losses in the suction and discharge pipe, as well as the increase of dynamic, kinetic and pressure energy of the liquid (in short, the energy requirements of the pumping system). The total energy height required by the system, which the pump delivers (par. 4.3.2), is calculated from the energy equation [eq (4.17)] (also known as extended Bernoulli equation for pumps).

$$H = H_{st} + \frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} + \Sigma h_f$$

A given pumping system is considered as known when we have the following data:

1) The qualities of the suction and discharge

pipes (lengths L, diameters d, relative roughness ε, local loss factor K).

2) The total static head $H_{st} = z_2 - z_1$

3) The prevailing pressures on liquid surface p_1 and p_2 in suction and discharge side respectively.

Once these elements are known, the energy equation can take the following form:

$$H = a + b \cdot \dot{V}^2 \qquad (6.12)$$

wherein: ***static coefficient (a)*** is given as:

$$a = H_{st} + \frac{p_2 - p_1}{\gamma} \qquad (6.12a)$$

and the ***dynamic coefficient (b)*** is given as:

$$b = \frac{8}{\pi^2 g}\left[\left(f_s \frac{L_s}{d_s} + K_s\right)\cdot\frac{1}{d_s^4} + \left(f_d \frac{L_d}{d_d} + K_d\right)\cdot\frac{1}{d_d^4} + \frac{1}{d_2^4}\right] \qquad (6.12b)$$

[If there is a discharge tank $d_2 \to \infty$, then $(1/d_2^4) = 0$].

The coefficients (a) and (b) are constant and dependent on the data of the pumping system[3]. Specifically, (b) varies a little at small flow rates, due to change of the friction factors (of suction f_s and discharge f_d), while it is stabilized for relatively high flow rates. These small changes have no practical value. The friction coefficients f_s and f_d can be calculated for fully developed turbulent flow.

Equation (6.12) is a function of second derivative of flow rate to the head. In the orthogonal coordinate system it is represented by the curve of figure 6.18 where abscissa is the flow rate and ordinate the head. This curve is defined as ***characteristic system curve*** (or pipeline) since it depends exclusively on the pumping system and is not related to the pump. Considering equation (6.12), it is shown that the characteristic system curve has a ***minimum value*** (a) equal to the sum of the static head and pressure head of pipeline.

The ***slope of the system curve*** depends on the value of the coefficient b. The larger it is, the more steeply the curve is rising, and hence rise the energy requirements of the system that the pump has to face. The value of coefficient b depends mainly on the diameter of discharge, but also on the length of the discharge piping system and the local loss coefficients therein (e.g. increasing the roughness inside the pipe surface by scale formation in the pipelines of a ship). Commonly, the contribution of the suction tube to the value of b is negligible, because suction is designed to have as small energy requirements as possible in order to prevent cavitation. In a given pumping system the value of coefficient (b) can be increased –hence the slope of the curve– by closing partially the valve on the discharge side of the pump [i.e., increasing the local losses (K_D) in the discharge].

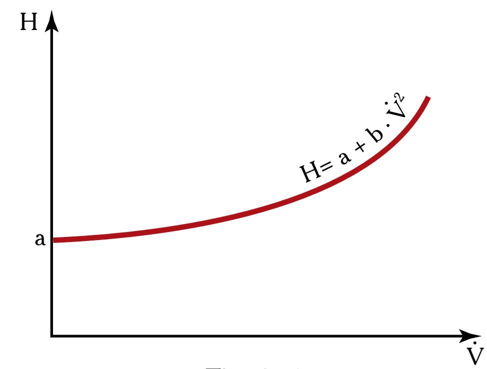


Fig. 6.18
The System Curve (where the system head is visualized).

Example 3.

The pipes of the pumping system in figure 6.19 are made of cast iron and the pipe diameters in suction and discharge are $d_s = 12$ cm and $d_d = 14$ cm. The length of the network is $L_s = 2$ m and $L_d = 65$ m, while the values of friction factors are given as $f_s = 0.0230$ and $f_d = 0.0252$. The local loss coefficient in suction is given as $K_S = 0.8$. Also, provided that the pressures on free surfaces of liquid are $p_1 = 101.3$ kPa, $p_2 = 150$ kPa, and the static head as illustrated in figure 6.19 is $H_{st} = (-1 + 15 + 1) = 15$ m, design the characteristic curve of the pipeline if (a) the discharge valve is fully open and K_D=6, (b) the discharge valve is partially open with K_D=10.

Solution:

Considerations: To plot the characteristic curve

[3] More analysis on this is found in the book *Μηχανική των Ρευστών*, ("Fluid Mechanics") by N. Pantzalis, Athens: Eugenides Foundation publications, 2008.

of the system, the pairs of values (H, $\dot{V}$) must be calculated. The required pairs (H, $\dot{V}$) are at least three, but the more are the calculated pair values the more accurately designed the curve. The typical analytical procedure is as follows:

1) Give values in flow rate $\dot{V}$.

2) Calculate the exact values of friction factors[4] f_s and f_d (for the example here are provided as $f_s = 0.0239$ and $f_d = 0.0252$). For laminar flow, the friction factor is given by relationship $f = 64/Re$.

3) Calculate the coefficients a and b utilizing equations (6.12a) and (6.12b).

4) Then, by placing values in flow rate, the respective values of total head are obtained.

According to the previous considerations:

a) For $K_S = 0.8$ and $K_D = 6$

$$H = H_{st} + \frac{p_2 - p_1}{Y} + \frac{v_2^2 - v_1^2}{2 \cdot g} + (h_s + h_d)$$

or
$$H = a + b\dot{V}^2 \qquad (1)$$

$$a = H_{st} + \frac{p_2 - p_1}{Y},$$

by applying the values resulting: a = 19.96 m

$$b = \frac{8}{\pi^2 g}\left[\left(f_s \frac{L_s}{d_s} + K_s\right)\cdot\frac{1}{d_s^4} + \left(f_d \frac{L_d}{d_d} + K_d\right)\cdot\frac{1}{d_d^4}\right] \qquad (2)$$

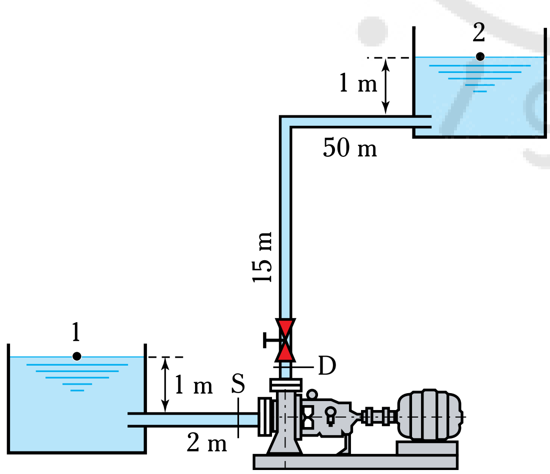


Fig. 6.19

From the data is calculated:

$$b = 20\,782\ (s^2/m^5)$$

And from the relation (1)

$$\Rightarrow H = 19.96 + 20\,782 \cdot \dot{V}^2.$$

For each value of $\dot{V}$, the relation (1) gives the corresponding values of H.

For example: If $\dot{V} = 100\ m^3/h = 100/3\,600\ m^3/s$, results H = 36 m.

Then a table of pairs (as shown in tab. 6.1) is established, and the pair are plotted in a diagram $H-\dot{V}$. The characteristic curve (a) resulting is the characteristic curve of the system and is illustrated in figure 6.19.

Table 6.1: *$H-\dot{V}$.*

$\dot{V}$ m^3/h	(a)H (m)	(b) H (m)
0	19.96	19.96
50	23.97	25.98
100	36.00	44.03
150	56.04	74.12
200	84.10	116.24
250	120.18	170.40
300	164.28	236.59

b) For $K_S = 0.8$ and $K_D = 10$

Now the only change compared to the previous calculations is the value of the coefficient b, which increases due to change in local losses:

By calculations $b = 31.194\ (s^2/m^5)$ and from (1)

$$(1) \Rightarrow H = 19.96 + 31\,194 \cdot \dot{V}^2 \quad (SI) \qquad (3)$$

By the new pair of values ($\dot{V}$, H) the column (b) H of table 6.1 is configured and plotted to the $H-\dot{V}$ diagram. The result is the characteristic curve (b) of the modified system as shown in figure 6.20, which has a steeper slope than curve (a).

6.4.2 Operating point of pumping system.

The ***characteristic curve of the pump $H-\dot{V}$***, which the manufacturer provides, shows the energy height that the pump can yield in the various flow rates. It has no dependence on the pumping system.

[4] The calculations of friction factors and the Moody Diagram are given the book *Μηχανική των Ρευστών*, ("Fluid Mechanics") by N. Pantzalis, Athens: Eugenides Foundation publications, 2008.

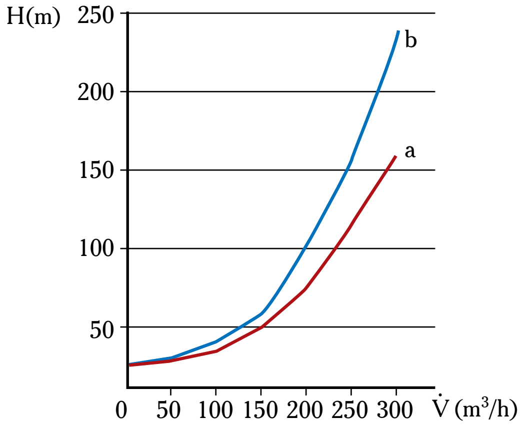


Fig. 6.20

Typical characteristic curves of a pumping system.

Frequently, the height H of this curve is defined as ***given head of the pump***.

The ***characteristic curve of the system*** $\boldsymbol{H-\dot{V}}$, which the user of the pumping system creates, shows the energy height which the system requires from the pump. It has no dependence on the pump, and defined as ***total head of pumping system***.

The energy height that the pump delivers to the system in which it is established meets the energy needs of the system. Therefore, the given head of the pump must be equal to the total head of the system. In other words, the two functions for $H-\dot{V}$ must have a common solution:

$$H = f_1(\dot{V})$$

$$H = a + b \cdot \dot{V}^2$$

Since both functions are depicted with curves, a common solution is the intersection of the two curves. Specifically, if in one graph with the same scale are plotted the characteristic curve of the pump and the characteristic curve of the system where the pump is placed, ***the intersection point of the two curves is the operating point of the pumping system*** (fig. 6.21).

The point of intersection of the pump characteristic curve $H-\dot{V}$ with the pumping system characteristic curve $H-\dot{V}$ is defined as ***system operating point*** (for short, OP). By defining the system operating point from the pump performance diagram, the flow rate ($\dot{V}$), the head (H), the efficiency (η) and the power (P) are determined.

If the features of the system are modified, the characteristic curve is varied (in accordance with the previous paragraph) and the operating point slides on the characteristic curve $H-\dot{V}$ of the pump.

If the pump is operated with an impeller of another diameter or other speed, its characteristic curve $H-\dot{V}$ is displaced and the operating point slides on the characteristic curve of the system. Thus, the operating point of a pumping system regulates itself on the given modifications either of the system characteristics or of the pump.

6.4.3 Calculation of a pumping system.

When a pumping system and the characteristic curves of the pump are given, the system will be self-regulated to the operating point.

The methodological approach for calculating the flow rate ($\dot{V}$), the total or given head (H), the shaft power P_s, and the efficiency (η), can be achieved by following the next steps:

1) If the diagram is composite, the characteristic curve $H-\dot{V}$ of the pump corresponding to the problem conditions is determined. The same is performed for the $P-\dot{V}$ curve.

2) The characteristic curve of the system is plotted on the diagram H – V of the pump.

$$H = a + b \cdot \dot{V}^2.$$

3) The operating point of the pumping system,

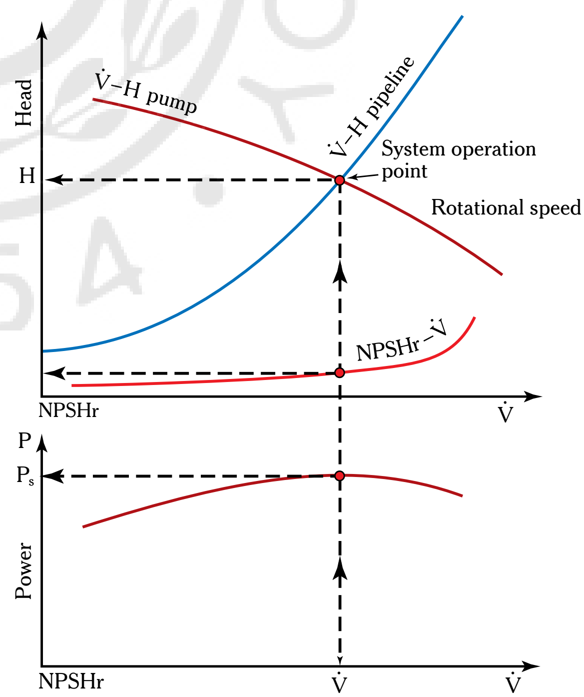


Fig. 6.21

Pumping system operating point on a typical performance characteristic curves diagram.

i.e., the intersection of the two curves $H-\dot{V}$ (pump and pumping system) is determined.

4) From the graph for the system operating point, the values $\dot{V}$, H, P and n are obtained.

The more laborious procedure is the plotting of characteristic system curve $H-\dot{V}$. Sufficient value pairs ($\dot{V}_i$, H_i) need to be identified. The minimum is three pairs, but if the drawing is based on such a small number, it will lack accuracy. The calculation is performed using the Bernoulli equation for the pumping system:

$$H = H_{st} + \frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} + \Sigma h_f$$

The corresponding value for the total head H_i is calculated by giving a value V_i to the flow rate, i.e., losses in the suction pipe and discharge are calculated and, replacing these in the energy equation, the total head H_i is calculated. Then a new value is given to the flow rate and the process is repeated. The energy height H_i or H must be added by the pump.

The delivered power is calculated by: $P_I = \gamma \cdot V \cdot H$, for which the relationship $\eta = P_I / P$ allows us to ascertain whether the values of the chart are read correctly.

Example 4.

An existing pump, having the characteristic curves of figure 6.22, is to be used to pump water in a dis-

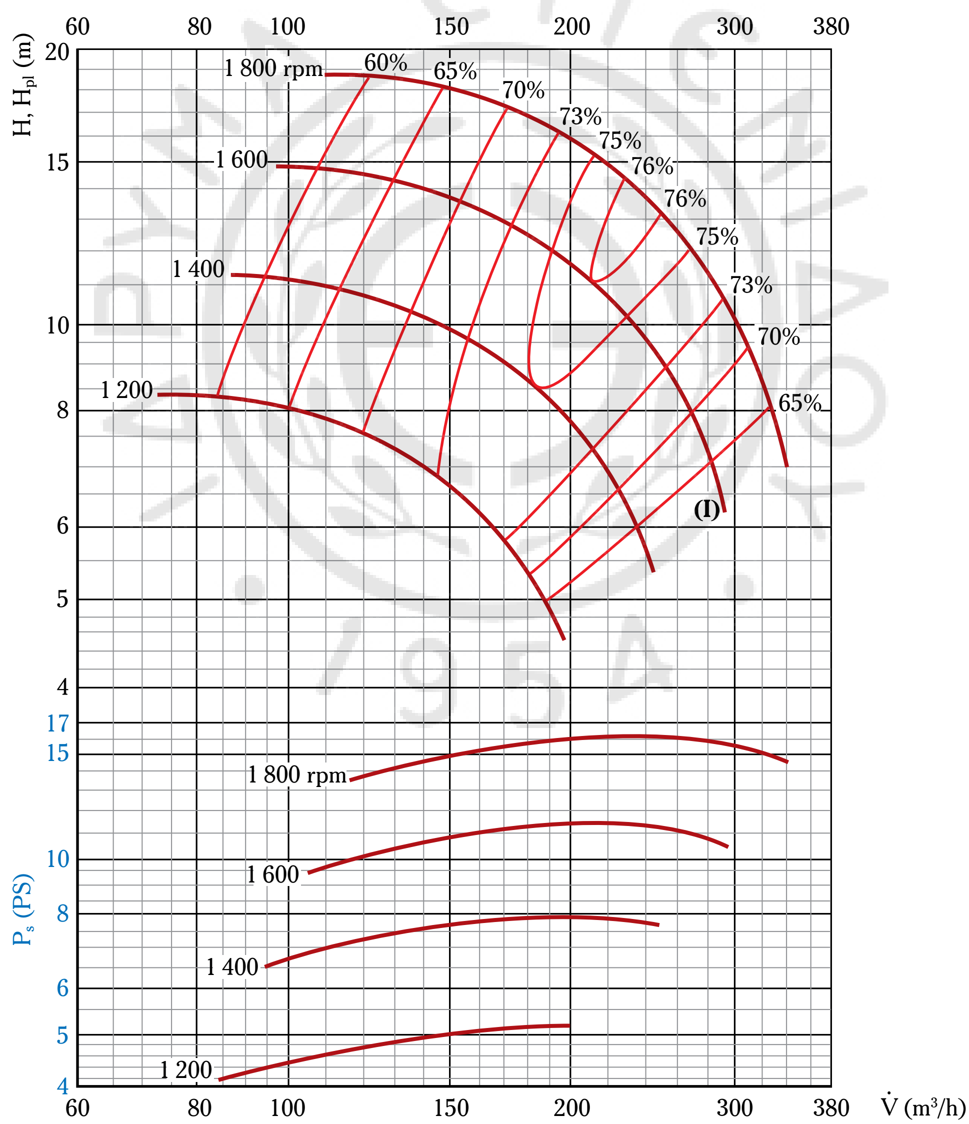


Fig. 6.22

Indicative characteristic curves of the pump (logarithmic scale diagram).

tance of 150 m and at a head difference of 6 m. The pump is operated at 1 600 rpm. Calculate the flow rate $\dot{V}$, the shaft power P_s, the delivered power (P_I) to water and the pump efficiency (η). The suction loss and the secondary losses are considered as negligible. The friction factor f is given indicatively for the first flow rate that is used as 0.0206.

Solution:

Data: L = 150 m, d = 0.15 m, H_{st} = 6 m, n = 1 600 rpm.

Requested: $\dot{V}$, P_s, P_I, η.

a) The curve $H-\dot{V}$ corresponding to 1 600 rpm is selected.

b) The corresponding values of H is obtained, by giving values to $\dot{V}$ and the use of Bernoulli equation. Then the curve of the system is plotted.

$$H = H_{st} + \frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2 \cdot g} + \Sigma h$$

or $H = H_{st} + h_{df}$ for pipeline, because $v_2 = v_1$ and $p_2 = p_1$

or $$H = H_{st} + \frac{8}{\rho^2 g} \cdot f \cdot \frac{L}{d^5} \cdot \dot{V}^2 \qquad (1)$$

where $$h_{df} = \frac{8}{\rho^2 g} \cdot f \cdot \frac{L}{d^5} \cdot \dot{V}^2.$$

By the Continuity equation the speed of liquid at flow rate $\dot{V}$=60 m^3/h is calculated, where:

$$v_1 = \frac{4 \cdot \dot{V}_1}{\pi \cdot d_1^2}, \text{ for } \dot{V} = 60\,\text{m}^3/\text{h} \text{ and } d_i = 15\,\text{cm}$$

By substituting in relationship resulting

$$v = \frac{4 \cdot \dfrac{60}{3\,600}}{\pi \cdot 0.15^2}$$

or $$v = 0.94\,\text{m/s}$$

Similarly are calculated the speeds at the other flow rates.

The friction coefficients (f) are given in the table below, where V = 60 m^3/h, f = 0.0206 etc. They are calculated using the Moody diagram.

By replacing the relation (1) results:

$$H = 6.93\ \text{m}$$

Then, taking into account the changes for each selected flow rate, the following table is filled in accordingly:

$\dot{V}$ m^3/h	v m/s	f	H m
60	0.94	0.0206	6.93
80	1.26	0.0202	7.62
100	1.57	0.0199	8.50
150	2.56	0.0195	**11.52**
200	3.14	0.0193	**15.72**
240	3.77	0.0192	**19.92**

By putting the points on the diagram $\dot{V}-H$ and their connection, the characteristic curve (II) of the system is plotted, as illustrated in figure 6.23.

c) The intersection of the pump (I) and system (II) curves is the operating point of the pumping system.

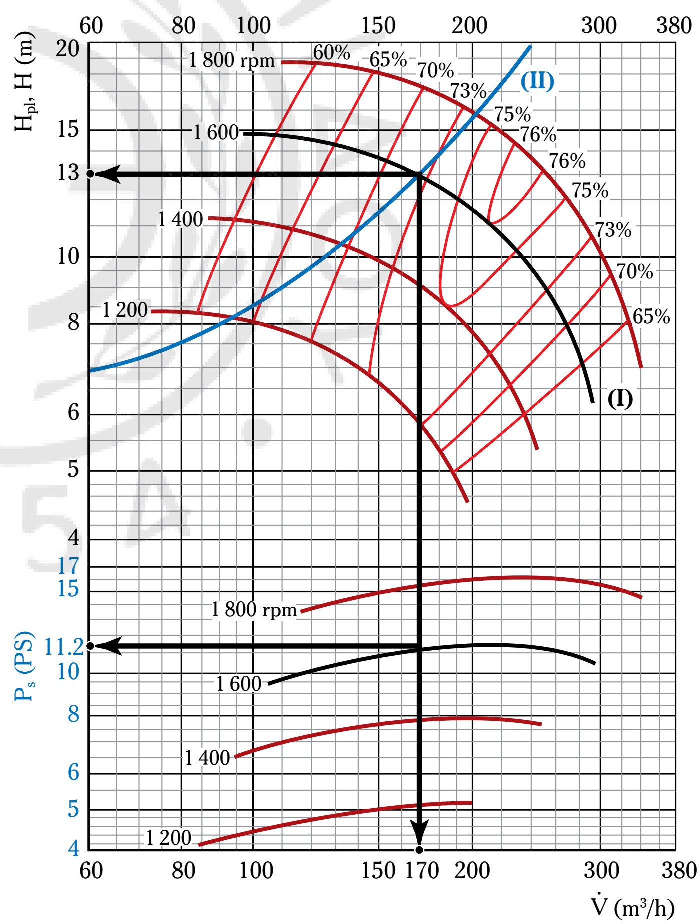


Fig. 6.23

Typical pump operating characteristic curves (on the vertical axis of ordinate has been placed the total head H and the pipeline head H_{pl}).

d) From the diagram (fig. 6.23) is calculated:

$$\dot{V} = 170\,m^3/h$$

$$H = 13\,m$$

$$\eta = 72.5\%$$

$$P_s = 11.2\,PS = 8.25\,kW$$

because $\qquad P_I = \gamma \cdot \dot{V} \cdot H$

if applying the values results: $P_I = 6.02\,kW$

(Also: $P_s = P_I/\eta \Rightarrow P_s = 8.25\,kW$).

Example 5.

Centrifugal pump with characteristic curves of figure 6.24 elevates water to a head of 20 m. The discharge pipe is made of cast iron and the total coefficient local loss is K = 3.5.

a) Since the pairs of flow rate and head values in the following table 6.2 are given, plot the characteristic curve of the system and find the efficiency, the shaft and the delivered power.

b) If local losses increasing in K = 30, the changes in value of flow rate and head are provided in pairs at table 6.3. By developing a new characteristic curve of the system, find the new values of efficiency, shaft power, and delivered power.

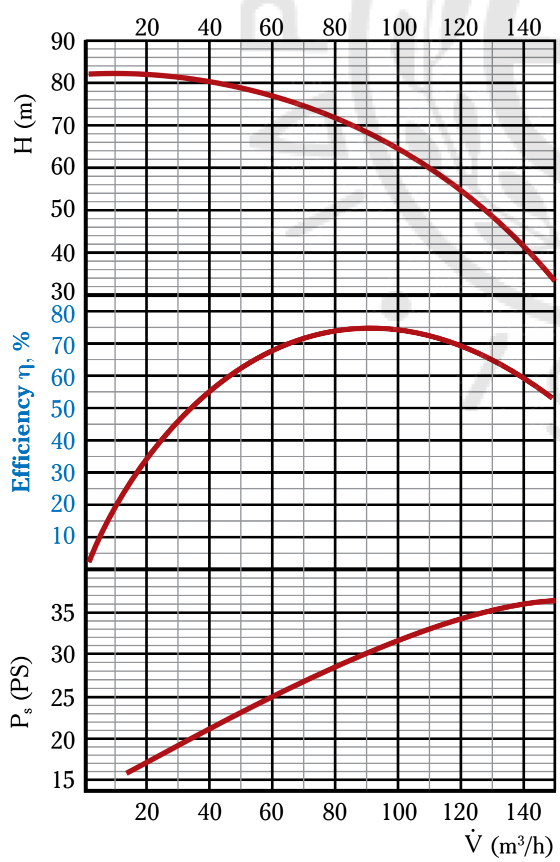


Fig. 6.24

Indicative characteristic operating curves of the pump.

Table 6.2

$\dot{V}$ (m^3/h)	H (m)	$\dot{V}$ (m^3/h)	H (m)
40	30.0	80	60.0
60	42.5	90	70.6
70	50.6	100	82.4

Table 6.3

$\dot{V}$ m^3/h	H m	$\dot{V}$ m^3/h	H m
30	31.9	60	67.8
40	41.3	70	85.1
50	53.2	80	105

Solution:

Data: a) K = 3.5 b) K = 30.

Requested: η, P_s, P_I.

a) The intermediate process concerning the calculation of H and related to the losses in the piping system is deemed as completed, since the value pairs in the table are taken into account and hence the head H is given, so:

1) The system curve (a) is plotted.

2) The OP (operating point) as the intersection of two curves $H-\dot{V}$ (fig. 6.25) is identified.

The corresponding values to the qualities are: $V = 89\,m^3/h$, the head is $H = 68.5\,m$, and $\eta = 75\%$.

$$P_s = 30\,PS = 22.1\,kW$$

$$P_I = \gamma \cdot \dot{V} \cdot H \Rightarrow P_s = 16.6\,kW$$

$$(\text{Also: } P_I = \eta \cdot P_s = 16.6\,kW)$$

b) If local losses increase in K = 30, the changes in flow rate and head are these of table 6.3.

Reflecting the points in the diagram, the system curve (b) is plotted in figure 6.25, with a steeper slope than (a). The new operating point slides onto the characteristic curve of the pump and gives less flow rate (throttling). The new qualities values are:

$$\dot{V} = 65\,m^3/h, \quad H = 75\,m$$

$$\eta = 70.5\%$$

$$P_s = 26\,PS = 19.1\,kW$$

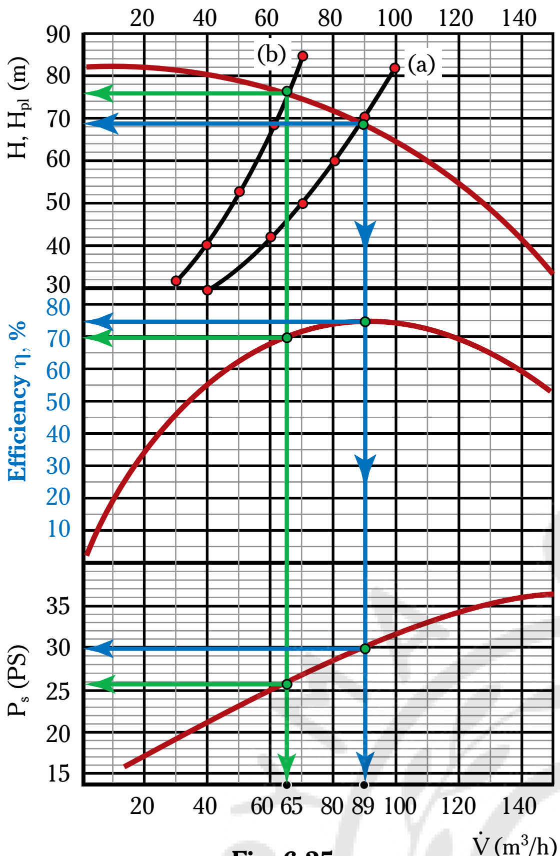


Fig. 6.25
Characteristic curves of the pump with the curves of pipelines and the operating points.

$$P_I = \gamma \cdot \dot{V} \cdot H \quad \text{or}$$

$$P_I = 10\,000 \frac{N}{m^3} \cdot \frac{65}{3\,600} \frac{m^3}{s} \cdot 75\,m = 13\,500\,W$$

$$\text{or} \quad P_I = 13.5\,kW$$

$$(\text{Also: } P_I = \eta \cdot P_s = 13.5\,kW)$$

6.4.4 The curves NPSH–$\dot{V}$ and conditions of cavitation.

One of the pump characteristic curves is the curve of NPSHr–$\dot{V}$. Therefore, by finding the operating point of the pump system, the net positive suction head required NPSHr from the pump may be determined. This does not have any particular significance if not compared to the available net positive suction head NPSHa of suction pipeline. The correlation of the available and required NPSH is extremely important since it is associated with the problem of pump cavitation.

When knowing the data for the suction pipeline, the net positive suction head available NPSHa in connection with the flow rate $\dot{V}$ may be calculated. Therefore, it is possible to design the characteristic NPSHa–$\dot{V}$ curve of the system. The NPSHa is given as:

$$NPSHa = \left(\frac{p_S}{\gamma} + \frac{v_s^2}{2 \cdot g}\right) - \frac{p_{vap}}{\gamma} \qquad (6.13a)$$

By applying the Bernoulli equation between the free surface (1 in figs 6.19 or 4.11) of the suction tank and the inlet of the pump, arises:

$$z_S - z_1 + \frac{p_1 - p_{vap}}{\gamma} + \frac{v_1^2 - v_S^2}{\gamma} = h_{sf} \Rightarrow$$

$$\Rightarrow \frac{p_S}{\gamma} = \frac{p_1}{\gamma} - \frac{v_S^2}{2 \cdot g} - H_{sts} - h_{sf} \qquad (6.13b)$$

where: $z_S - z_1 = H_{sts}$ is the static suction head. Substituting in equation (6.13) is obtained:

$$NPSHa = \frac{p_1 - p_{vap}}{\gamma} - H_{sts} - h_s \qquad (6.13c)$$

According to the losses equation:

$$h_s = \frac{8}{\pi^2 g \cdot d_s^4}\left(f \cdot \frac{L_s}{d_s} + K_s\right) \cdot \dot{V}_2$$

Therefore equation (6.13c) takes the form:

$$NPSHa = a - b \cdot \dot{V}_2 \qquad (6.14)$$

$$a = \frac{p_1 - p_{vap}}{\gamma} - H_{sts} \qquad (6.14a)$$

$$b_s = \frac{8}{\pi^2 g \cdot d_s^4}\left(f \cdot \frac{L_s}{d_s} + K_s\right) \qquad (6.14b)$$

Equation (6.14) indicates that the available net positive suction head of the system decreases with increasing flow rate. This function is plotted as ***curve NPSHa–$\dot{V}$***. To construct it, a similar procedure to that for the construction of the characteristic curve of the system is followed, by giving values to the flow rate $\dot{V}$ and taking the respective values of NPSHa. The curve has the form shown in figure 6.26.

It is observed that the NPSHr required by the pump (whose curve is given by the manufacturer) is increased by increasing the flow, while the NPSHa available from the system (the curve that is plotted) decreases with increasing flow rate. Therefore an intersection point of the two curves is created, which corresponds to a flow rate $\dot{V}_{ar}$, where:

$\dot{V} = \dot{V}_{ar}$ applies NPSHr = NPSHa

$\dot{V} > \dot{V}_{ar}$ applies NPSHr > NPSHa

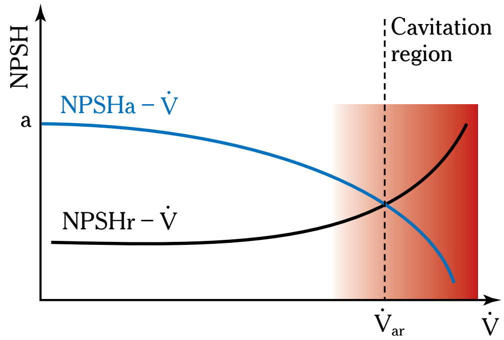


Fig. 6.26
Curves of available and required NPSH-V.

An indispensable prerequisite for a smooth operation of the pump without cavitation is for the available NPSH to remain higher than required, and indeed the difference must be significant. The characteristic curve $H - \dot{V}$ is given by the manufacturer for normal operating conditions of the pump. When these are not ensured, the manufacturer's curve $H - \dot{V}$ does not apply. Hence, in cavitation region the curve $H - \dot{V}$ drops sharply, as shown by dashed line of figure 6.27.

Generally, the operation of a pumping system in the cavitation region must be prohibited. And this not only because of the vertical fall of the head as shown in the figure, but mainly due to the destructive effects resulting to the pump if the operation is

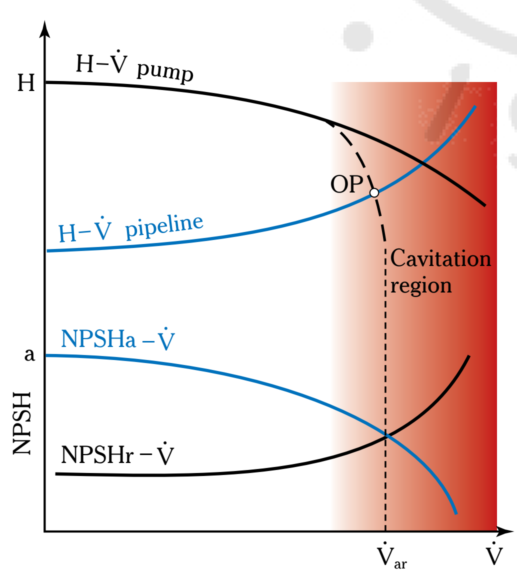


Fig. 6.27
Typical characteristic curve $H - \dot{V}$ and cavitation.

under cavitation conditions. Therefore, since the characteristic curve of NPSHr–$\dot{V}$ pump is known, the corresponding curve NRSHa–$\dot{V}$ of the suction system should intersect with the first one at the greatest possible flow rate. As is apparent from equation (6.14), this occurs either when the coefficient ***a*** increases (moving the curve NPSHa–$\dot{V}$ upwards), or when the coefficient ***b*** decreases (smaller slope of the curve). The first case is achieved by reducing the suction static head (or increasing the pressure in the suction tank). The second case is obtained by reduction of losses in the suction pipe (i.e., using a pipe with larger diameter or smaller length, in order to minimize the local losses in the suction).

Example 6.

A pump with characteristic curves shown in figure 6.28 is placed in a system that pumps water at 40 °C. On the free surface of the suction tank atmospheric pressure prevails. The optimum operation of the pump is defined by the region of the diagram with the green color. According to the flow rate and NSPHa values of table 6.4, find the cavitation criti-

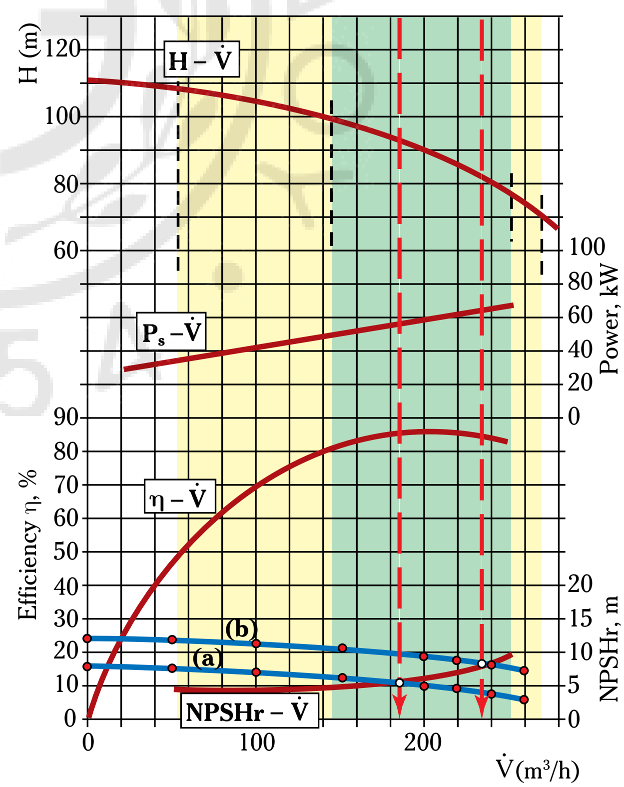


Fig. 6.28
Characteristic curves of pumping systems and pump optimum operating regions.

Table 6.4

$\dot{V}$ m^3/h	NPSHa m (a)	NPSHa m (b)
0	7.57	10.75
50	7.43	10.61
100	7.00	10.18
150	6.32	9.5
200	5.35	8.58
220	4.88	8.11
240	4.37	7.60

cal points of flow rate, if the suction tank surface is located:

1) 2 m below the pump inlet, and
2) 2 m above the pump inlet.

Solution:

1) By plotting the points of table 6.4 on the chart and designing the curve (a) as illustrated in figure 6.4k, the intersection to the NPSH curve is revealed. At the intersection point the critical flow rate for cavitation is obtained:

$$\dot{V}_{ar} = 185\ m^3/h$$

2) In the second instance, the NPSHa curve moves higher by 4 m. The new designed curve (b) of $H-\dot{V}$ chart as shown in figure 6.27 intersects the NPSHr curve at a point corresponding to higher flow rate:

$$\dot{V}_{ar} = 235\ m^3/h$$

It is noted that the positioning of the pump as in case (a) should be avoided, because the utilization by the pump of its optimum operating region is precluded.

6.5 Pumps in coupling. Alternative pump installations.

The discharge of liquids with high capacity (flow rate) in places located very high by using one pump only may not be a possible or the most economical solution. In certain circumstances, the pumping system requirements cannot be reached by the existing pump. Therefore pumps in connection or coupled pumps are used. So, a satisfactory solution to such pumping problems is provided by the connection of two or more pumps in parallel or in series. Frequently, the parallel connection is used to increase the capacity to the pumping system, and the connection in series to increase the head.

6.5.1 Pumps coupled in parallel.

1) Parallel connection of two identical pumps.

Two (or more) identical pumps are coupled together when arranged to discharge in the same pipe (the suction is frequently performed by different pipes). Such connection is shown in figure 6.29

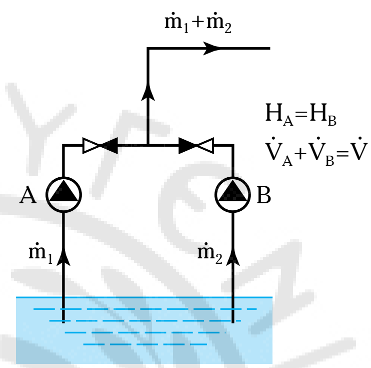


Fig. 6.29 *Principle of connection in parallel.*

The design of the characteristic curve of two identical pumps connected in parallel $H = f_1(\dot{V})$, if the characteristic curve of each pump is $H = f(\dot{V})$, is performed as follows:

Construction of characteristic curve $H-\dot{V}$ for pumps in connection: If two parallel pumps are linked in a system and deliver liquid at a certain head, the corresponding flow rate $\dot{V}$ is the same and the rotational speed is the same ($n_1 = n_2$). So, each pump delivers the same head, i.e., the same power is delivered to the weight unit of liquid that passes from the one or the other pump. But the flow rate of the liquid that is discharged from each pump is equal to half of the total (in parallel connection the flow rates are added):

$$H_i = H_{1i} = H_{2i}$$

$$\dot{V}_i = \dot{V}_{1i} + \dot{V}_{2i} = 2\dot{V}_{1i}$$

Consequently, by moving each point of $H = f_1(\dot{V})$ horizontally (i.e., at the same head) until the flow rate is doubled, the characteristic curve of parallel connected pumps $H = f(\dot{V})$ (fig. 6.30) is generated.

If the connected pumps were three, then

$$\dot{V} = \dot{V}_1 + \dot{V}_2 + \dot{V}_3 = 3\dot{V}_1$$

i.e., in the horizontal direction of flow would be tripled.

Calculation of the pumping system: after the design of curve $H = f(\dot{V})$, the characteristic curve of the system $H = a + b \cdot \dot{V}^2$ is calculated and drawn.

The operating point of the connected pumps system is determined at the intersection of the two curves $H = f(\dot{V})$ and $H = a + b \cdot \dot{V}^2$. Care is needed because each pump delivers a head H and supply $\dot{V}/2$ (if the pumps are two). Therefore ***the operating point of each pump*** is on the same head and at half the flow rate (fig. 6.30). In this point the efficiency, the shaft power and the NPSHr of each pump will be obtained.

The operating point, if only one pump was placed in the pumping system, would be the OP′ and the head the H_1'.

In a particular pipeline system where two parallel pumps are placed, it is observed that the given flow rate $\dot{V}$ is greater than the flow which one pump located in the same system would give, but not double. Accordingly, the head H would be greater than the head which one pump would yield. So, the following relationships are applied:

$$\dot{V}_1' < \dot{V} < 2\dot{V}_1' \quad \text{and} \quad H = H_1 = H_2 > H_1'$$

If the identical pumps have unstable characteristic curve $H = f_1(\dot{V})$ and the total head of the system is greater than H (the height corresponding to $\dot{V} = 0$), it is necessary for the two pumps to start simultaneously. If not, the pump which delays in starting cannot overcome the pressure in the discharge pipe in order to provide its capacity (flow rate). The parallel connection of these pumps is avoided. This is a significant drawback of centrifugal pumps with unstable characteristic $H - \dot{V}$ curve.

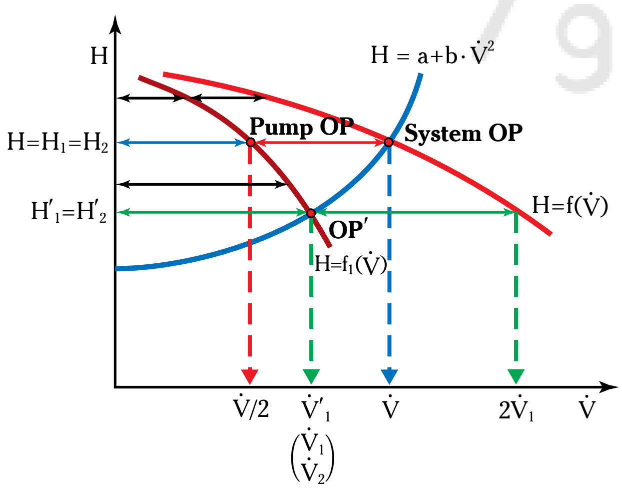


Fig. 6.30
Characteristic curves of identical pumps in parallel connection.

Example 7.

Two identical pumps, with characteristic curves shown in figure 6.22, are connected in parallel in a water pumping system. The pumps operate at 1 200 rpm.

1) Designate the head on the limit of pump operation using the diagram of figure 6.4e, and the flow rate for pumps connected in parallel.

2) Given the pairs of values of head and flow rate resulting from the parallel operation of the pumps as shown in table 6.5, plot the characteristic curve of the system and calculate the flow rate, the head, the shaft power, the delivered power, and the efficiency of the system.

Table 6.5

$\dot{V}$ m^3/h	v m/s	f	H m
60	0.65	0.0231	**3.34**
80	0.87	0.0228	**3.60**
100	1.09	0.0226	**3.91**
150	1.64	0.0222	**5.03**
200	2.18	0.0221	**6.57**
240	2.62	0.0220	**8.13**
300	3.27	0.0219	**11.00**

3) If only one pump operates, find what the corresponding qualities will be (given 1 PS = 0.736 kW).

Solution:

1) The curve corresponding to the two parallel connected identical pumps is obtained from the curve (I) of a pump operated at 1 200 rpm. That is why, at fixed heads, flow rates are doubled:

$$H = 8.4\,m \Rightarrow \dot{V}_1 = 75\,m^3/h \Rightarrow \dot{V} = 150\,m^3/h$$
$$H = 8.2\,m \Rightarrow \dot{V}_1 = 93\,m^3/h \Rightarrow \dot{V} = 186\,m^3/h$$
$$H = 7.8\,m \Rightarrow \dot{V}_1 = 110\,m^3/h \Rightarrow \dot{V} = 220\,m^3/h$$
$$H = 7.5\,m \Rightarrow \dot{V}_1 = 123\,m^3/h \Rightarrow \dot{V} = 246\,m^3/h$$
$$H = 7.0\,m \Rightarrow \dot{V}_1 = 140\,m^3/h \Rightarrow \dot{V} = 280\,m^3/h$$

The characteristic curve of the two connected pumps (Ia) results in this way (fig. 6.31).

2) The given pairs of values of flow rate and head from table 6.5 are plotted to the diagram $\dot{V} - H$, and connected in line so as to design the characteristic curve (II) of the system (fig. 6.31) (the first two points cannot be captured in the diagram due to low total head).

The intersection of the characteristic curve in parallel-connected pumps (Ia) and that of the system (II), illustrate the operating point of the pumping system (OP). So from the diagram of figure 6.31 is obtained that:

$$\dot{V} = 230\ \mathrm{m^3/h}$$

$$H = 7.7\ \mathrm{m}$$

The requested of the example that constitute qualities of the pumps are revealed in the operating point (OP′) of each pump. Therefore:

$$\eta = 68.5\ \%$$

$$P_I = 4.7\ \mathrm{PS} = 3.46\ \mathrm{kW}$$

3) If in the same system only one pump operates, the characteristic curve of the system will not change. Therefore, the operating point (OP_b), will be the intersection of the characteristic curve of the pump (I) with the characteristic curve of the system (II).

From the diagram of figure 6.31 obtained:

$$\dot{V} = 173\ \mathrm{m^3/h}$$

$$H = 5.7\ \mathrm{m}$$

$$\eta = 72.5\ \%$$

$$P_s = 5.1\ \mathrm{PS} = 3.75\ \mathrm{kW}$$

$$P_I = \rho \cdot g \cdot \dot{V} \cdot H$$

by replacing the values $P_I = 2.7\ \mathrm{kW}$

Observe that, in the parallel connection of two identical pumps of the example, the flow rate in-

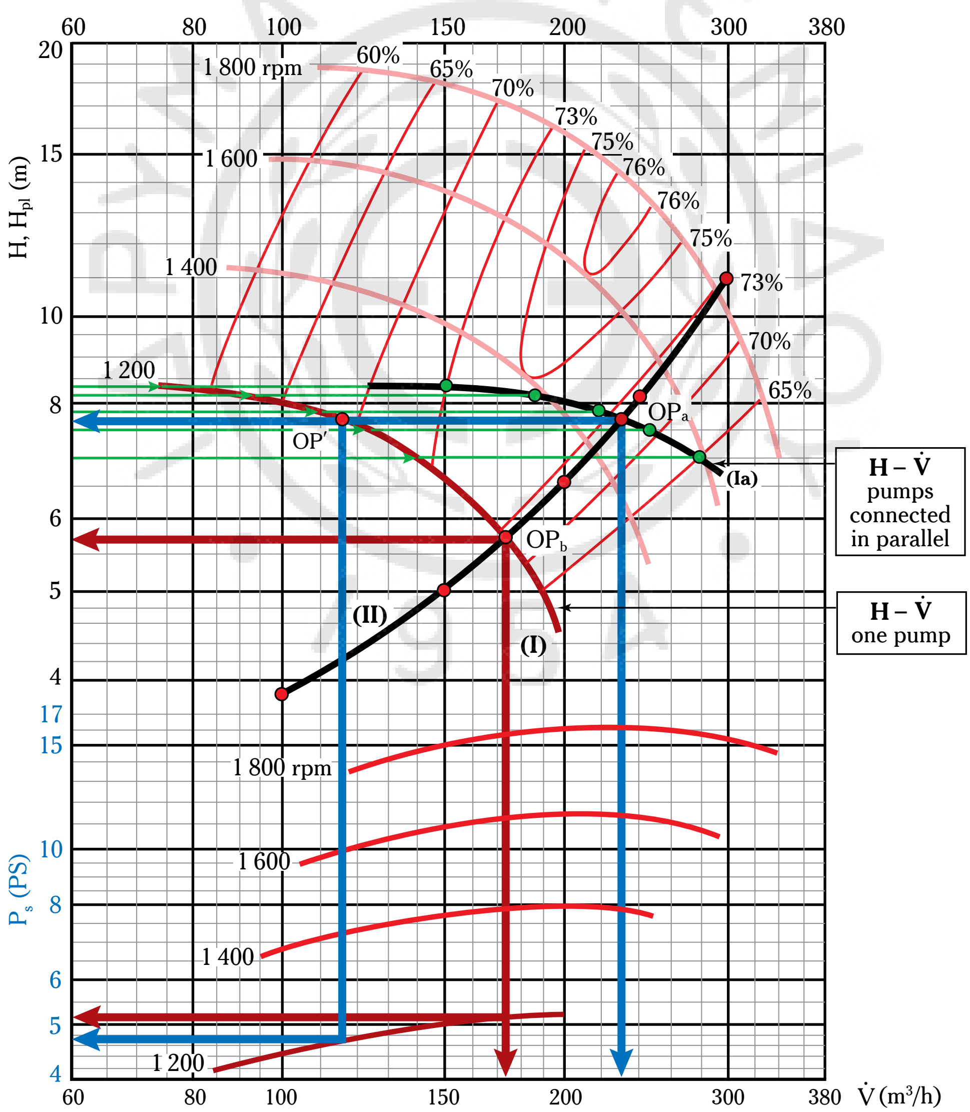


Fig. 6.31

Typical operating curves of two parallel-connected pumps.

creased by 33%, but with significant energy burden because of greater head loss, and a small decline in the efficiency of the pumps.

2) Parallel connection of two different pumps.

If the two pumps have different characteristic curves head-flow rate $H = f_1(\dot{V})$ and $H = f_2(\dot{V})$ and discharging to the same pipe, the characteristic curve of the two parallel connected pumps system is $H = f(\dot{V})$.

The design of the new curve is based on the same rationale as the parallel connection of two identical pumps. If the two pumps system delivers head H_i, then each individual pump will yield the same head, because the system flow rate will be equal to the sum of the two pumps flow rate:

$$H = H_i = H_{2i} \text{ and}$$

$$\dot{V}_i = \dot{V}_{1i} + \dot{V}_{2i}$$

By summing the flow rates at different heights, enough points for construction of the ***characteristic curve of the parallel connected pumps*** $H = f(\dot{V})$ are obtained (fig. 6.32). It is noted that for the heights which cannot be yielded by one of the two pumps [in figure the first curve, $H = f_1(\dot{V})$], the characteristic curve of the system coincides with the characteristic curve of the pump with greater maximum head (in fig. 6.32, the first dotted part of the new line, on the left side).

Then is calculated and plotted ***the characteristic curve of the system $H = a + b \cdot \dot{V}^2$*** and the operating point of the system (OP) at the intersection with the $H = f(V)$ is determined. The operating points of the pumps are OP_1 and OP_2. In these we can find the efficiency, the shaft power and the NPSHr of each pump. It is observed that if $\dot{V}$ and H are the flow rate and head of the pumps system in parallel connection, and V_1', H_1', V_2', H_2' the corresponding points, if each pump operated alone in the same pipeline, then:

$$H > H_2' > H_1'$$

$$\dot{V}_1' < \dot{V}_2' < \dot{V} < \dot{V}_1' + \dot{V}_2'$$

Namely, although the parallel pump connection is primarily used to increase the flow rate in a system, the result that is obtained is an increase in total head. This is because the flow rate is less than the sum of flow rates that each pump would give if it operated alone in the pumping system. This happens because the increase in flow rate significantly increases the head loss, and thus the total height.

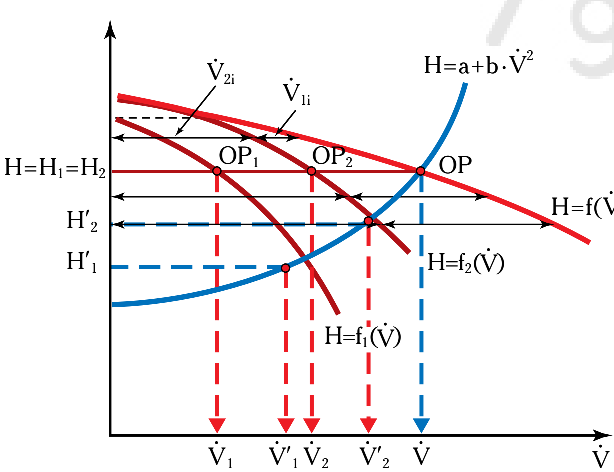


Fig. 6.32
Different pumps in parallel connection.

3) Different pumps in parallel connection that are located far apart.

The difference with the previous connections of pumps occurs due to:

1) the difference in static head which each pump is facing, and

2) the difference of the discharge portion of each pump before it is connected to the common pipe (sections 1-3 and 2-3 of figure 6.33).

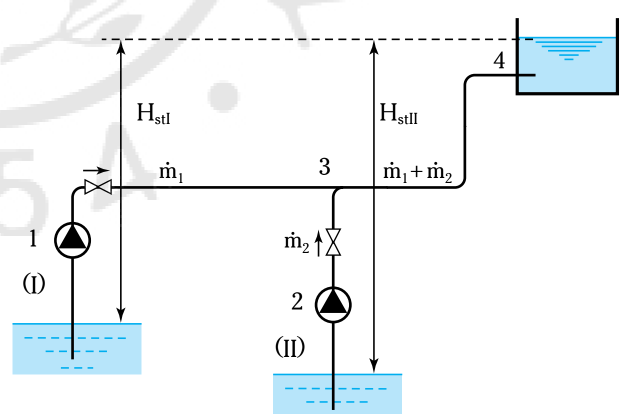


Fig. 6.33
Pumps in parallel connection which are apart from each other.

Thus, before identifying the curve of operation of the two pump system, reduction of the pumps to the same hydraulic conditions (prevailing in portion 3-4) should be performed as described below.

Let us suppose for simplicity that the portion 2-3 is small, and so with negligible losses.

– ***Reduction of the pumps to the same hydraulic conditions.***

Suppose H_I and H_{II}, the characteristic curves of two pumps.

According to figure 6.33: $H_{stI} < H_{stII}$. Because the pump (I) has to face less static head than the (II), its curve is displaced upwards by ΔHs so that: $\Delta H_{st} = \Delta H_{stI} - \Delta H_{stII}$ and resulting the H_I' (fig. 6.34). But pump I has to face the losses of portion 1-3, so in point 3 is not assigned the height that is indicated by the H_I' curve. Therefore, the losses are removed from the H_I', so as: $h_{13} = a_1 \cdot \dot{V}^2$, and resulting the H_I''. For the removal, first the tube curve 1-3 is plotted where: $h_{13} = a_{13} \cdot \dot{V}^2$ and then removed from the H_I' so as: $H_I'' = H_I' - h_{13}$.

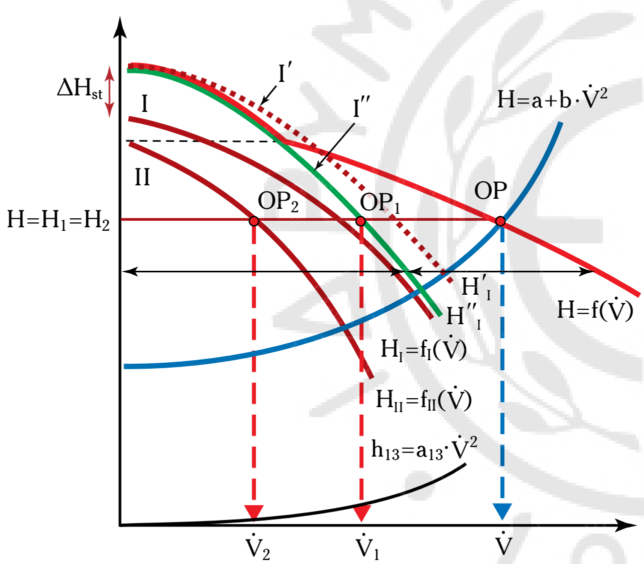


Fig. 6.34
Curves of pumps in parallel connection which are apart from each other.

Thus, at the common portion of the discharge 3-4 and with static head H_{stII}, the pump (I) contributes to the static head and to the head loss with the curve H_I'' and the pump II by the H_{II} curve. For the pump II no reduction was needed, because the H_{stII} is used and the losses in the portion 2-3 are assumed negligible. From the composition of the two curves H_I' and H_I, as in the previous section, the dotted line $H = f(\dot{V})$ resulted.

Plot the ***characteristic curve of the system*** 2-3-4 in a static height H_{stII}:

$$H = a + b \cdot \dot{V}^2$$

The intersection of curve $H = f(\dot{V})$ and $H = a + b \cdot \dot{V}^2$ is ***the operating point of the pumping system***, which together with the curve is shown in figure 6.34.

6.5.2 Two identical pumps connected in series.

As the parallel connection of pumps frequently is used to increase the capacity (flow rate), the connection in series is not so common. This is because when facing large heads (without corresponding high flows) which are not addressed by one (single-stage) pump, the preferable solution is using a multi-stage pump. But if the large head needs are not permanent, the solution may be provided by the pumps connected in series. It is noted that increase in total height can also be achieved by parallel connection. But this increase is due to an increase of the head loss (due to increasing flow rate) and does not serve any additional energy requirements of the pumping system.

In pumps connected in series, discharge of the first pump performs the suction to the next. Therefore, the flow rate is common, since the heads are added, so:

$$\dot{V} = \dot{V}_1 = \dot{V}_2 \quad \text{and}$$

$$H = H_1 + H_2 = 2H_1$$

To plot the characteristic curve of two identical pumps connected in series, we double the coordinates H of the characteristic curve of one pump $H = f_1(\dot{V})$. Thus the curve resulting is given by $H = f(\dot{V})$ (fig. 6.35).

Calculating and plotting the characteristic curve of the system $H = a + b \cdot \dot{V}^2$, the operating point of the pumping system is determined at the intersection of the two curves. The flow rate and head of each pump respectively are:

$$\dot{V}_1 = \dot{V}_2 = \dot{V}$$

$$H_1 = H_2 = H/2$$

Therefore each pump operates at the "pump OP" of figure 6.35 and at this point from the graphs are taken the efficiency, the shaft power and the NPSHr of the pump.

If there was only one pump in the pumping system, the system operating point would be the intersection of $H = f_1(V)$ to $H = a + b \cdot \dot{V}^2$, i.e., OP′ of figure 6.35. We observe a very large increase of head that is attributed to the liquid (the head H would not be covered by one pump), and there is a remarkable increase in flow rate (as a result of the increase in head that is attributed). Hence, it is observed that:

$$H_1' < H < 2H_1' \quad \text{and} \quad \dot{V}_1' < \dot{V}$$

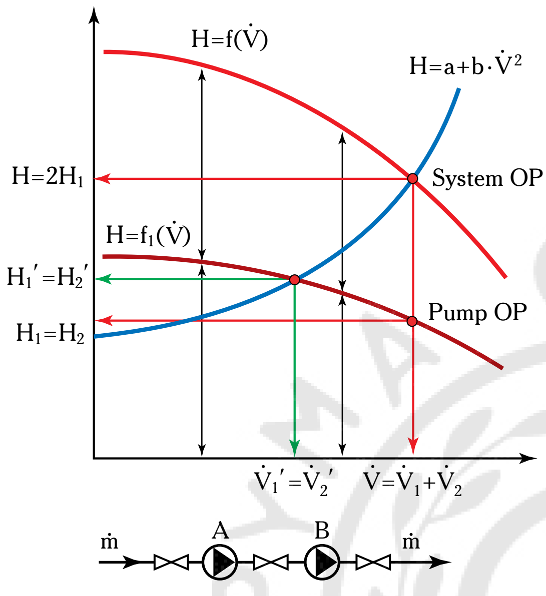


Fig. 6.35
Curves and connection of two identical pumps in series.

6.5.3 Pump supply in two pipelines.

A pump can supply liquid in two (or more) pipelines which branch off in its discharge. In this case, the characteristic curve of the pump $H = f(\dot{V})$ does not alter, but there are two characteristic curves for pipelines, the: $H_1 = a_1 + b_1 \cdot \dot{V}^2$ and $H_2 = a_2 + b_2 \cdot \dot{V}_2^2$, that are created. The question that arises is whether there is an operating point for the system and how it could be determined.

If the two discharge pipes are transferring the liquid at the same static head and the same pressure, then $a_1 = a_2 = a$, where $a = H_{st} + (\Delta P/\gamma)$ and the characteristic curves are the (1) and (2) that are illustrated in figure 6.36.

We assume that the pump delivers to the fluid a head H (energy on unit weight). This is the head for the liquid moving through either pipe (1) or pipe (2) so that: $H = H_1 = H_2$

The flow rate at pipe (1) is $\dot{V}_1$ and at (2) is $\dot{V}_2$, so the total flow in two pipes is:

$$\dot{V} = \dot{V}_1 + \dot{V}_2$$

Hence, adding the two flow rates in the pipes for various heads, sufficient points are obtained to plot the ***characteristic curve of the two pipe system*** (3). The intersection of this curve with the characteristic of the pump $H = f(\dot{V})$ gives ***the operating point of the pumping system***. In the OP of the system, the flow rate $\dot{V}$ and the head H are determined, which are:

$$H = H_1 = H_2$$

Therefore at intersections of the head with the curves (1) and (2) the flow rate in each pipe $\dot{V}_1$ and V_2 are identified, and the total flow rate $\dot{V}$ is given as $\dot{V} = \dot{V}_1 + \dot{V}_2$.

If the supply in one pipe is closed, for example in pipe (1), the operating point of the system is moved to the intersection of the characteristic curve of the other pipe on the typical curve of the pump (i.e. the OP_2 with head H_2' and flow rate $\dot{V}_2'$).

It is observed that when a pump feeds two parallel pipes, the total flow rate increases while the head decreases (in relation to the values shown when feeding only one pipe). This is because if liquid is supplied in both pipes, the total diameter of discharge pipe is given as:

$$A = A_1 + A_2 = \pi \cdot (d_1^2 + d_2^2) / 4$$

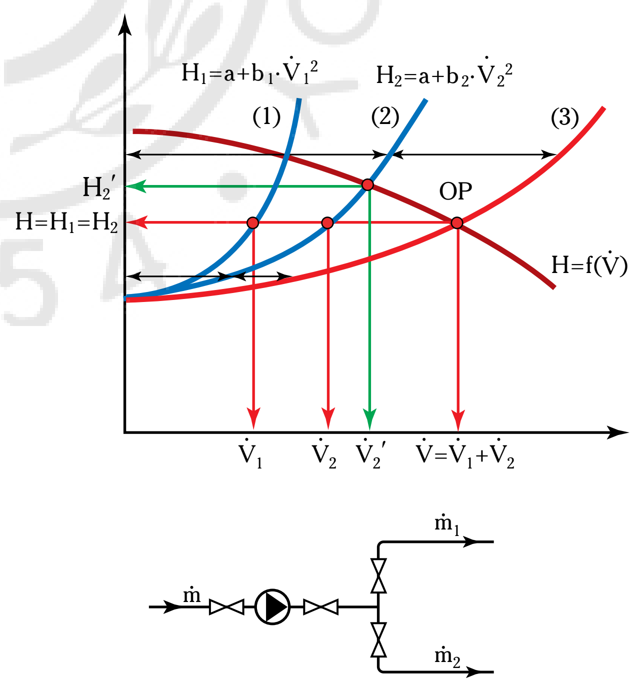


Fig. 6.36
Typical curves and arrangement of pumping system where pump feeds liquid in two pipelines.

CHAPTER SEVEN
Pumps established on ships

Introduction.

The types of pumps that are described below are typically those used on ships. The description is general and may not apply in all cases, because the pumps used depend on various factors resulting from the propulsion plant study (see Ch. 8), or the specific design characteristics of each manufacturer.

7.1 Piston direct acting stem pumps which are used on ships.

Direct acting steam pumps are used on ships (usually old technology ships) as boiler feed water pumps, transfer pumps, fuel oil supply pumps, ballast pumps, fire, fresh water pumps, etc., a characteristic type being the Weir pump and the Worthington feed pump. More extensively:

1) The ***Weir pump*** (fig. 7.1) is a type of vertical arrangement pump which is single cylinder, when the flow requirements are low, and double cylinder when they are high. It is a double acting pump, with separate suction and discharge valves located on each side of the piston cylinder, and their speed ranges from 25 to 40 full strokes per minute. The cross section of the steam cylinder is about twice the cross section of the liquid cylinder, so that, from the total power which is conveyed through the piston rod to the relatively smaller surface of the liquid piston of the pump is caused a much higher increase in the pressure of the liquid, compared to the steam pressure.

The increase of liquid pressure becomes necessary especially when the pump is used as a feed water pump for the steam boiler, where the water, during the inlet to the drum, should prevail over the steam pressure. The operating steam distribution mechanism is of the simplex-type (par. 2.6.1).

2) The ***Worthington feed water pump*** is used as an auxiliary pump in boiler piping systems (fig. 7.2). The pump is placed in horizontal or vertical arrangement. It is a double acting piston pump, and when they are in pairs, the movement of the pistons is realized through a coupled connection of piston rods, which regulates steam flow rate. The distribution device of steam is of the sliding valves type (par. 2.6.1).

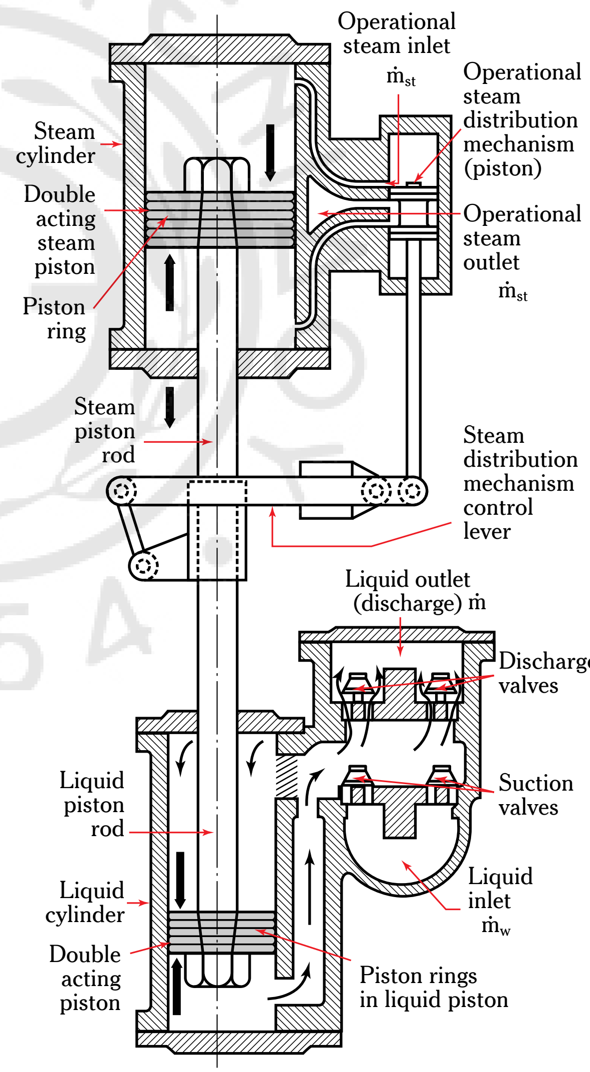


Fig. 7.1
Weir type piston pump.

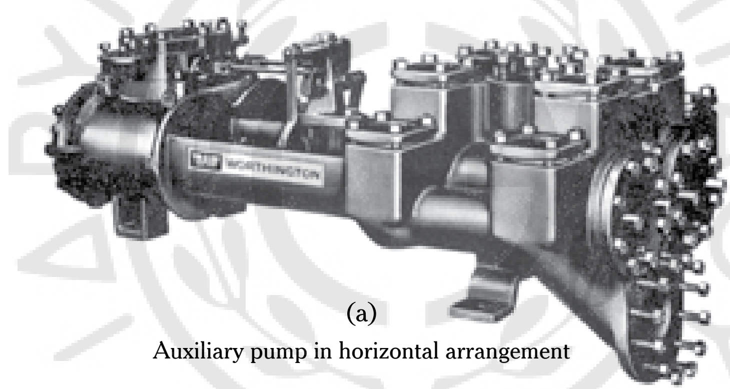
The protection against piston impact to the cylinder head is realized with entrapping part of the steam from the piston, which covers the discharge valve as approaching the end of its route. The compression of trapped steam achieves a smooth deceleration of the piston, and a change at the dead spots in the end of the route of its course. As in Weir type pumps, in these pumps too the steam cylinder pistons are bigger than the liquid pistons, achieving the required increase of performance and compression. The steam supply for pump operation is controlled by a flat sliding distribution valve, as shown in figure 7.2.

The main piston pumps in maritime applications are:

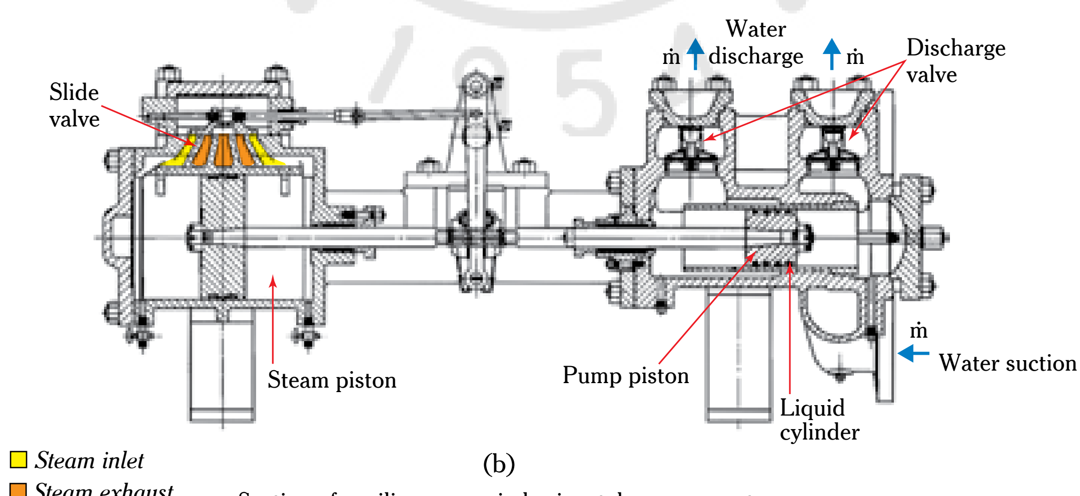
1) The simple acting pump with three rows of valves, located on the piston (fig. 7.3).

In this pump, the liquid, under the piston reciprocal movement, enters into the cylinder through suction valves due to the vacuum which is created from the movement of the piston to TDC [fig. 7.3(a)]. Subsequently, as the piston moves towards the BDC, pushing the liquid that has filled the space under the piston, the liquid passes to the upper side through the valve [fig. 7.3(b)]. As the movement is repeated, the liquid located on the upper side of piston is pressed and discharged by the cylinder valves [fig. 7.3(c)], while a new liquid fills the cylinder through the valve [fig. 7.3(a)]. This process is repeated at each piston reciprocating movement.

The pumps of this type are used for extraction of condensate in boiler systems, and for pumping the amount of liquid remaining in cargo tanks when a level is low. The pistons rods are connected with the yoke, which transmits the motion to the steam distributor and causes the pistons to move alternately creating an almost constant discharge flow.



(a)
Auxiliary pump in horizontal arrangement



(b)
Section of auxiliary pump in horizontal arrangement

Fig. 7.2
Steam piston pump Worthington type.

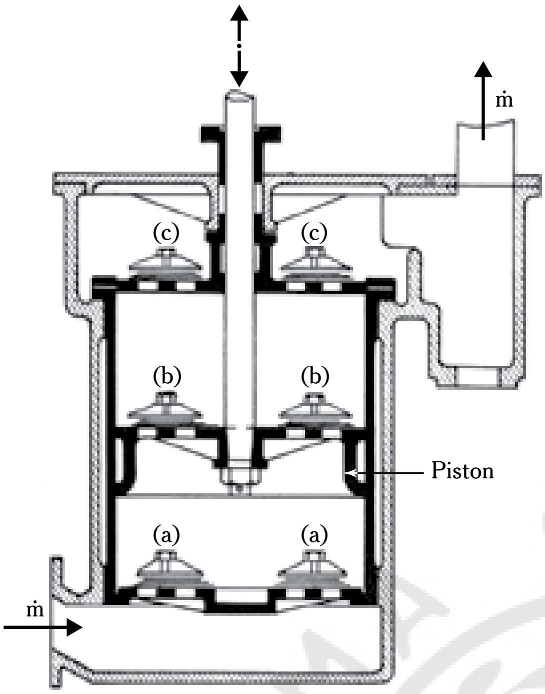


Fig. 7.3
Piston pump with three rows of valves on the piston.

2) Edwards type pump with ports (fig. 7.4).

The water enters into this pump due to the gravity in the cavity of the pump bottom. The piston head has the same conicity with the cylinder, so, as it moves to BDC, it pushes the liquid along the cavity of the cylinder to be ejected circumferentially and to enter over the top of the piston.

As the piston moves toward BDC, and reveals the ports of the cylinder, air is conveyed together with the water. The water together with the air is compressed by the piston and discharged from the discharge valve (d), which is positioned on the upper side of the cylinder. Edwards pumps have advantages compared to pumps manufactured with three rows of valves, because a possible damage is limited by the reduced number of valves, while the control, as well as the maintenance are easier.

3) Dual air pump or Weir type twin air pump.

The dual air pump is mainly found in older steam powered ships plants, serving systems of high vacuum requirements. It consists of two separate pump units,

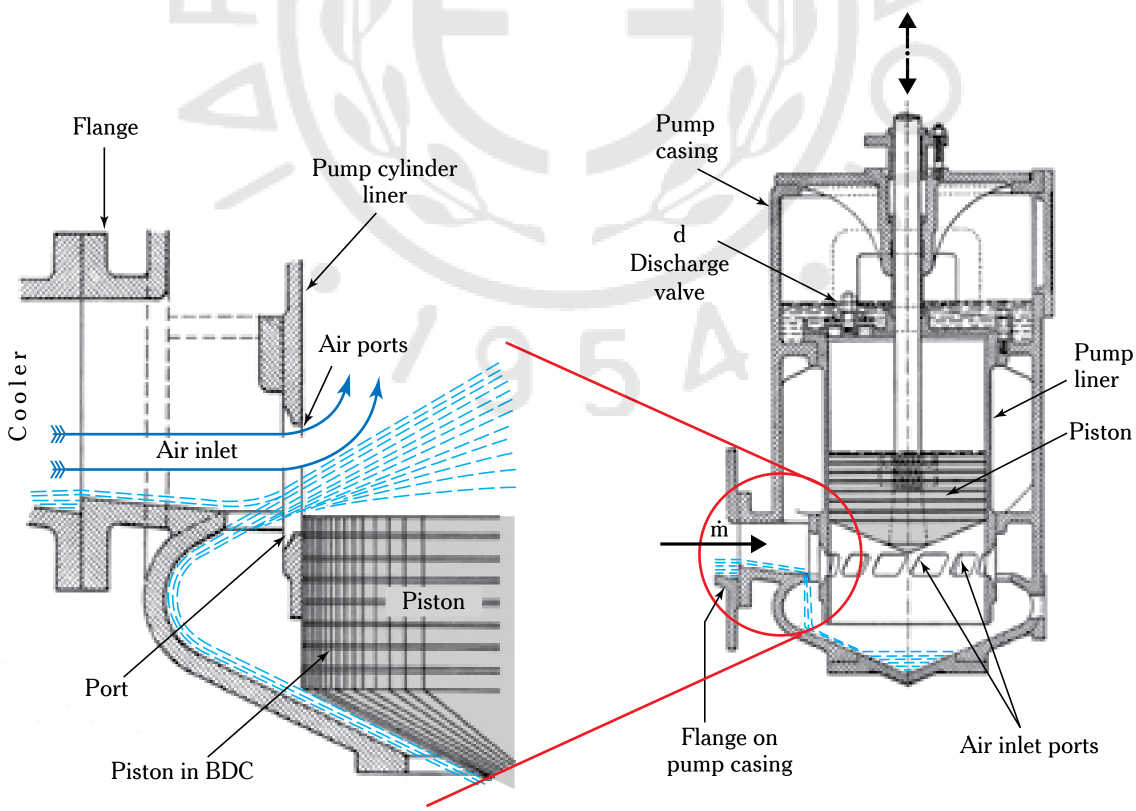


Fig. 7.4
Edwards type pump.

pump A, which is called dry, and pump B, which is called liquid pump. Pump A is used for the suction of air from the coldest part of the refrigerator, and pump B is used for the suction of condensates.

The pump is illustrated in figure 7.5, where the two pumps are shown in parallel arrangement. The movement of the pump piston is achieved by the steam cylinder, with the piston rod being connected to the piston of the liquid pump, while the same rod of the steam piston, through the yoke, moves the dry pump, and the two pistons in the pumps are moving in opposite direction.

The liquid pump (B) sucks the condensate and part of the air through the pipe (C), while the dry pump (A) only sucks air and non-condensed vapors from the highest point through the conduit (D). The entire facility is equipped with a cooler that uses sea water as cooling medium.

In order to increase the yield of the system, the air that is sucked is mixed with water from the concentrate which the pump itself draws, and injected into the suction of the dry pump through the injection tube. This concentrate is cooled at the condensate cooler by sea water and then sucked by the dry pump. The mixture of air and condensate is passed again, through the injection pipe, to the cooler, where it is cooled, creating a continuous recirculation.

The excess of condensate and air that accumulates during the pump operation is guided through the tube and the discharge valve to the top of the liquid pump piston. Then it is discharged (P) with the remaining condensate, which is pumped from the liquid pump to the cascade or feed water tank. From the cascade tank the feed pump sends the water to the boiler, where it evaporates again.

On the cylinders of the pumps are placed pressure relief valves, which remain closed in the dry air pump, whereas, in the liquid pump, they are opened in an emergency, in order to avoid the knocks. In this pump, lower temperature of air and concentrate mixture (that is sent to the storage tank or water supply tank) is achieved, due to its recirculation and the cooling from the cooler. The steam cylinder that drives the pump is the same to that in the Weir pump, and when an educator is added on the device, the system is called ***Weir-Paragon jet pump***.

The use of direct acting piston pumps allows the pressure and flow of liquid to occur at any point within the limits specified by the manufacturer. This is achieved by limiting the steam supply from the manual control valve, or through an automatic control device. The maximum pump movement speed

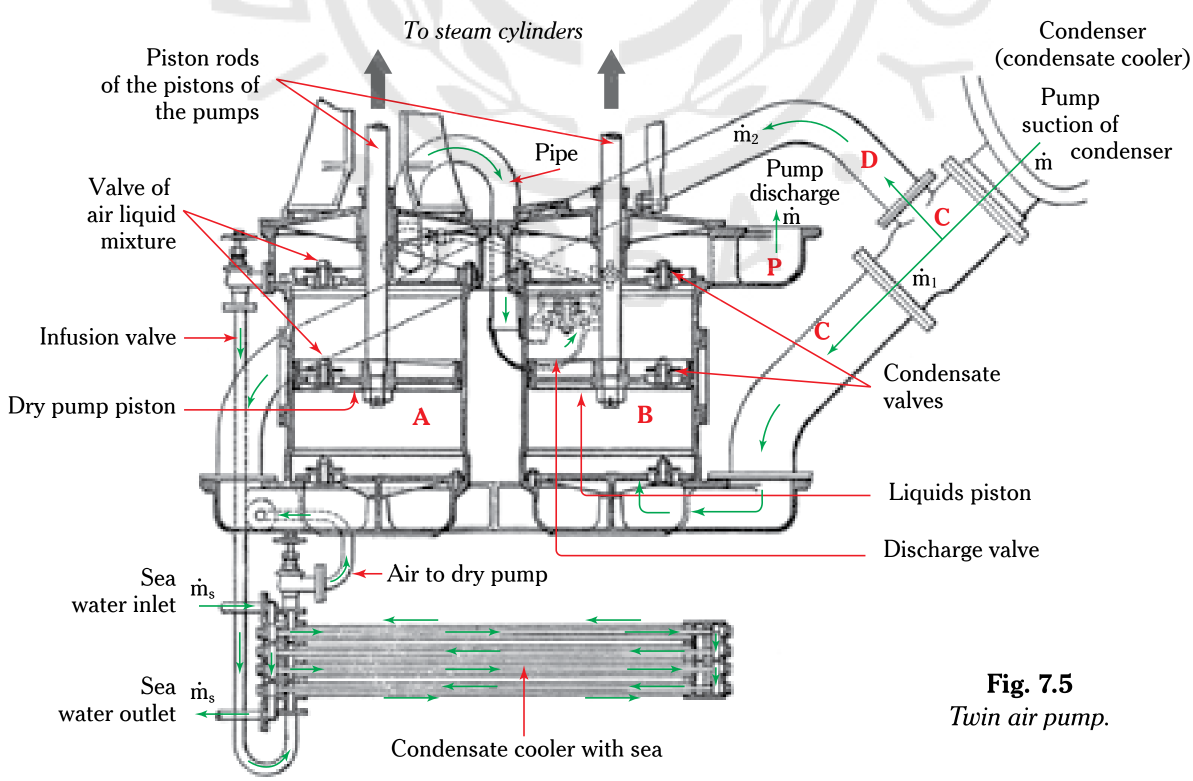


Fig. 7.5
Twin air pump.

depends on the frequency with which the valves open and close smoothly, while the liquid flow may range from the minimum to the maximum rating of the nominal pressure rate of the piston in the cylinder of the liquid. This value is limited by the available steam pressure or the differential ratio cross section of the steam piston with the piston in the pump cylinder, which distributes the liquid.

7.2 Dynamic pumps that are used on ships.

The diversity of functions on a ship allows the distinction of dynamic pumps into various groups, such as the following:

1) Pumps that are used to serve the function of propulsion machinery, depending on the installed, power unit, e.g. for steam engines or ICE.

2) Pump specialized installations, e.g. the pumps for the fresh water generator or the transfer pumps, etc.

3) Pumps for bilge pumping, and ballast pumps.

4) General Service pumps, and hygiene service pumps for the crew and passengers, etc.

5) Booster pumps.

6) Cargo oil pumps on tankers.

Irrespective of the construction characteristics of dynamical pumps and their classification (page 6), these pumps are designed to transport liquids with high or low temperature, which may have high corrosivity, while there may be suspended solids transferred in the liquid. For these reasons, the materials that are used for the construction of the pumps must match up accordingly, in each case.

These are:

1) For the pumps that are used for the cooling water of main engines and diesel generators, at the fresh-drinking water system, the casing is made of high quality cast iron, the impeller is made of copper alloy and the shaft of stainless steel, for higher corrosion resistance.

2) For sea water pumps, through which passes not only oceanic water but also sea water from coastal areas, as well as river water, the construction material of their casing is brass, the impeller is made of copper alloy and aluminum, and the shaft of stainless steel. When sealing is performed with gland packings or when mechanical seals are being used, the shaft is manufactured from stainless steel alloys, which confer such properties as not to counteract with the sealing material, and to withstand friction and temperature, when they are developed.

3) In the feed water pumps of boilers, due to their operation at high pressures and temperatures, cast iron is used for the shell, while the shaft and the rotor are made of stainless steel.

4) On cargo pumps, depending on the type of ship, there may be used:

a) Brass for pumps in M/T.

b) Stainless steel for the shell, the impeller and shaft, suitable to withstand corrosion caused by most chemical cargoes to chemical ships.

c) Alloy of nickel and steel for low temperature pumps in Liquefied Gas Carriers.

Also, ***the pumps should be designed*** in a way that:

1) Resists the dynamic loads that are generated by ship movements, because they operate on a moving platform where, probably, there are vibrations.

2) Makes them withstand the hot environment of the engine room, which may be simultaneously damp and corrosive because of the moisture carried by sea air through the ventilation ducts.

3) Makes them suitable for operation with variable flow rate, in accordance to the different conditions prevailing in port or at sea or during ship discharging (e.g. the changes in the ship trim, operation during heavy ship rolling at high seas).

4) Minimizes their size and weight; this is an important factor because of the limitations of installation space. For this reason, the pumps are placed vertically, while their support base is larger than those on shore.

7.2.1 Single-stage, single-suction centrifugal pumps.

The single-stage, single-suction pumps are constructed with a horizontal or vertical arrangement of their axis and a flow rate that ranges from 100 m^3/h to 7 000 m^3/h. They have a high manometric head and their speed depends on the supply requirements and the height of discharge.

Depending on their size and manufacturer, they are designed with axially split casing, which can be single or double spiral shell (fig. 3.8). In smaller size pumps, the shell may be integral around the impeller, and a cover is applied with screws on the upper side of the shell, permitting disassembly for controlling the impeller and the interior. On horizontal pumps, flanges are located on the lower half-shell, where the suction and discharge pipes are connected, and it is installed firmly on the supporting base. On vertical pumps, the flanges of the pipe connections, respectively, are at the side of the stable half-shell, in order to facilitate disassembly for inspection or repair.

The impeller drive shaft is connected to the shaft of the drive machine (usually an electric motor) by a coupling that is consisted of two parts. The one part is applied on the pump shaft, while the other on the motor shaft. The connection of the two parts is carried out by screws and nuts. At the points where the coupling screws pass from, there are properly shaped rubber rings around the screws, through which the smooth start and transmission of movement from one part of the coupling to the other is achieved, and, thus, from the motor to the pump shaft.

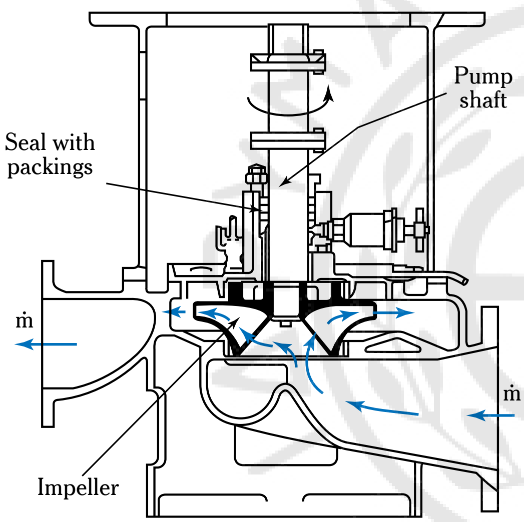
The support, as well the alignment of rotation shaft of the impeller is performed by ballbearings at the side of the drive machine (fig. 7.6). In pumps with long rotation shaft, the support may be achieved by two ball bearings at an appropriately shaped housing over or on the upper pump casing (bearing housing) (fig. 7.7).

The sealing of the shaft at the output from the pump casing is carried out with a mechanical seal or with soft seal material packings into the sealing stuffing box, suitable to the liquid flowing into the pump and its temperature.

The impeller may be of a closed type with one side suction, but to achieve high flow, or due to manufacturing needs, the impeller can be double suction, to reduce the axial thrust (fig. 7.8).

The seal of the impeller with the housing, on the separation of the suction from the discharge, is effected by suitable rings that are called mouth or wear rings, leaving a small gap which limits the amount of the losses.

The rings fit with great accuracy and are placed in the casing ***either by being pressed on a suitably formed position, or by means of metal pins keeping their position constant, or by being***



Section

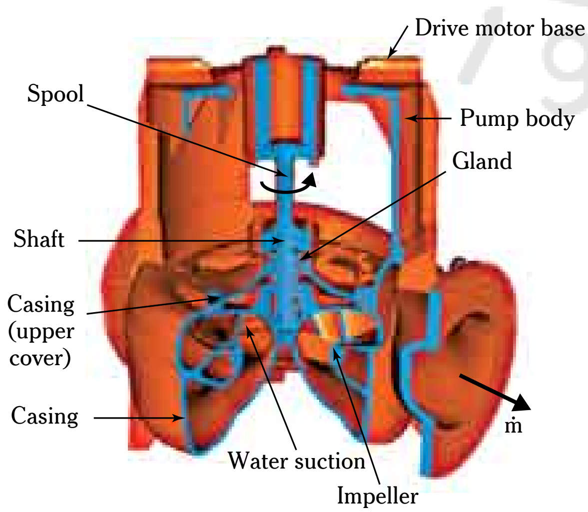


Fig. 7.6
Single-stage centrifugal pump.

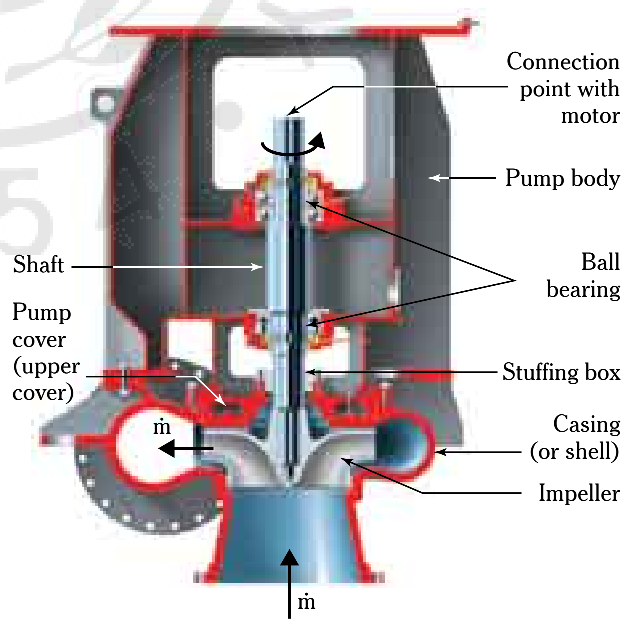


Fig. 7.7
Vertical pump supported by ball bearings at two points.

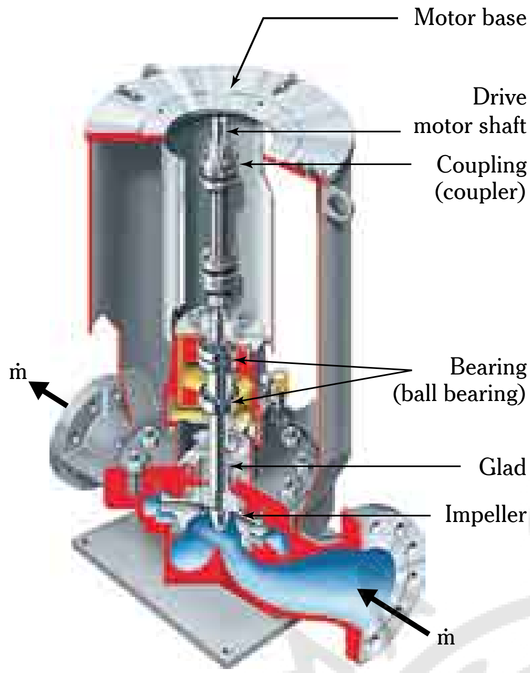


Fig. 7.8
Single-stage centrifugal pump with double suction impeller.

secured by small screws (only threaded). The mouth rings must be replaced when wear exceeds the limits set by the manufacturer, because they contribute to the efficiency and smooth functioning of the pumps.

Single-stage centrifugal pumps can be used as sea or fresh water pumps, fuel and lube oil pumps, as well as cooling pumps for the main engine (jacket cooling pumps), for cooling the pistons (piston cooling pumps), for sea water circulation in ship piping systems (sea water pumps), for the transfer and unloading of oil cargos (transfer and cargo oil pumps), etc.

If the ends of the impeller shaft rotation penetrate the casing and protrude in both sides, the rotor support is achieved by ball bearings on both outputs of the pump shaft. The layout may be vertical or horizontal and, respectively, these pumps are called vertical or horizontal between-bearings centrifugal pumps.

7.2.2 Multi-stage centrifugal pumps with electric motor.

These pumps are usually placed in horizontal arrangement, with the pump and the electric motor on a common base. They are used as boiler feed water pumps and are composed by many impellers which are placed on the same shaft. The number of impellers depends on the desired flow rate and discharge pressure.

The method of construction of multi-stage pumps may vary, but essentially the liquid flow, for increasing the pressure and flow rate, follows the same procedure. Hence, the pump may:

1) Be composed of many annular disks, equal to the number of the impellers. In this way, stages are created, between which the liquid passes alternately from the discs and the impellers. The discs of the stages are retained either by tightening bolts (and the whole system is placed between half-shells, one for suction and one for discharge, wherein the pipes are connected), or the annular discs are surrounded by a casing that keeps them together, and within it the stages of liquid flow are created (fig. 7.9).

2) The casing may be divided axially into two half-shells in the plane of its shaft. Each half-shell is formed with flow diaphragms which actually form the chambers containing the closed type impellers. The liquid creates the flow from one stage to the next, where gradually the pressure is increased as it passes alternately from the chambers and the impellers (fig. 7.10).

Due to the number of impellers the shaft has a long length and exits from both sides of the casing. The support of the shaft on each end is achieved with ball bearings, while sealing of the exit points of the shell is realized by glands with gland packings of high quality and durability, or with mechanical seals.

The reduction of axial thrusts, that are developed during the operation of the pump, depending on the manufacturer as well as the size of the pump, is realized either by disc springs, or by hydraulic systems. The disc springs, depending on their settings, receive the load and determine the position of the shaft. In hydraulic systems, an appropriate hydraulic balancing device maintains the position of the shaft in the appropriate position at all operating loads.

In this type of pumps, it is very important that the discharge pressure reaches the desired operating level in a short time, to reduce the wear and fatigue of the mounting points. For this purpose, a non-return valve is placed on the discharge, in order to ensure that the pump achieves the load which is set for its operation in a short time. The non-return valve in some valves operates under the influence of spring tension.

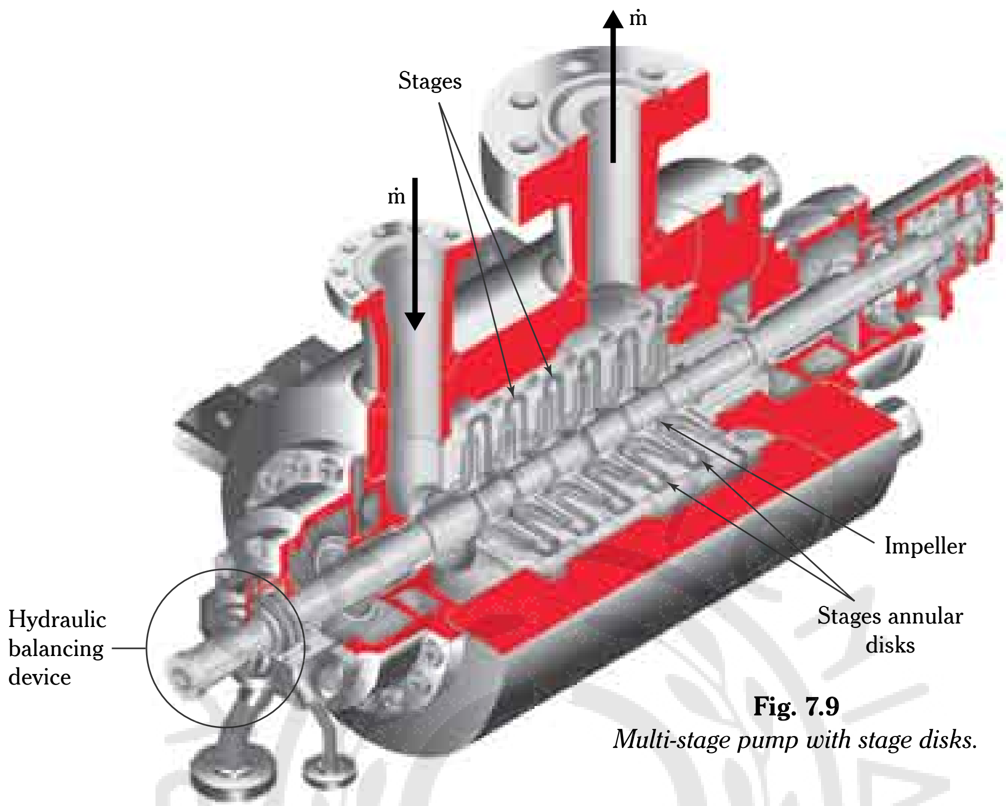


Fig. 7.9
Multi-stage pump with stage disks.

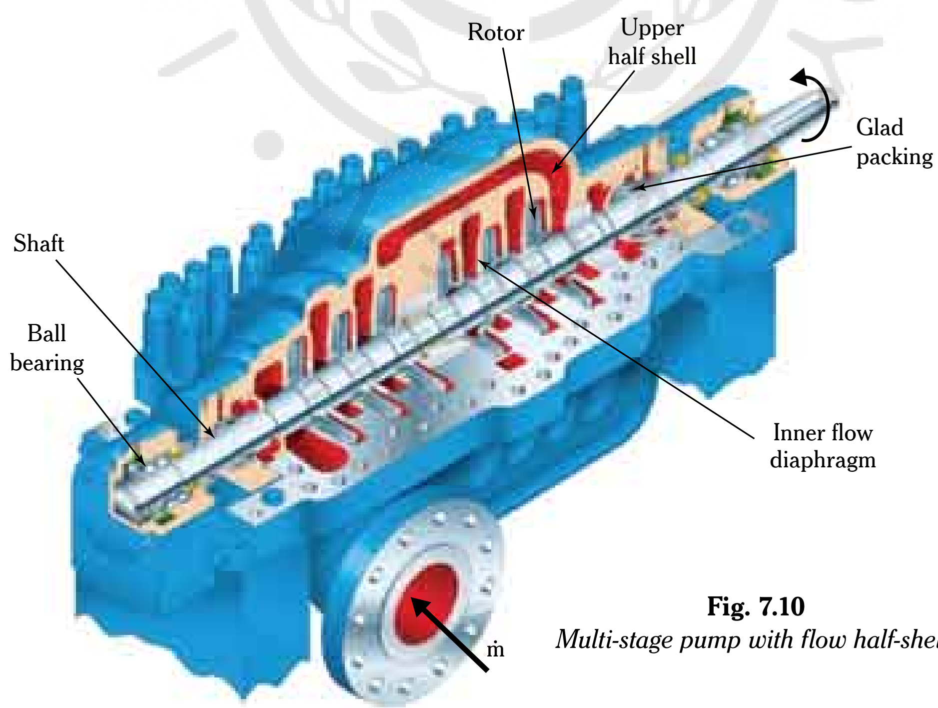


Fig. 7.10
Multi-stage pump with flow half-shell.

To protect the pump when the suction pressure in the first stage falls below a predetermined level, a pressure switch[1] is positioned, preventing the abnormal pump operation in conditions that can cause cavitation to the shell surfaces of the impeller.

7.2.3 Steam turbine driven centrifugal pumps.

Centrifugal pumps are usually driven by an electric motor, but, in the case where there is available power from steam production, or high discharge flow rate is required, a steam turbine is then used as a means of power. In this case, they are called steam turbine driven pumps.

The performance and reliability of a centrifugal pump that is powered by a steam turbine providing constant flow makes it ideal to cover the needs of a boiler for water.

With these pumps, the ability of the boiler to store satisfactory quantity of feed water, compared to the quantity that is required for steam production, is met. At the same time, their installation at the boiler feed piping systems enhances the efficiency of the system, utilizing the exhaust steam of the pump for the heating of feed water.

The ability of the centrifugal pumps to develop high flow is utilized at the unloading of tankers, where it is needed to face high volume cargo.

Steam turbine driven pumps can be either oil lubricated or water lubricated.

The following three are characteristic types of these pumps:

1) Steam turbine pump, oil lubricated.

One of the main parts this pump is consisted of is the impulse turbine which has three speed gradations, with its shaft extended so as to transfer directly the rotational movement to the impeller of the single stage centrifugal pump (figs 7.11 and 7.12).

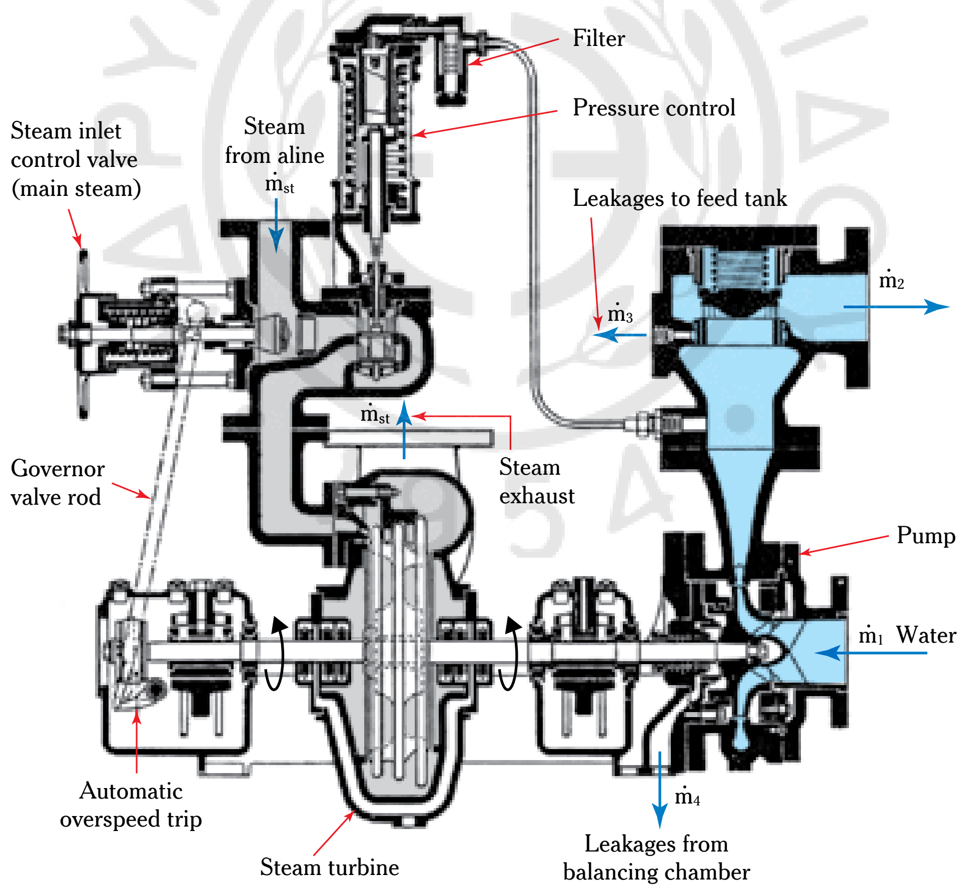


Fig. 7.11
Steam turbine oil lubricated pump.

[1] The ***pressure switch*** is a device which controls and maintains the pressure to a desired level through an actuator.

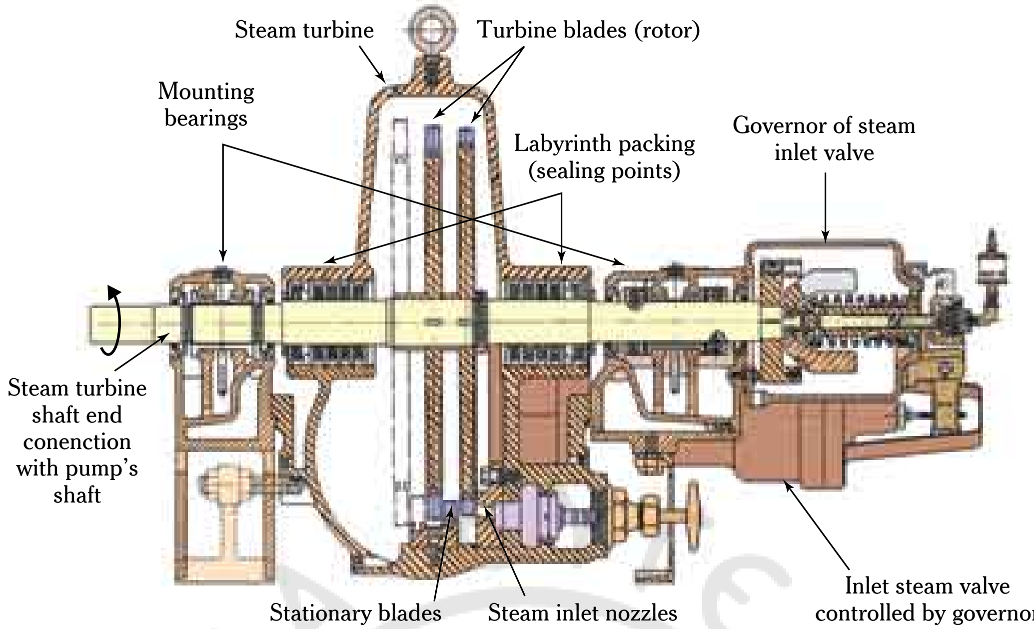


Fig. 7.12
Parts of steam turbine of oil lubricated pump.

The support of its shaft is carried out on bearings and lubrication is made with oil.

The steam is inserted in the turbine through a permanently open main steam valve. Next, through a dual mounting valve, it goes through fixed nozzles that are located on the casing, which guide the steam to the driven vanes for the turbine rotation. The opening or the closure of the dual mounting valve is controlled from its rod, which is connected to a pressure regulator.

The regulator is consisted of a cylinder with a piston, with the water pressure discharge of the pump applying on its top side, while a spring applies on its bottom side. Therefore, the quantity of the steam which is inserted in the steam turbine depends on the water pressure discharge of the pump. Furthermore, the ability is given to check the position of the double seat valve by adjusting the volume of the spring with a regulatory screw which applies to the piston of the cylinder. If the regulator is completely opened and the steam for boiler feeding is not sufficient, the auxiliary nozzles fitted on the casing of the turbine are opened. Thus, the pressure regulator provides a stabilized operation according to the consumption needs.

The rotating speed of the common shaft of the turbine and the pump is regulated by an overspeed trip, which protects it from failures-malfunctions that the development of overspeed could cause (fig. 7.13). This switch is consisted of an unbalanced disc, whose weight is eccentrically distributed, while, with the assistance of an internal spring and a guiding pin, it remains concentric to the shaft. When the rotating speed reaches the predefined volume limit which is regulated from the internal spring, its connection rod to the main steam valve is pushed, releasing the spring. Then, instantaneously, the rod is released and closes the main steam valve, which interrupts the flow of operating steam to the steam turbine. To restart the pump, the control rod of the main steam valve must be "armed" by pressing the spring, so that dragging back the valve will bring it again to the open position.

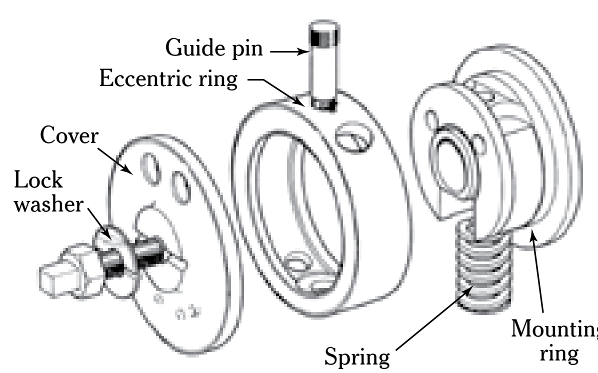


Fig. 7.13
Overspeed trip mechanism.

2) Steam turbine pump, water lubricated.

The successful development of water lubricated bearings leads to the development of a more modern form of Weir type turbine pump, allowing the construction of a unit with the turbine and the pump in a common shell [fig. 7.14(b)]. It is an integral two-stage turbine-driven unit, with the impulse turbine and the pump impeller mounted on the same axis,

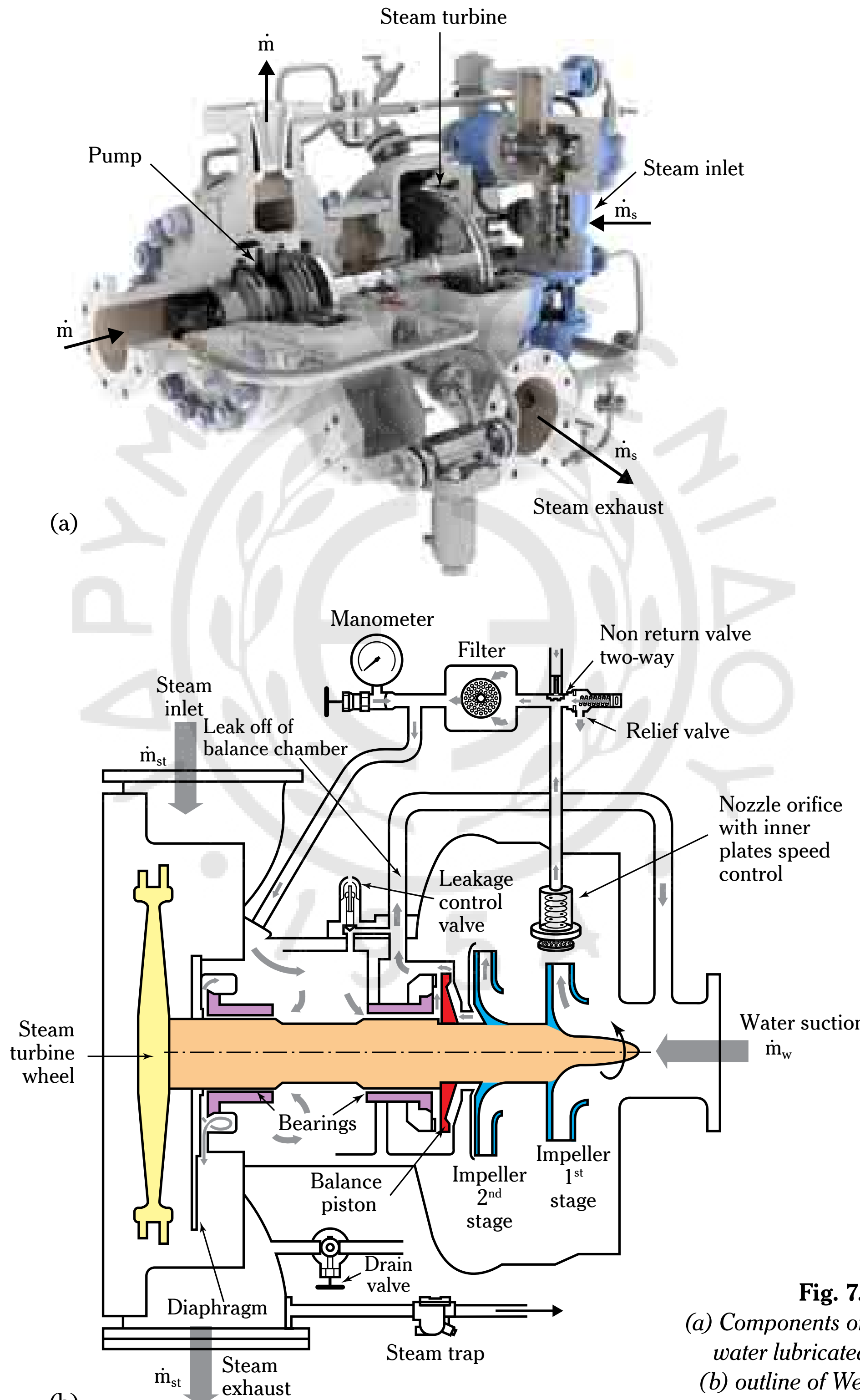


Fig. 7.14
(a) Components of steam turbine water lubricated pump and (b) outline of Weir type steam turbine oil lubricated.

without external bearings at the ends of the shaft. The rotor support is achieved with intermediate bearings.

The rotor of the turbine is made of forged steel, and is connected to the pump shaft via radial toothing association, with special design to allow free radial expansion and simultaneously to maintain its alignment. This connection prevents the transfer of heat to the bearing of the pump shaft, acting as a barrier. Moreover, it facilitates disassembly of the pump for repairs, without affecting the ***dynamic balancing***[2] of the machine.

The side of the pump consists of two single suction impellers, which are dynamically balanced and made of polished stainless steel. The diffusers and the annular shell discharge nozzles are also made of stainless steel.

The balance of the rotating system, under the influence of the steam impulse and the action of the hydraulic thrust, is achieved with accuracy by an automatic controlling device with a balancing piston, which rotates with the shaft.

The water lubrication of the bearings, during the operation, is effected as water is passing through an orifice consisting of ***several barrier diaphragms***, located in the discharge of the first stage of the pump impeller. This water flows through a two-way valve and filter toward the bearings, while the pressure increase is prevented by a leak control safety valve, which is installed in the water pipe, toward the side of bearings. Then, the water passing through the bearings is mixed with the steam at the turbine outlet and escapes with the steam of the turbine exhaust, while leakages from the chamber, where the balancing piston is, are returning to the pump outlet.

During pump starting, stopping, or during standby, the water is supplied to the bearings from an external source, usually it is the concentrate pump in the vapor return system through the two-way check valve, which is connected to the water pipe to the bearings, as illustrated in the water lubricated system of figure 7.14.

The resulting wear of the bearings, over time, causes the passage of greater quantity of water through the interstices between the bearing and the shaft, which gradually leads to pressure drop of lubricating water within the central chamber. To address the lubrication water pressure drop, the return valve of the leaks is adjusted towards the suction of the pump, so as to maintain the desired level.

3) Turbo pumps.

Turbine driven pumps, one of which is the Coffin pump, are single stage high speed centrifugal pumps and are used as feed water pumps for steam boilers (fig. 7.15). They are constructed so as to satisfy the following characteristics:

1) Compact construction with a small volume and weight.

2) Reliability for continuous operation without special supervision.

3) Simplicity in construction in order to facilitate periodic inspections and

4) High performance with low running costs.

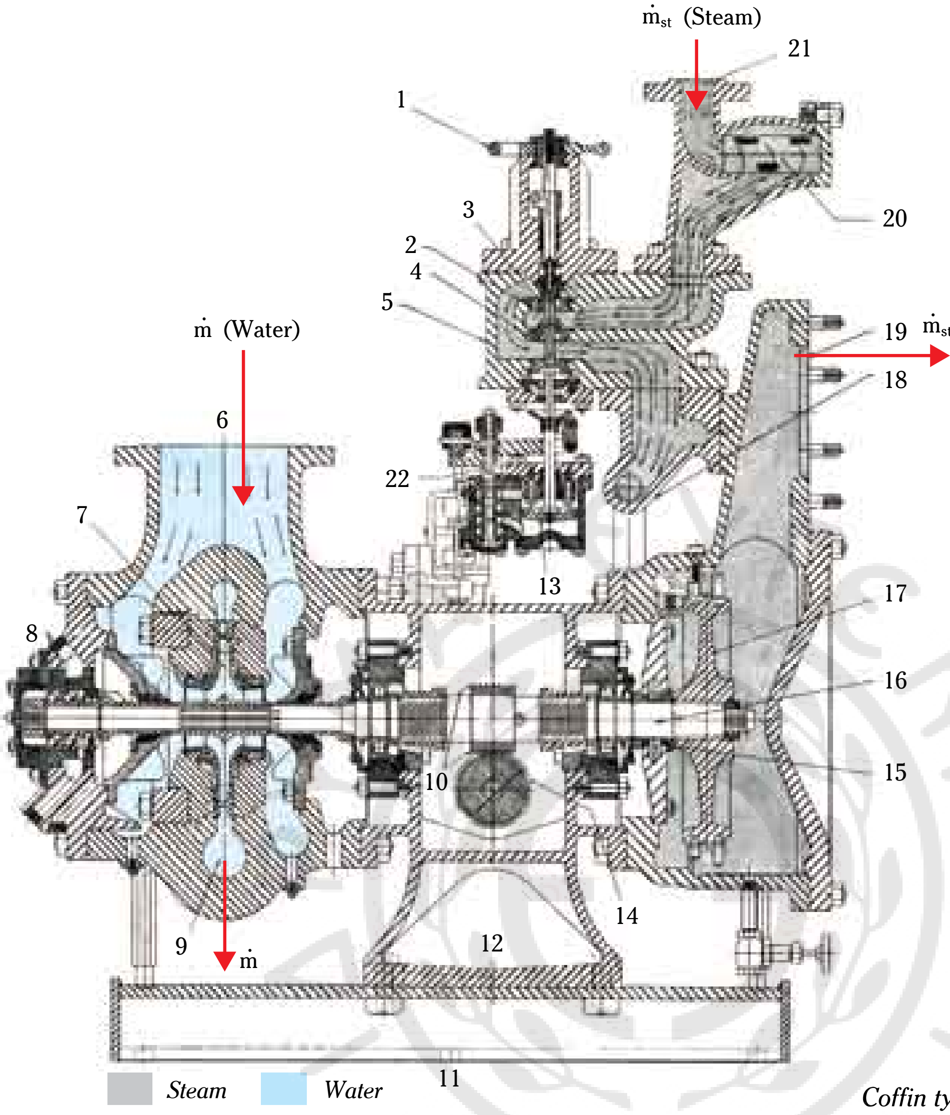
Turbine pumps may operate with steam temperature up to 520ºC and their rotation speed reaches 9 000 rpm, which is almost double than conventional pumps. The improvement of their performance is due to the reduction of mechanical losses, since the turbine and the impeller are mounted on a common shaft of stainless steel or Monel metal. The components, such as the turbine rotor, the nozzles, the casing of the turbine, the wear rings, the diaphragms for the steam introduction, are made of stainless steel, while the shell can be made of cast steel.

The rotor ***of the Coffin type turbine pump*** is designed so as to achieve perfect balance in axial thrusts. The impeller in the pump side is double suction, or single suction at smaller pumps, and has balancing holes that help to reduce the hydraulic thrust. The speed of the water, under the influence of the pump impeller, is converted to pressure in the diffuser, thus the speed in the volute casing is relatively small. The impermeability is ensured by specially designed seals.

The bearings of the turbine pump are cylindrical type bearings (cylindrical roller bearings), designed with precision, in order to be resistant to high loads and high rotation speeds. By these bearings, the concentric rotation of the axis is ensured, reducing the clearances of wear rings to the minimum.

The turbo pump has a thrust bearing, double

[2] Dynamic balance occurs when no centrifugal forces arise on a rotating mass, e.g. on a pump shaft with one impeller or with more impellers fitted on both ends. The process for successful dynamic balance of a rotating mass is called ***dynamic balancing***.



1. Governor steam valve.
2. Upper seat of the valve.
3. Steam control valve.
4. Lower seat of the valve.
5. Steam chest.
6. Pump suction.
7. Pump impeller.
8. Thrust bearing.
9. Pump discharge.
10. Interminable.
11. Base plate.
12. Oil reservoir.
13. Dash piston of oil motivated control system for steam valve.
14. Interminable wheel of oil control system of the pump.
15. Turbine shaft gland.
16. Shaft.
17. Turbine wheel.
18. Steam exhaust.
19. Steam inlet to turbine.
20. Steam strainer.
21. Steam inlet from line.
22. Oil control guide valve.

Fig. 7.15
Coffin type Turbine Driven Pumps.

ball bearing type, which is placed in the outer casing. This provides automatic adjustment of the axial position of all the moving parts, without having to use plug-ins or other regulatory instruments.

The lubrication of the turbine pumps is carried out by a gear pump, driven by an interminable screw mounted on the main shaft. The pressure of the lubricating oil is constant, regardless of the change in speed and the unit load, and it is achieved by controlling the automatic regulator and safety valve.

The turbine driven pump also has:

1) An ***oil motivated control system*** for the steam valve, comprising of a cylinder with piston and valve regulator. It has high power and sensitivity, in order to actuate the steam control valve and make it responsive to the impulses of the constant pressure regulator and the overspeed trip regulator.

2) A ***constant pressure regulator***, responsive to momentary pressures of the water discharge with the help of an oil-motivated control system for the steam valve, in order to maintain constant pressure of discharge, regardless of the pump supply.

3) A ***centrifugal type overspeed trip regulator*** for the protection of the pump turbine, which automatically intervenes at a predetermined speed. In case of suction loss, the regulator automatically reduces the speed and restores the turbo pump to normal operation when the suction is restored.

4) An ***excessive back pressure switch***, which automatically closes the inlet valve of the steam in the event of excessive pressure rise in the outlet of the turbine.

For the operation of the turbine pump, the steam is provided through the main steam valve. The pump control, after the passage of the steam from the main steam valve, is carried out by the control valve via the oil-motivated control system, so that the steam valve itself can be used to handle the whole machine through the fixed pressure regulator. At the same time, the control valve is used for the operation of the overspeed trip regulator, the back pressure switch, and the stopping of the turbine.

By the effect of the oil-motivated control system, during pump operation, the dash piston of the control system is in continuous motion, so as to be closing by the pressure of steam which enters the steam chest, in case of loss of pressure of the lubricating oil or excessive backpressure. The operation is repeated automatically as soon as the oil pressure is restored.

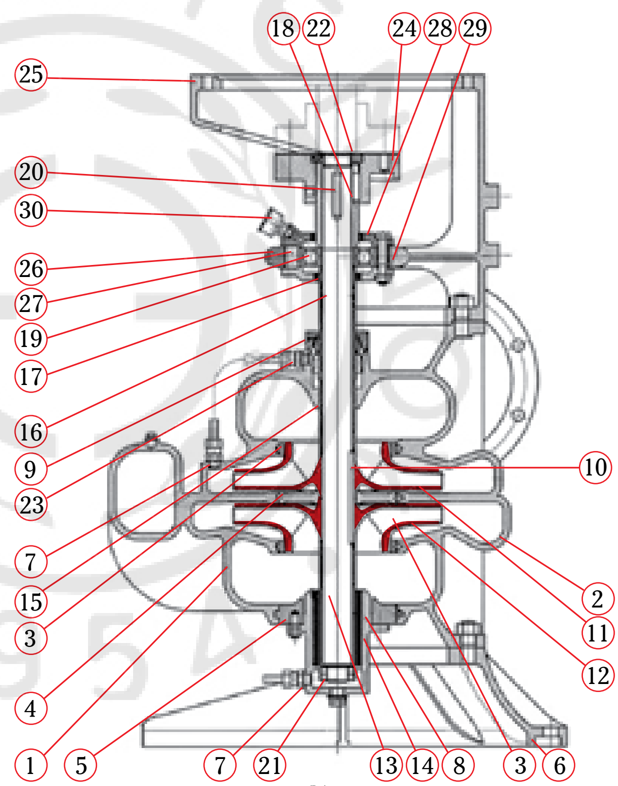
When the load is very low, or when the pump is stopped, a short circuit orifice is placed in the discharge pipe to allow the necessary recirculation of a quantity of water.

7.2.4 Two-stage vertical centrifugal pumps.

Centrifugal pumps with two pressure stages are vertically arranged; there may also be pumps of this type with more stages (fig. 7.16).

The pump suction is realized through a tube connected to the bottom of the tank or to intermediates of the suction pipeline, in order to ensure the continuous flow to the impeller of the first stage. When the pump is used for pumping condensate, the suction is connected to the bottom of the steam condenser.

(a)



(b)

1. Volute casing (upper side)
2. Volute casing (bottom side)
3. Wear Ring
4. Wear plate
5. Flange
6. Base of the pump
7. Pump bearings cooling pipe
8. Sleeve bearing
9. Cover of mechanical seal
10. Safety
11. 1st stage impeller
12. 2nd stage impeller
13. Shaft
14 - 18. Sleeves of shaft
19. Ball bearings
20. Wedge on shaft
21. Security nut
22. Nut on shaft for coupling
23 - 24. Mechanical seal
25. Base for driving machine
26. Bearing housing
27. Connection
28. Ring
29. Screws for bearing shell support
30. Greaser

Fig. 7.16
(a) Two-stage pump in arrangement, (b) parts of pump.

The motion of the pump is performed by connecting its shaft to an electric motor, and in some installations the motion is carried out by a turbine connected to the pump shaft through gear wheels. At the upper end, the pivot shaft is supported by ball bearings, which absorb the axial and radial thrusts of the pump, and, grease is used for their lubrication.

The impeller which is installed on the bottom of the shell constitutes the first pressure stage, and it is positioned with the suction hole of the liquid downwards. The second pressure stage impeller is positioned with the suction hole upwards, achieving balance in hydraulic pressure which comes from the two impellers reducing the axial thrust exerted on the support bearing of the shaft. Thus, through the passage of the liquid by the two impellers, high discharge pressure is achieved. For the cooling of the shaft gland, a pipe from the shell where the second rotor is installed sends liquid to the stuffing box wherein the sealing material is located.

When the two-stage pump is used to remove the condensate from the main steam return condenser, the impeller that is installed at the bottom of the shell, and constitutes the first level, is placed with the liquid suction hole (eye of impeller) upwards. Thus, it automatically eliminates any air amount entrained by the fluid sucked. For the extraction of air gathered, a tube is connected on the top of the impeller housing of the first stage with the top of the condenser shell. In this case, the impeller of the second level is mounted with the suction hole downwards, achieving a balance in the hydraulic pressure of the two impellers by reducing the axial thrust.

Between the first and the second stage, in the pump housing, a sealing ring is inserted, reducing the radial thrusts of the shaft, while the lubrication and cooling is carried out by water circulation. The sealing of the shaft is achieved either by gland and packings or by mechanical seal.

When, for the movement of the pump shaft, a steam turbine is used, lubrication of the bearings and transmission reducers is carried out with an attached pump that sucks oil from the sump of the gear unit of the turbine, and discharges it to the bearings. The system is also equipped with an electric speed regulator to regulate the pump rotation speed according to the load, maintaining the pump operation stable. In addition, the pump speed is controlled by an automatic overspeed trip switch.

– *Air removal from centrifugal pumps.*

The satisfactory operation of centrifugal pumps, according to their characteristics, is achieved by a suitable seal. If an air amount is entrained with the fluid, this has the effect of reducing pump performance. So, upon starting, the pump must be filled with liquid to create the initial suction. That is why the initial suction of centrifugal pumps is achieved when the pump is located below the surface level of the liquid to be pumped. Therefore, the centrifugal pumps that are used on ships are placed below sea level, while in other pumping systems they are installed with the suction positioned below the liquid level of the tank whose liquid is to be transported, ensuring initial suction.

A centrifugal pump cannot initiate suction with the presence of air, because the impeller does not create an airtight chamber in order to develop a vacuum, which would provide the pump the possibility to pump the liquid from the suction pipe. Also, the seal between suction and discharge is not sufficient, because of the gaps that are present to prevent contact between the rotor and the housing, during pump operation. The presence of liquid in the small gap between rotor and housing, in this case, can replace the absence of the sealing material, but this is not possible when there is only air inside the pump.

Despite the fact that pumps are usually installed below fluid level, the change in ship's draft can vary the suction head. This happens, for example, to the emergency pump when the ship is not loaded, or to the bilge pump suction that is necessarily located above the level of the liquid surface.

In this case, smaller attached positive displacement pumps are adapted on the main pump, which are called priming pumps, to remove the air from the suction and the pump housing, creating the necessary vacuum in order for the tube and the casing to be filled up with liquid. Only then is the impeller able to continue suction, and the centrifugal pump operates under normal conditions.

The priming pumps we meet onboard ships are as follows:

1) Piston priming pumps.

A priming piston pump [positive displacement priming (vacuum) pump] is connected to a single stage centrifugal electric pump (fig. 7.17). The motor rotates the main pump shaft, which, through an interminable screw, moves a wheel. Therein is keyed the crank shaft of the priming pump.

Immersion pistons are connected on the crank shaft of the priming pump. Then, as they reciprocate within the cylinder liners which are connected to the suction pipe, vacuum is created, which fills the pump with liquid. The air is discharged together with liquid in a separation chamber [fig. 7.17(b)] and through a needle valve, situated in the chamber, the air exits to the environment. The opening or closing of the valve is controlled by a spring float, and, as long as there is air in the valve, the chamber remains open.

However, when the liquid level in the chamber rises, the spring float, overcoming the spring tension, closes the needle valve. When the suction pipe and the main pump are full of liquid, the priming pump continues to rotate idle, so as to absorb the minimum power of the motor.

2) Rotary priming pumps.

A rotary priming (vacuum) pump may have a liquid piston or a vane (see par. 2.7, p. 31). In the liquid piston pump, the suction is performed from the main pump suction, while the discharge is performed in a separation chamber, as in the piston priming pump. In the rotary priming pump, the connection for transmitting the motion from the shaft of the centrifugal pump is carried out via a wheel coated with a synthetic anti-slip material (friction lining) at the contact points of the driveline disk with the disk that exists on the drive shaft (fig. 7.18).

By starting the main pump, the wheel installed on the pump shaft starts to rotate, and the wheel (drive disk) of the priming pump drags the rota-

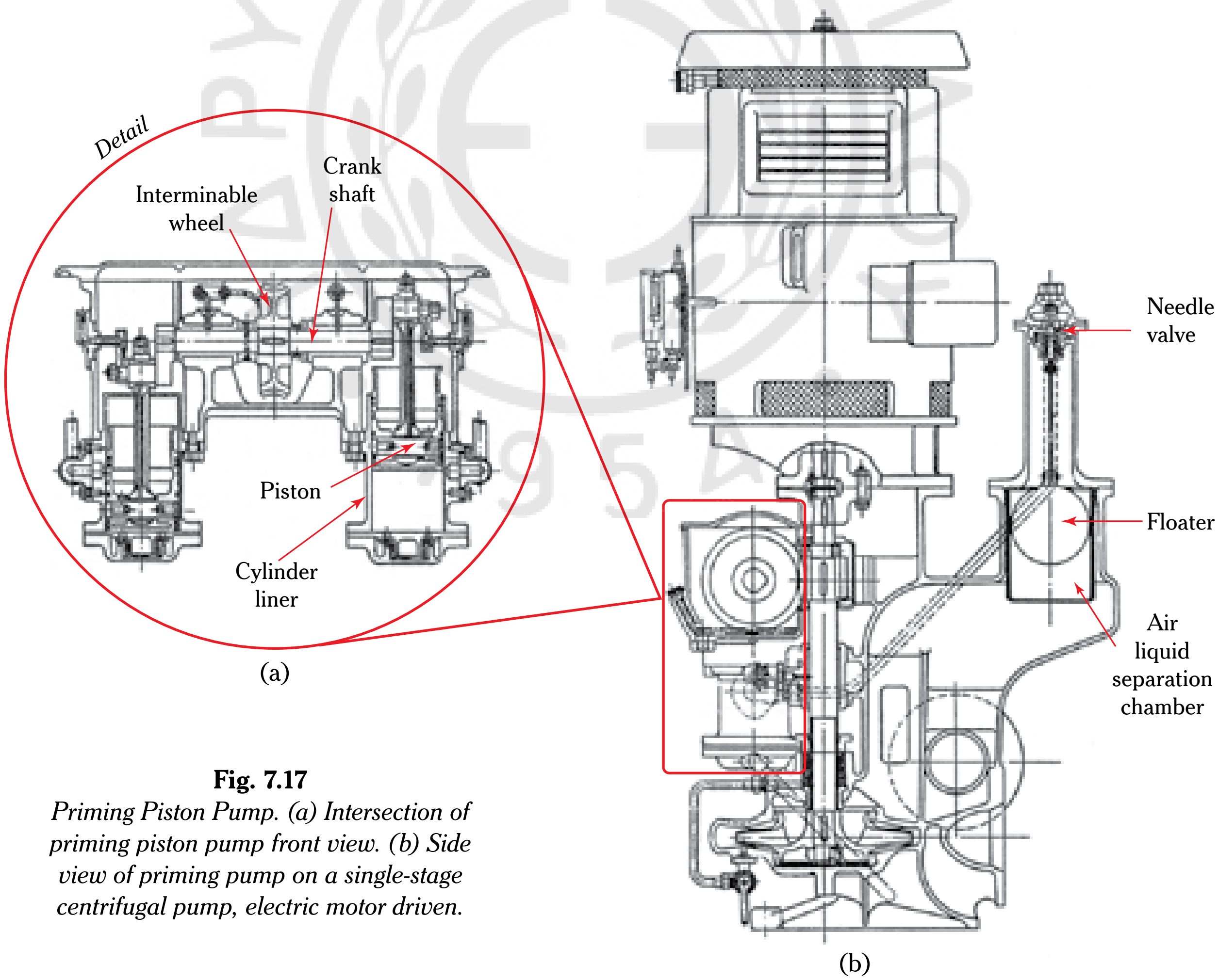
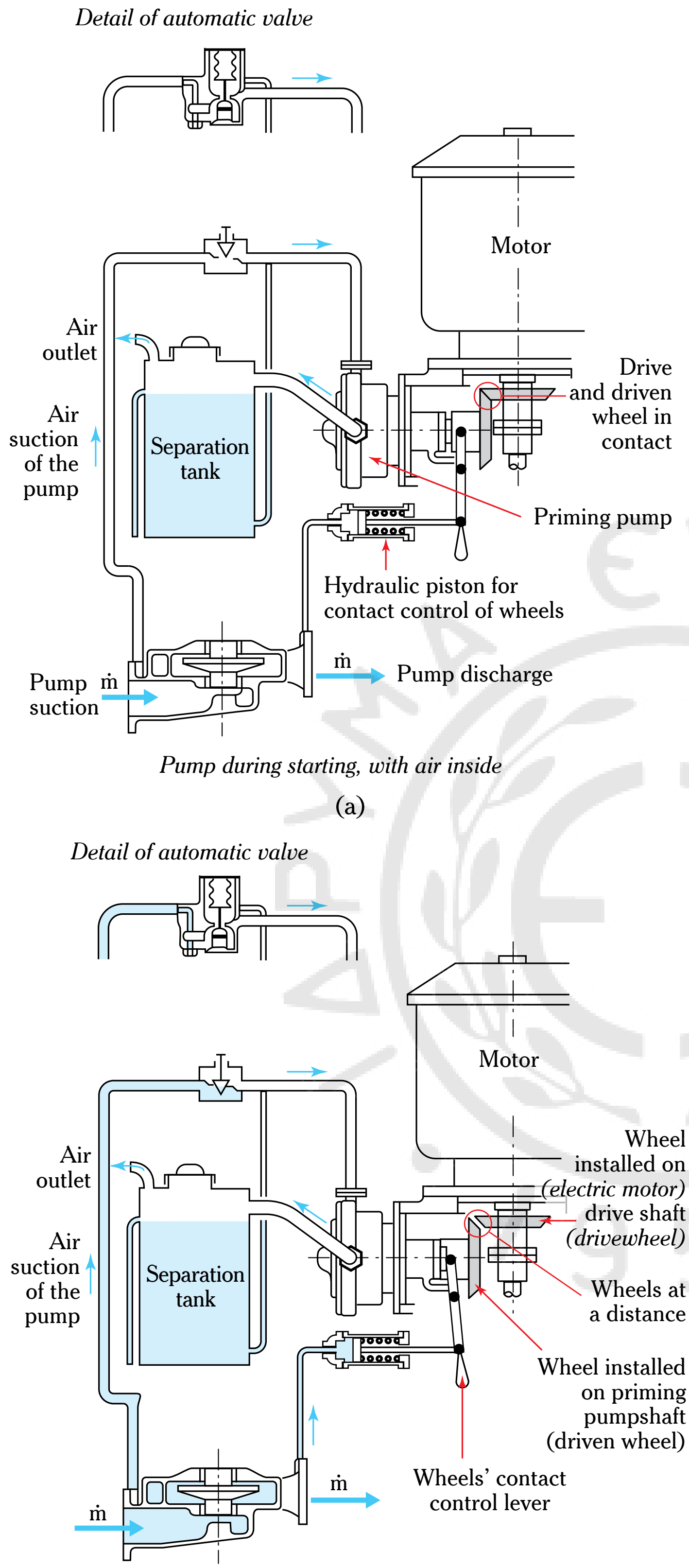


Fig. 7.17
Priming Piston Pump. (a) Intersection of priming piston pump front view. (b) Side view of priming pump on a single-stage centrifugal pump, electric motor driven.



Pump during starting, with air inside

(a)

Pump with water, after the removal of air

(b)

Fig. 7.18

Typical arrangement of a rotary priming pump established in a centrifugal pump (a) during starting and (b) during operation.

tion axis of the liquid piston pump. The contact of the wheel with the shaft of the centrifugal pump is controlled by a liquid piston, and the piston cylinder is connected to the pipe at the discharge of the centrifugal pump. When the entire amount of air is removed and the centrifugal pump creates pressure at the discharge, the piston cylinder is filled with fluid. Then, the piston rod, which is connected to the rotary wheel of the liquid piston pump, stops the priming pump by removing the wheel from the rotational axis of the main pump, and the centrifugal pump continues to function normally.

3) Priming ejectors.

The ejector as a priming pump is mounted on the pump shell (fig. 7.19) and the operating fluid is steam, water, or the air which is supplied to the ejector before starting the pump. The discharge of the ejector is carried out in the atmosphere or at the discharge pipe of the pump after the outlet valve, which should remain closed until the pump is filled. The filling is perceived by the water discharge at the outlet of the ejector. Then, the ejector is isolated and the pump starts, while, simultaneously, the discharge valve opens.

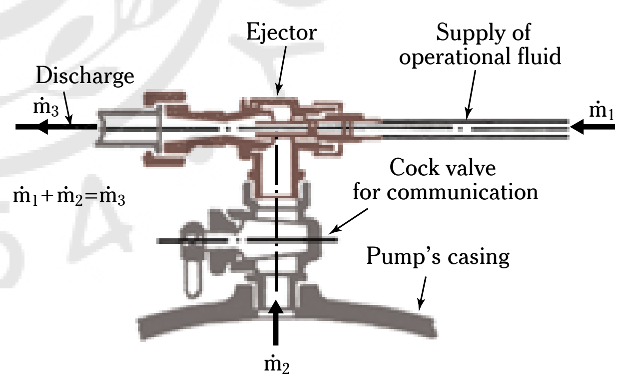


Fig. 7.19

Priming ejector on a pump casing.

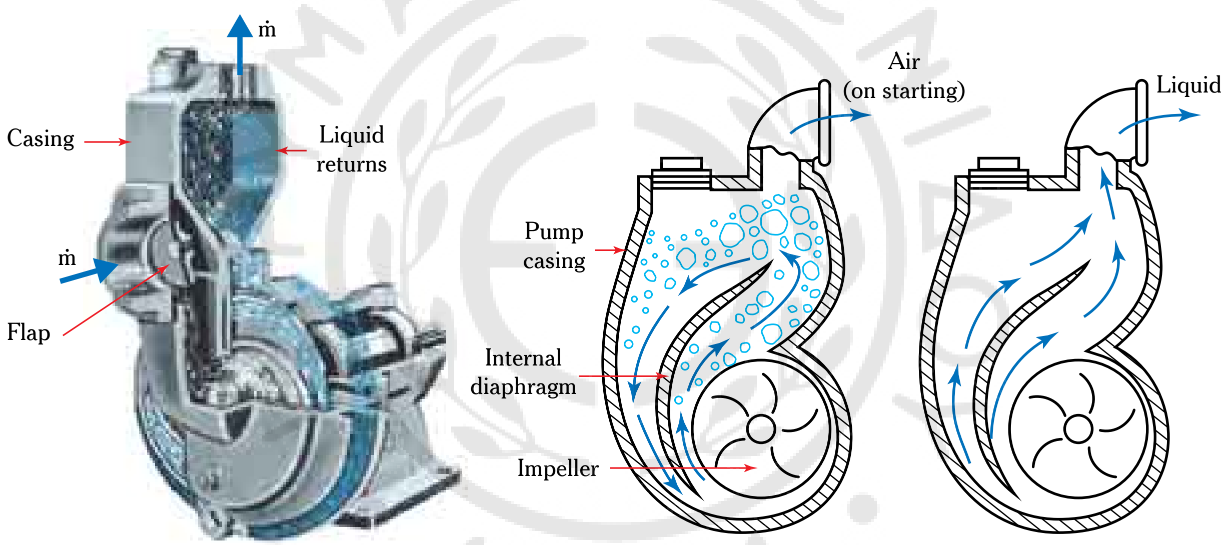
7.2.5 Self-priming or automatic filling centrifugal pumps.

In accordance to the structural characteristics of centrifugal pumps, as mentioned previously, air removal is necessary for the initial suction of the pumps, and should be performed by a priming pump. Yet, there is a type of pump that is manufactured with automatic filling or suction based on the "principle of diffusion", which eliminates the need of the installation of a priming pump.

Centrifugal self-priming pumps (fig. 7.20) have a special formation in the shell. They have enlarged suction, specially designed, with double casing which also consists the shell at the pump inlet, and a separation chamber at the discharge. In the suction, a non-return valve is installed (commonly flap), allowing flow only towards the pump rather than in the opposite direction. The pump is filled with liquid only during the initial starting, so that any time it subsequently stops, there is a quantity of liquid that is kept in the pump housing and in the separation chamber.

When the pump starts, and there is air in the suction pipe, the liquid that is already in the pump passes through the impeller, generating vacuum suction. Thus, the impeller discharges a mixture of liquid and air to the separation chamber. The air is separated there by gravity, and is driven to the discharge pipe, while the liquid is returned to the impeller through the diaphragm, or through an external recirculation pipe that leads to the impeller suction, where it is mixed with new air by the suction pipe. The process is repeated until all the air is removed and the pump is operating normally.

Of course, because of the recirculation, an amount of fluid remains throughout the operation of the pump, and this decreases the pump efficiency. This is the reason why these pumps are usually manufactured with low power and are used for pumping bilges.



Flow of liquid and air to the pump

Initial operation. Section of the pump where liquid air is discharged

Pump in operation. Section of the pump where only water flows

Fig. 7.20
Self-priming centrifugal pump.

CHAPTER EIGHT
Pumps in ship propulsion systems

8.1 Introduction.

The following chapter presents the pumps commonly installed on ships which, depending on the ship's propulsion system, actually contribute to the smooth operation of the propulsion plant. Furthermore, the chapter describes the operational process they serve. Particular reference is made to tankers and liquefied gas carriers, due to the specificity of the pumps used for the completion of loading and unloading operations.

Hence, it is expected that pumps frequently used on ships are divided into several groups. These groups can include pumps associated with the operation of the vessel's propulsion system, the operation of the diesel generator that produces electricity, and the transfer of fluids to various piping auxiliary systems.

It is highlighted that in most cases the pumps installed for each operational process on ships are two. In spite of that, each pump seperately has been designed in order to be able to serve the relevant functions effectively. Furthermore, it is possible that for some applications, due to high requirements, three or more pumps are used (e.g., the cargo pumps of a tanker). The frequent installation of pumps in double number ensures two basic needs, which are:

1) ***The minimization of possibility to interrupt a function*** due to sudden failure of the operating pump, by providing, through the second pump, the safety that is required for the passengers, the ship, the cargo and the environment.

2) ***To provide the possibility of maintenance to a pump***, by isolating it from the system that it serves, without affecting the accomplishment of a process, the readiness or the safety of the vessel.

Typically, for all pumps there is a control mechanism of the pressure of the network in which they are installed. This mechanism activates the ***spare pump***[1] to start when the pressure in the system drops below the desired level, which the control mechanism is adjusted in. In addition, on all pumping systems, filters are installed (usually prior to the suction of the pump) which are suitable for the fluid that is transferred.

Because ship pumps must operate under particular conditions, they must be designed and constructed to withstand dynamic loads resulting from vessel motion (for example, pitching, rolling, trim and draft differences, and so on) as well as operation in a hot, humid, and potentially corrosive environment. Additionally, pumps must frequently be suitable to operate under a range of loads and at various flow rates in order to facilitate, immediately and safely, any changes in vessel operations (e.g. at full speed, or arrival to port and departure from port, where rapid changes in propulsion equipment load occur).

Furthermore, as is the case with all shipboard machinery due to space requirements, the minimization of pump size and weight is important. This happens when designing marine equipment that is either in the engine room or on deck. For this reason, many shipboard pumps are mounted vertically on a wide base compared to shore-side pump installations (fig. 8.1), whereas they are possibly mounted horizontally (fig. 8.2) if operating conditions require. Also, the length of the shaft that transmits motion from the motor driver is limited to the one absolutely necessary (fig. 8.3), while in small pumps it is possible for the motor and the pump to constitute one single system.

What follows is a description of the features typically incorporated into the designs of pumps used in shipboard applications. This aims to illustrate

[1] A spare pump is the one that is on stand by, ready to automatically replace the pump that has stopped working.

the distinguishing features of pumps that are used in selected applications, which can be found on most ship types. ***This information is general in nature, and may not apply in all cases, based on the requirements for specific installations or the preferences of vessel owners and ship designers.*** Thus, for each specific case the information about the system and the installed pump is provided by specific constructional designs which exist on all ships.

Fig. 8.1
Pumps in vertical arrangement in ship engine room.

8.2 Pump installations on board – depending on the ship's propulsion system.

8.2.1 Ships with steam turbine propulsion.

Usually, the pumps that serve the function of a propulsion system with steam turbine are the following six:

1) Main feed pumps of boilers.

These pumps are used for supplying water into steam boilers. In a typical ship system installation with steam boilers where the heat is produced by heavy oil combustion, the feed pump sucks water from the deaerating feed tank DFT (or cascade tank) and discharges it into the steam drum of the boiler(s) (see fig. 8.8, process 3, 4). In many cases, in the discharge of the feed water pump two separate pipe lines can be connected, one main and one auxiliary, which end at the steam water drum. The purpose of the auxiliary line is the immediate supply of water to the boiler in case of failure in the main line or in the automations that are installed on the piping system. Heat exchangers are also placed in the boiler network, for preheating feed water. The purpose of the heaters is to increase the thermal efficiency of the steam turbine plant.

The number of feed pumps used on the feed water system of the steam boiler is two or more, and they can operate alongside, if the boiler's needs during full load operation are high.

The typical configuration of feed pumps includes single or two stage centrifugal pumps and multistage pumps. These pumps can be moved by steam turbines or motors. We also often meet pumps manufactured with axially divided volute casing (fig. 8.4 and 8.5), as well as pumps with disk casing[2] with dif-

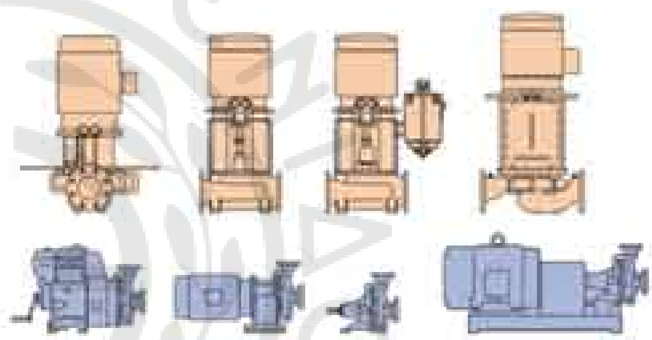


Fig. 8.2
Vertical and horizontal pumps arrangement.

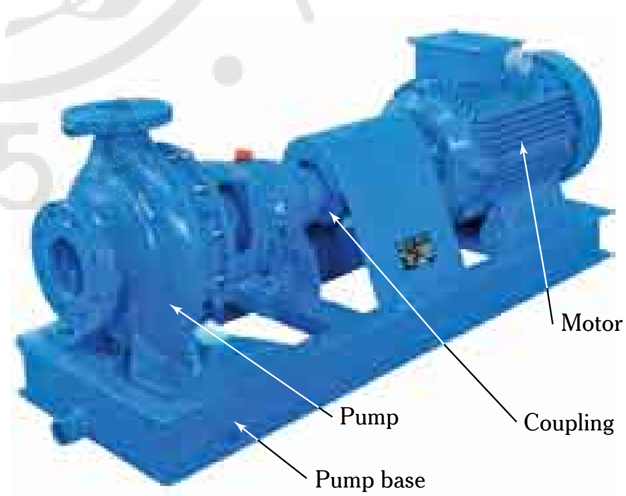


Fig. 8.3
Pump with motor on a single base.

[2] ***Disk casing*** is the casing consisting of disks in suitable configurations, which are arranged one next to another, forming the pump stages. The impellers are mounted between the disks.

fusers (fig. 8.6). The arrangement of the pumps that are moved by steam turbine is frequently horizontal, while those powered by a motor are used in horizontal as well as in vertical arrangement.

The sealing of the pumps, depending on their type and their construction specifications, is realized with packings which are restrained and regulated by a gland follower, or mechanical seal.

On steam turbine pumps, steam glands (labyrinths) are used, which ensure the sealing. As the steam flows from steam gland to steam gland, the pressure drops (deterioration) and, as a result, the final pressure in the last steam gland is equal to atmospheric pressure.

During feed pump operation at constant speed, the water supplied to the boiler is controlled locally by an automatic valve of feed water flow adjustment. When steam is used for the movement of the pump (turbine-pump), the steam supply control is achieved by a mechanism which adjusts the steam flow (the constant pressure governor). So, depending on the pump discharge pressure, the steam supply governor regulates the supply of steam to the steam turbine in order to reduce it when the discharge pressure is increased. Respectively, when the discharge pressure drops, the flow rate of the steam increases again. Hence, the pump operates at the desired pressure of the feed water. Changes of the steam boiler water requirements cause similar changes in the provision regulating valve.

The steam turbine that is used to drive the feed pump is protected by an over speed mechanism, which shuts it down. Also, for the efficient and safe operation of the pump, suction pressure control mechanisms are installed, because the suction pressure would cause cavitation of the pump surfaces, as well as a shutdown mechanism, in case of load loss on the pump (namely, operation without liquid, loss of flow).

To protect the network from excessive pressure, an expansion valve is installed at the discharge of the pump, which, when activated, leads the feedwater in the deaerating feed tank. The whole process is realized through a recirculation piping system. To prevent the increase of pressure from the recirculation to the DFT, the flow to the recirculation network is realized by a specific pipe where an orifice plate[3]

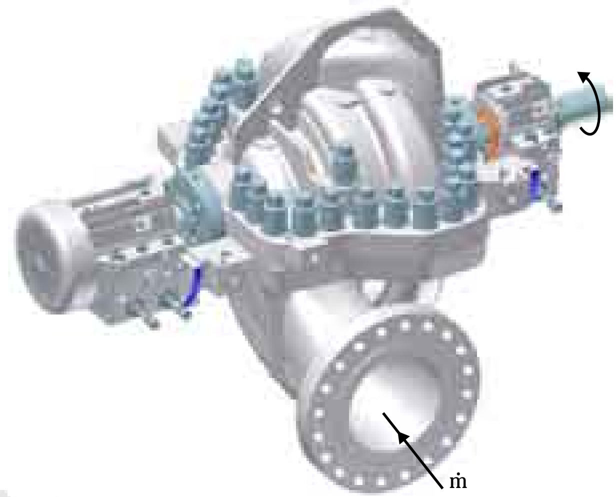


Fig. 8.4
Single-stage pump with axially split volute casing in horizontal arrangement.

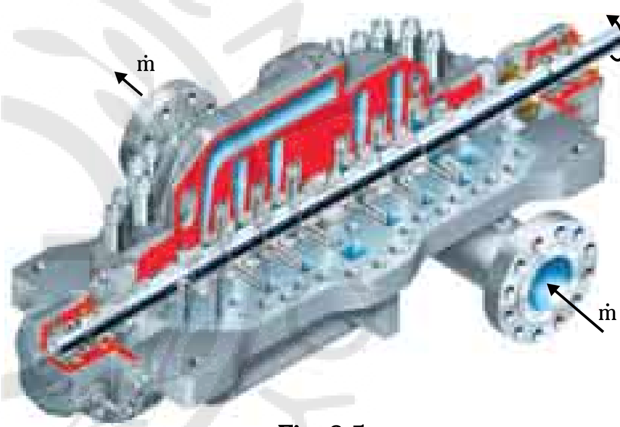


Fig. 8.5
Multi-stage pump with axially split volute casing in horizontal arrangement.

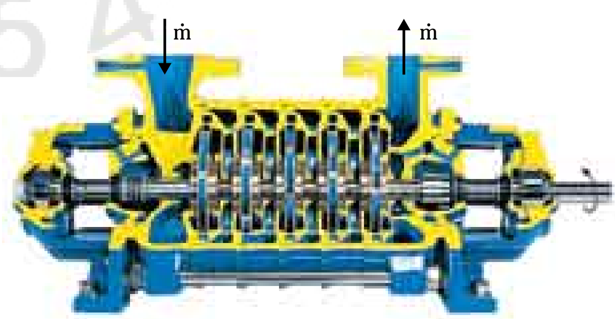


Fig. 8.6
Multi-stage pump in horizontal arrangement with disk shell with diffusers.

[3] An ***orifice plate*** is a device that is positioned perpendicular to the flow into the flow pipe and achieves reduction of passing fluid pressure due to the increase of local resistance.

is installed (local resistance). There also may be a valve which remains closed when the pump flow toward the boiler is high (high load on the pump).

Because of the suction of the feed pump by the deaerating feed tank, where the pressure on the water surface is equal to its vapor pressure (wet steam conditions), the net pressure suction head available of the pump (NPSHa) is equal to the height of the water level inside the deaerating feed tank if deducting losses of the suction pipeline. This difference may not be sufficient, due to the position of the deaerating feed tank or the water level in it, to create the necessary pressure on the pump suction that will prevent cavitation conditions. Thus, at the piping system, between the feed pump driven by the steam turbine and the tank, a lower power centrifugal type booster pump is installed. The purpose of this pump is to increase the pressure of the available net pressure suction head on the suction of the main feed pump.

Also, when the steam consumption of the boiler is low, auxiliary centrifugal pumps of smaller capacity are used for the feedwater provision (fig. 8.7).

Fig. 8.7
Multistage auxiliary feedwater pump.

2) Main condenser pumps.

Main condenser pumps are horizontal or vertical layout centrifugal pumps driven by a motor. They are used to transfer the condensate that comes from the exhausts of the main steam turbine and is concentrated at the main cooler or main condenser (steam returns of the turbine) (fig. 8.8, process 2, 3). In some networks, for steam turbine condensate management, there may be smaller steam turbines used to drive these pumps.

The suction of the main condensate pump takes place from the bottom of the steam condenser, wherein the condensate is concentrated (saturated water). In some networks, before the condensate is led to the feed water deareation tank (DFT), the pump discharge may be through heat exchangers, so as to reduce its temperature (fig. 8.9).

To be able to cope with the axial and radial thrust forces developed by the impellers, these pumps have two pressure stages with ball bearings mounted on the rotation axis of the rotor between the pump and the motor (fig. 8.10).

Moreover, between the impellers on the pump casing a liner-shaped watercooled bearing is placed. This is made of special alloy (non-leaded copper, aluminum copper, brass manganese, etc.). The rotating shaft passes through the liner and so the management of the radial thrusts exerted on the pump is enhanced.

The first stage constitutes the casing of the bottom rotor-impeller, while the second stage is the upper one. The impellers are fitted with the suction opening of the lower impeller upward, and the upper impeller downward respectively. By this installation, hydraulic pressure balance is accomplished during shaft rotation by two impellers since opposite axial thrusts are generated, so these exercised in the bearings of the pump are reduced. To dissipate the air that is collected into the casing, a tube is connected over the casing of the impeller of the first stage to the upper space of the condenser shell. Because of vacuum (underpressure) prevailing inside the condenser, the communication of the two spaces helps to remove the air accumulated inside the pump casing.

The sealing of the shaft at the point where it enters the pump casing is accomplished by gland packings or mechanical seal. To optimize the sealing, but also to remove the heat developed by friction, an amount of condensate from the discharge of the second stage of the pump is injected through a tube in the staffing box of the packings or in the mechanical seal.

The condensate removed of the condenser is usually at a vaporization pressure or near it. Consequently, the net positive suction height (NPSH) entering the pump is substantially equal to the height

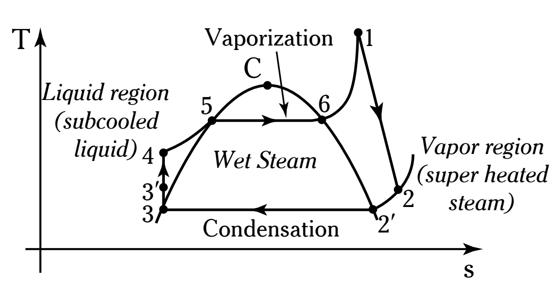


Fig. 8.8
Cycle of vaporization and condensation on a diagram of Temperature (T) and specific entropy (s).

of the water level within the condenser. Therefore, in order to increase the pressure difference that will generate effective NPSH conditions in the impeller of the first stage of the pump, the pump is placed at a point of the engine room as low as possible below the condenser. In achieving the desired NPSH, the piping system losses (albeit small) must be considered. Also, the impeller of the first stage is often designed to operate at low NPSH.

If the pump motion is performed by steam tur-

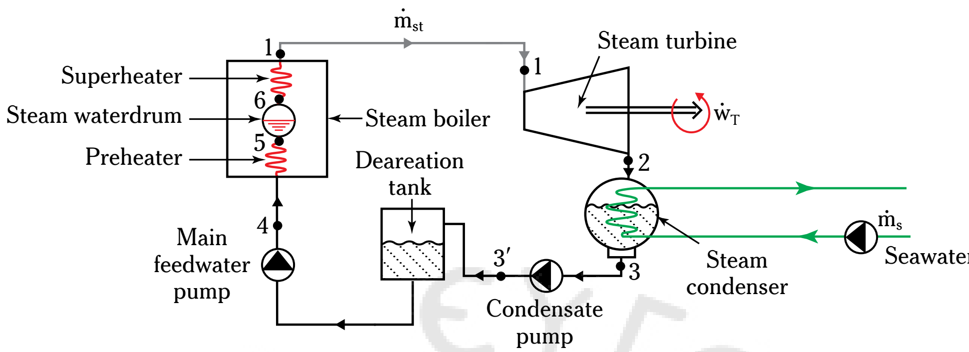


Fig. 8.9

Simplified system of vaporization and condensation cycle.

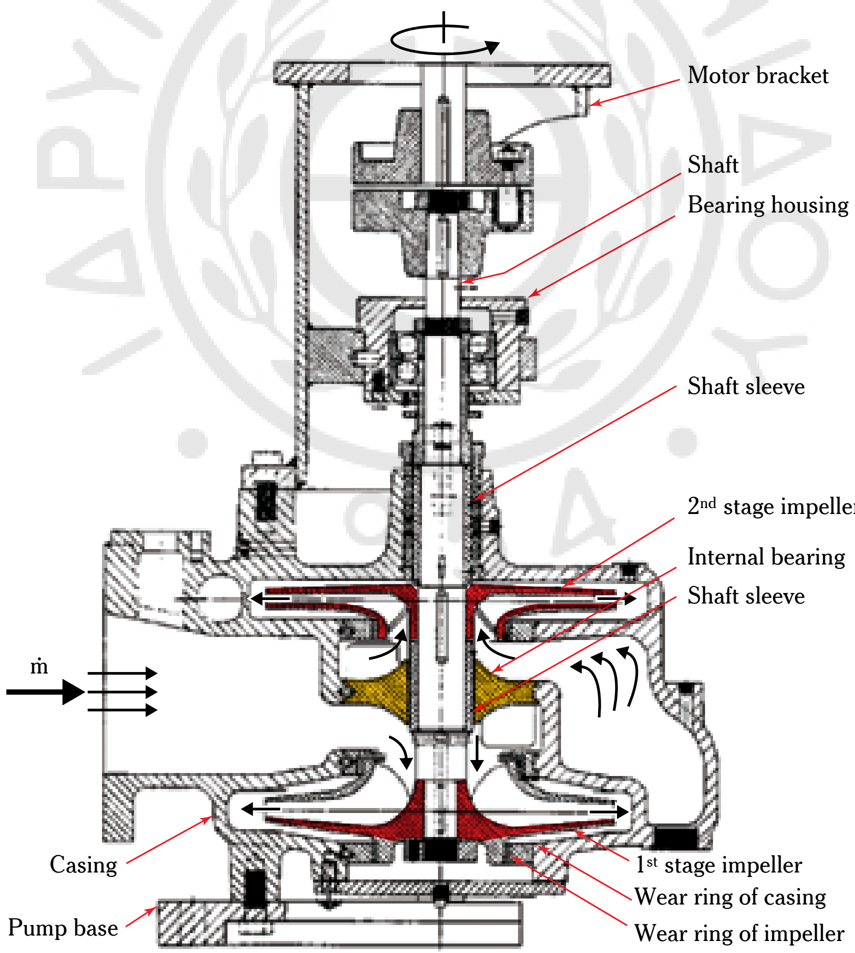


Fig. 8.10

Two pressure stage condensate centrifugal pump.

bine or electric motor which may change its speed, the speed of the pump is adjusted for the removal of the condensate according to the system's requirements. That is, the pump speed is adjusted to realize a balance in the flow rate of wet steam towards the condenser and in the flow rate of condensate removal.

To reach this balance, a control system of the condensate level is used, that acts on the drive machine and increases or decreases the rotations correspondingly. Alternatively, when a constant speed motor is used for the pump movement, the condensate removal flow rate by the pump may be effected by controlling the degree of immersion (namely the height between the surface of the liquid and the impeller) or otherwise by regulating the cavitation conditions.

By controlling the immersion degree, the reduction at system load (because of steam circulation to the steam turbine, therefore less wet steam to the condenser), results in the reduction of the condensate level in the condenser. A corresponding reduction of the available NSPH to the condensate pump will take place, which will fall below the required NSPH. The reduction of available NSPH below the required NSPH will cause a reduction of the pump capacity to remove the condensate, and, simultaneously, the occurrence of the phenomenon of cavitation. As the level in the condenser will continue to fall, the phenomenon of cavitation will increase, while the flow rate of the pump will decrease, until it corresponds to the flow rate of the introduction of wet steam and the creation of condensate. Then the level of condensate within the condenser will be stabilized and the available NSPH will be equal to the required NSPH of the pump, creating a new reduced flow rate from the pump.

The pumps used with an immersion degree control are specially designed to operate at variable NSPH, whereas they are constructed so as to withstand the cavitation and vibration.

To avoid increasing the cavitation phenomenon, the provision of constant speed pumps can be adjusted by limiting the discharge valve so as to maintain the level within the condenser. However, the reduction in supply may result to an insufficient cooling of the ejectors and reduction in their yield or to recirculation of the concentrate within the pump between the suction and discharge. The use of an ejector aims to create the vacuum inside the condenser. To prevent this kind of problem and to maintain the satisfactory level of liquid in the condenser, a recirculation pipe is used for the recirculation of the condensate to the condenser with an automatic or manual valve. Also, a control valve is used for the control of temperature. This control valve keeps the temperature of the concentrate in the desired predetermined value by recirculation.

Generally, constant speed pumps are constructed to operate at their nominal flow rate. Therefore, their operation at low flow should be avoided.

3) Liquid ring vacuum pumps.

Liquid ring pumps create vacuum and are driven by motors (fig. 8.11). They are designed and are used to create vacuum as well as to remove the air of the condenser, and sometimes replace the ejectors (fig. 8.12). However, the vacuum created by the liquid ring pump, is limited by the vapors pressure of the operating water of the pump. This limitation is due to the increase of temperature of the operating water due to the elevated temperature of the liquid-vapor mixture that is sucked from the condenser. For this reason, the water which is separated from the mixture and mixed with the operating water is discharged to a heat exchanger (cooler) to reduce its temperature before returning to the pump, and so to improve the pump performance.

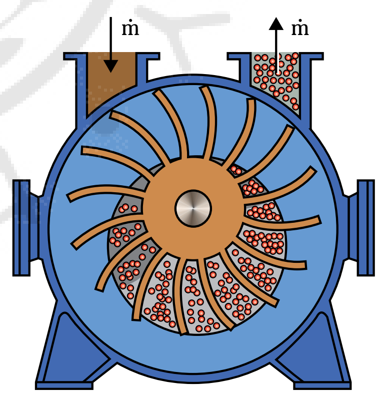


Fig. 8.11
Liquid ring vacuum pump.

4) Condensate pumps.

These pumps are used to pump the condensate that is returning from the general use steam system of the ship and is collected in the vapor return reservoir, except the steam return of the turbine (fig. 8.12). These condensates are not mixed with anoth-

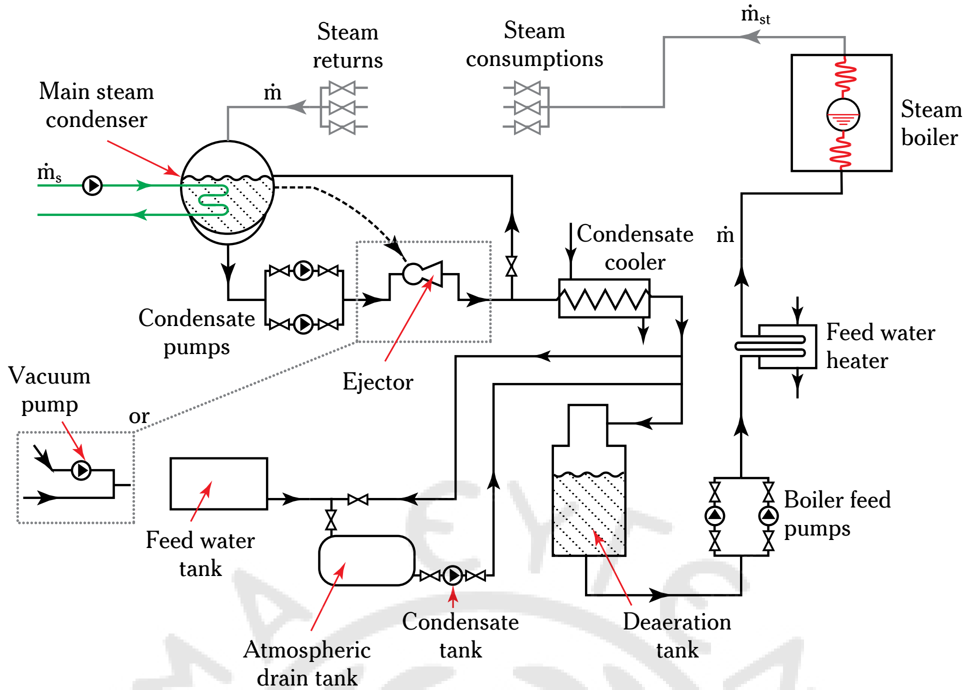


Fig. 8.12
Simplified feedwater and steam circulation system.

er fluid (pure concentrate), and the tank of steam returns is commonly referred to as ***atmospheric drain tank*** because the condensate is collected at a pressure equal or less than atmospheric pressure. So, the pumps are usually called ***atmospheric drain tank*** pumps, accordingly.

Usually they are single stage centrifugal pumps driven by an electric motor. The control of operation of the pump is done automatically using floaters properly installed. So, when the tank level reaches the upper floater, the pump is activated and runs until the level of liquid reaches the lower floater that is installed to stop the operation.

The pumped concentrate is discharged in the DFT, while the concentrate temperature usually ranges from 93 °C to 99 °C. Because the temperature is near the water boiling point, the pumps are placed as lower as possible from the tank, in an effort to maintain the available net positive suction head of the pump at a high level.

5) Main seawater circulation pumps.

The main seawater pumps in a ship with propulsion by steam turbine are designed to supply the pipe system that serves the main condenser (or steam condenser – condensation heat exchanger) of steam returns from the steam turbine. Also, they are used to provide seawater to networks where other heat exchangers are installed, such as main lube oil coolers etc. (fig. 8.13).

The pump suction is performed through the network connected with the valve that is installed on

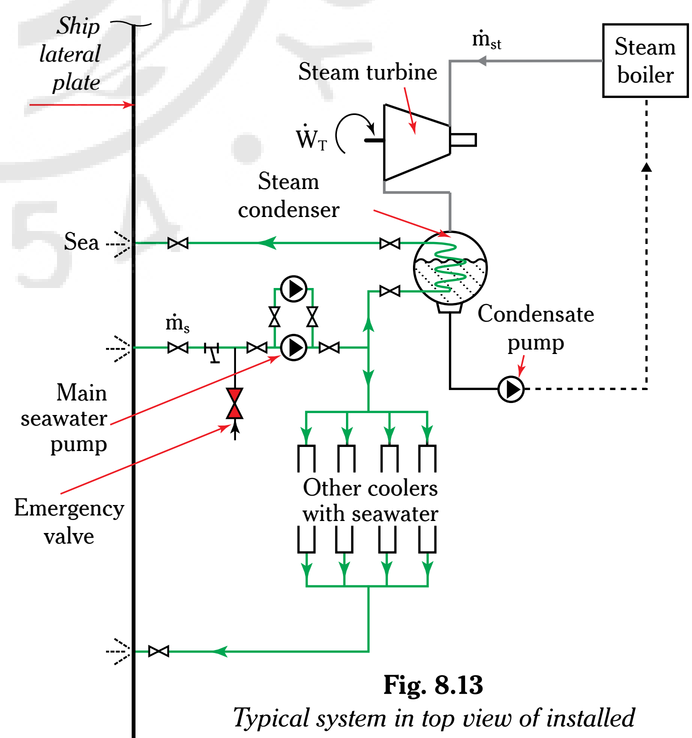


Fig. 8.13
Typical system in top view of installed main seawater circulation pump.

the ship's side, while the discharge of seawater after the condenser or any other heat exchanger returns back to sea. ***For the ship's safety, a valve is installed to the suction system which enables the pump to suck the liquids from the bilges of the engine room in an emergency. This valve is called emergency valve (EM'CY) and is connected to the main seawater pump due to its high capacity.***

In ships with a steam turbine, due to the high flow required by the steam return cooler (condenser), the pumps employed have relatively low discharge pressure and are single stage axial flow pumps with propeller (fig. 8.14). When higher discharge pressure is required, mixed flow pumps are installed. The drive machines of the pumps are steam turbines with reduction gears or motors of two "speed" rotation. By these, the ability provided is to control the pump speed, and hence the flow rate, especially when starting or stopping the steam turbine where, due to low steam consumption, the requirements in seawater are reduced.

Because of the high electric power load which is required for the operation of the main seawater pump, a pump with lower supply is installed in the seawater piping system. This pump is usually a motor driven pump and is called ***port use pump***. The pump is used at port or at anchorage when the requirements of seawater for cooling are small because the steam turbine does not operate, and is used to provide sufficient supply of seawater to the other coolers.

In many cases the main seawater pumps do not have thrust bearings to absorb the axial thrust. Hence, the pump shaft is fixedly connected to the shaft of the drive machine so that the axial thrust is taken by the thrust bearing that is installed on the shaft of the motor. For the axial thrusts during operation, the bearing is installed on a transverse brace above the impeller of the pump. The necessary lubrication of friction points (affected by the presence of mud and sand drifting from seawater when the ship sails close to shore or in rivers) is achieved by providing clean water from an auxiliary network. In several pumps, the bearing surfaces are coated by resistant synthetic material.

6) Boiler fuel oil pumps.

In steam turbine propulsion vessels, these pumps are used to supply the fuel for the operation of the steam boiler. Depending on the type and design of the fuel piping system and the boiler mode of operation, there are usually pumps for Heavy Fuel Oil and pumps for Diesel Oil.

1) The ***diesel oil pumps*** are motor driven rotary pumps and are used, during initial starting of the boiler, because ignition of Heavy Fuel Oil is not easy. Then, with

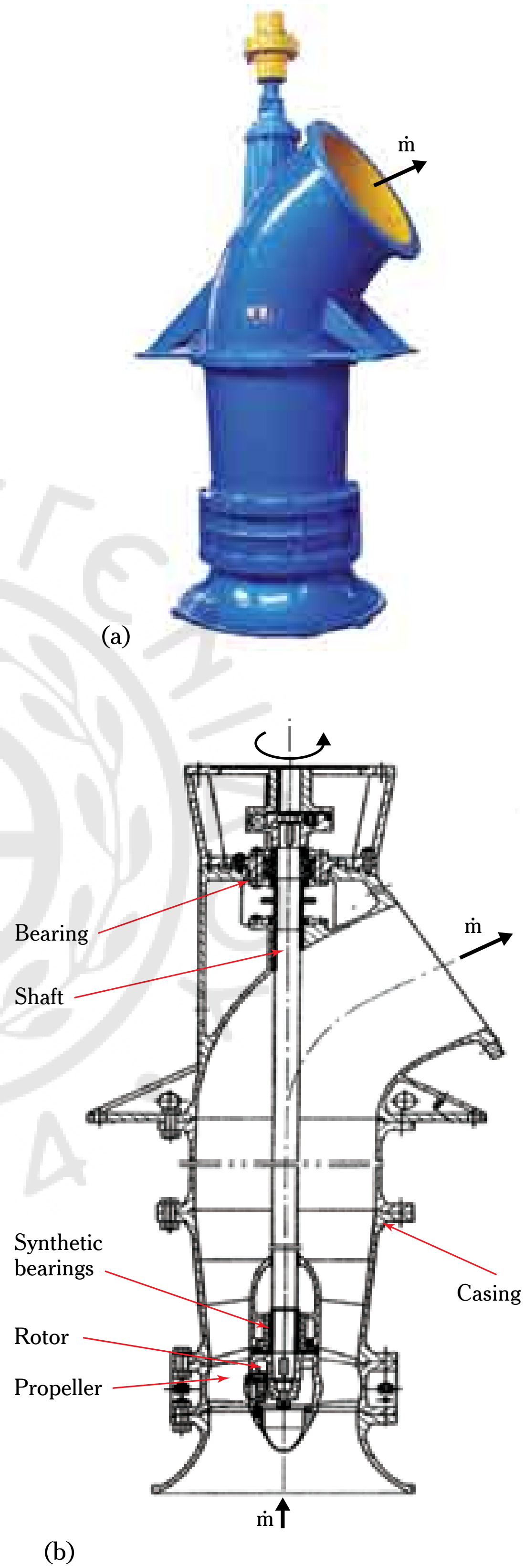


Fig. 8.14
(a) axial flow pump and
(b) cross section drawing and its parts.

the provision of heavy fuel oil, the combustion is continued while the operation of the diesel pump is interrupted.

2) The ***heavy fuel oil pumps*** are screw type pumps with multiple rotor and are placed in horizontal or vertical arrangement. Their motion may be accomplished either by compact steam turbines, or by two-speed motors. The potential use of reciprocating pumps for fuel oil provision is only existent in old ships.

The pumping of the oil is carried out from the F.O. daily service tank or from the F.O. settling tank. Also, for efficient pumping, injection by the burner and combustion, the temperature of fuel is increased with preheating and this is realized at a steam heat exchanger.

In steam turbine driven pumps, whether they are rotary or direct acting reciprocating, the steam provided for operation is usually controlled by a constant pressure regulating arrangement. This acts to maintain the fuel discharge pressure by the pump constant. By these arrangements, without being affected by any fluctuations in the steam network, constant flow is achieved. When a motor is used for the motion of the pump, it is usually with two rotation speeds. This happens in order to operate the pump at low speed, providing less fuel to the boiler, when the consumption requirements are low (in port) and at high speed when the boiler is operating at full load (at sea) respectively.

However, at each operating speed, the supply of fuel from the fuel pump is greater than that consumed. That is the reason why fuel recirculation is needed, and it is achieved through an appropriate network where a bypass valve is installed.

For the protection of the network during pump operation if the pressure increases beyond the desired level, a relief valve must be installed from discharge toward the suction in the pump discharge. The control of the pump is carried out automatically by the control panel, while the possibility of remote control and shutdown of pump operation in an emergency is supplied.

The entry point of the rotational axis on the casing of the pump is sealed by mechanical seal. Above this point, fuel oil diffusion in case of leakage is prevented by a safety cover. Also, in case of small leakages below the pump, a collecting tray exists, that leads these leaks to a dirty oil drain tank.

It is important for any leakage of the fuel pumps to be treated directly, due to the flammability of the fuel, by replacing the sealant.

7) Steam turbine lubricating oil pumps.

These pumps are used for the lubrication of the turbine. The suction of lubricant oil is realized by the ***reduction gear sump tank*** and discharged in the turbine bearings, in the gearbox with the gears of the gear unit, and in the thrust bearing of the shaft of the propulsion propeller of the ship (fig. 8.15). Usually, this lubricant is also discharged in the speed regulator as well as in the flow control valve of the steam nozzle in the turbine. A portion of the pumped lubricant is also discharged in a gravity tank, and in case of a sudden interruption of the lubricant supply, the pump maintains the lubrication to the propulsion mechanism for several minutes. Most vessels with propulsion by steam turbine have two and three horizontal or vertical arrangement gear type pumps. In vertical arrangement, the pumps may be submerged in the oil sump tank.

The pump drive mechanism may be a steam turbine, a motor or the propulsion shaft through an appropriate arrangement (e.g. transmission by gear at the attached pumps). If the pumps are driven by steam turbines or by propulsion shaft, a pump should be installed in the network driven by an electric motor, in order to ensure the oil supply to the steam turbine at starting. Also, in some ships additional piston pumps moved by small diesel engines may be used, or electric motors that are connected to electric accumulators (batteries) and ensure the provision of lubricant in case of emergency.

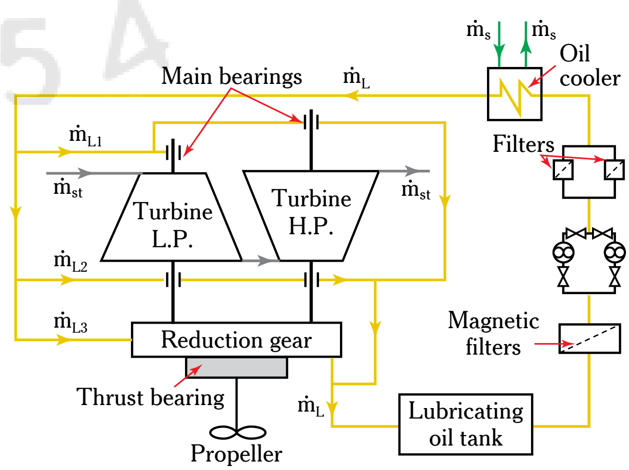


Fig. 8.15
A typical piping arrangement of steam turbine lubrication system.

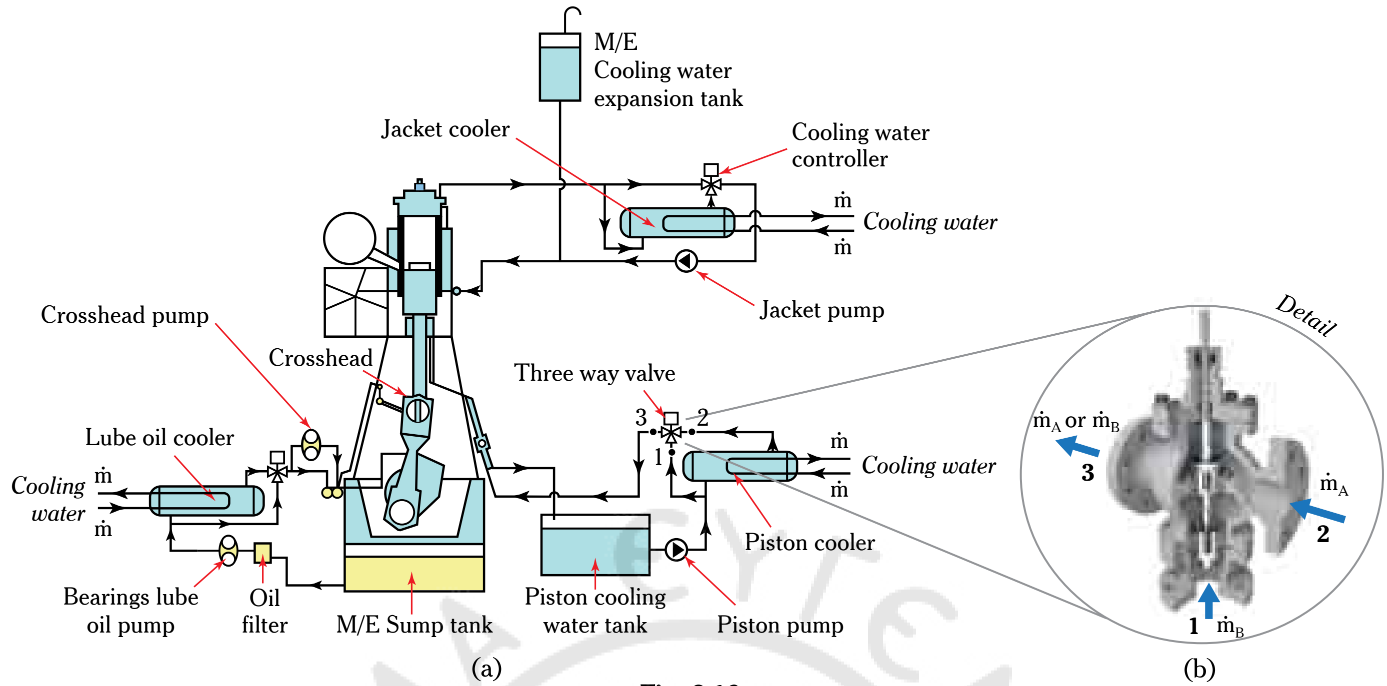


Fig. 8.16
(a) Typical piping arrangement of cooling water and lubrication system of M/E. (b) Three-way valve section.

The pumps for the supply of lubricating oil are positioned low, near the lube oil pump tank. In the piping system the lubricant provision pressure is adjusted, in accordance with the requirements of mounting bearings and gears, by placing orifices. This is required because, usually, the pump discharge pressure is in accordance with the existing pressure on the propulsion turbine governor.

8.2.2 Ships with Internal Combustion Engine (ICE).

Internal combustion engines operate at different speeds so that they could be classified as ***Slow***, ***Medium*** and ***High speed*** diesel engines. The pumps used on ships to serve the proper functioning of the ICE are: the cooling water pumps, the seawater pumps in cooling systems (par. 8.4.2), and the booster fuel pumps and lube oil pumps.

1) Jacket fresh water cooling pumps.

These pumps are used to circulate the water (usually distillate) in the cooling jackets of the main engine cylinder covers, in the turbocharger cooling system, and/or through the elements of the heat exchanger of the fresh water generator (fig. 8.16).

Cooling water that is handled by the jacket cooling pump of the engine is cooled by a heat exchanger (cooler) in the corresponding system. When the engine is stopped, the same water, through a heat exchanger (preheater), is preheated to maintain the engine temperature a little lower than the operating temperature, so that it is continuously in standby mode. The engine cooling pumps are single stage centrifugal pumps in vertical or horizontal arrangement (fig. 8.17), each of which is able, when the engine operates at full load, to meet the needs of the cooling system that it serves. The impeller is usually of single suction. The sealing is achieved either by packings and gland follower or by mechanical seal. Their movement is performed by electric motors,

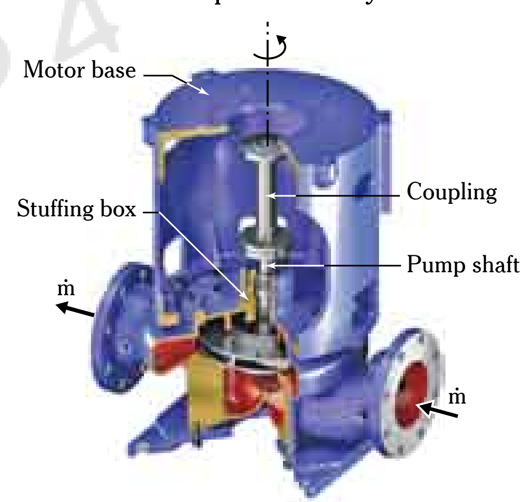


Fig. 8.17
Jacket cooling pump.

while, when used as cooling medium of high speed diesel engines, the motion transmission to one of the cooling pumps may be supplied by the main engine (dependent or attached pump).

2) Main engine piston cooling pumps.

These pumps are used to cool the pistons in certain types of engines, providing cooling water through pipes that send water to the interior of the piston heads (fig. 8.16). They are motor driven pumps, usually in vertical arrangement. The pump is submerged in the tank containing the water for cooling the pistons (fig. 8.18). The network where they are installed is separated from the jackets cooling system, preventing mixing and contamination of the water by leakages that may occur either from the side of the jackets or the pistons.

3) Heavy fuel oil supply and booster pumps.

These pumps which are installed in ships must have the ability to handle fuel oil with different characteristics, such as the high temperature of fuel oil used in ICE (fig. 8.19). Furthermore, various types

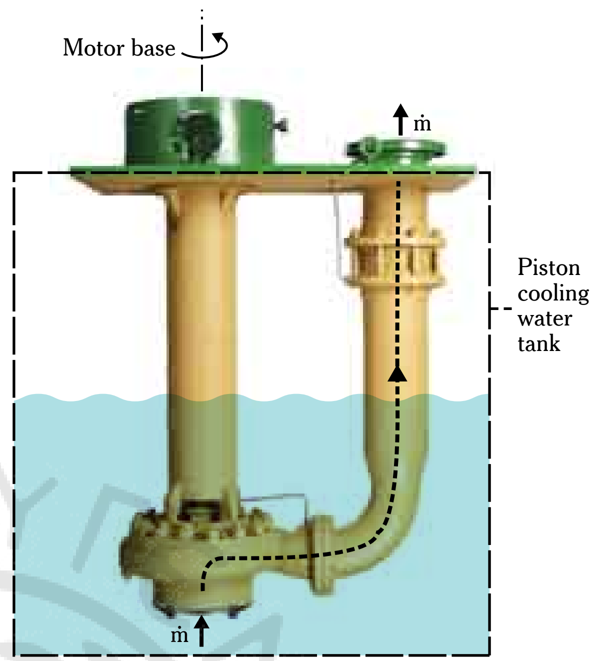


Fig. 8.18
M/E piston cooling pump installed within the cooling water tank.

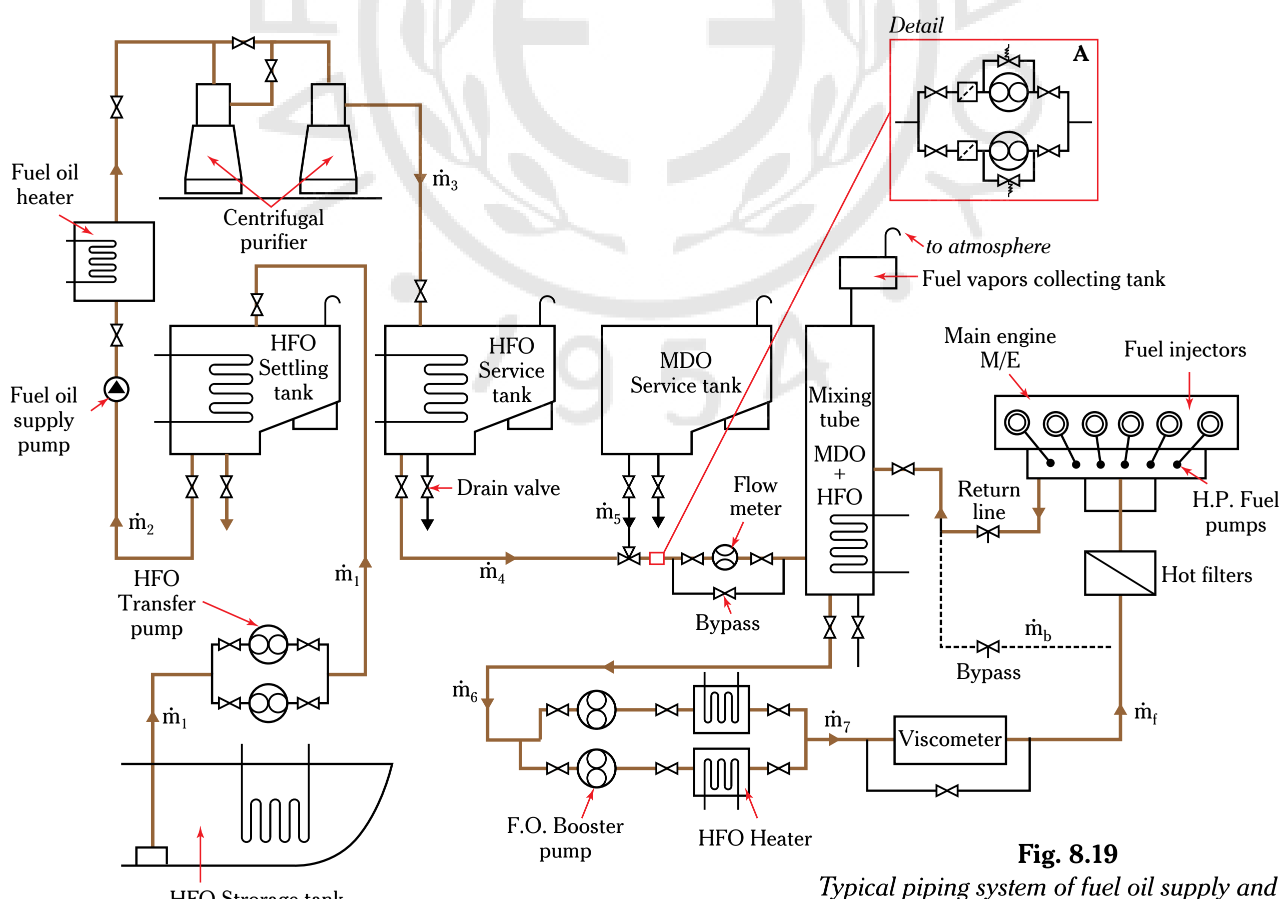


Fig. 8.19
Typical piping system of fuel oil supply and purification for M/E. (A = fuel oil supply pumps).

of fuel oil are used, and this is another important difference that should be added. The types of fuel oil range from heavy fuel oil to marine diesel or marine gas oil, used in certain engines during ship maneuvering, when the main engine operates at low speeds, or for the operation of the main engine in specific sea areas (Sulphur Emission Control Areas – SECAs).

They are pumps with two or three screw impellers or gear type pumps, motor driven (fig. 8.20).

Usually, they are of horizontal arrangement. In high speed and medium speed diesel engines, movement may be effected through gears that are moving by the engine itself (attached pumps). The supply of each pump must correspond to the consumption of the engine at full load.

The suction of the fuel supply pump to the fuel system is performed primarily by the daily service tank, but, for safety reasons, the possibility of suction from the settling tank is given, to ensure the fuel availability to the machine only in case of emergency.

In certain high temperature networks in the fuel system, in order to increase the pressure of the fuel, after the supply pump, there may be a booster pump connected in series. These pumps are installed either between the daily consumption tank and the engine after the supply pump (fig. 8.21), or after the mixing tank. Due to the excess in the supply of oil discharged by these pumps, even if the engine is operating at full load, a fuel quantity is re-

(a)

(b)

Fig. 8.20
Heavy fuel oil supply and booster pumps. (a) Installation of two pumps in parallel. (b) Pump section.

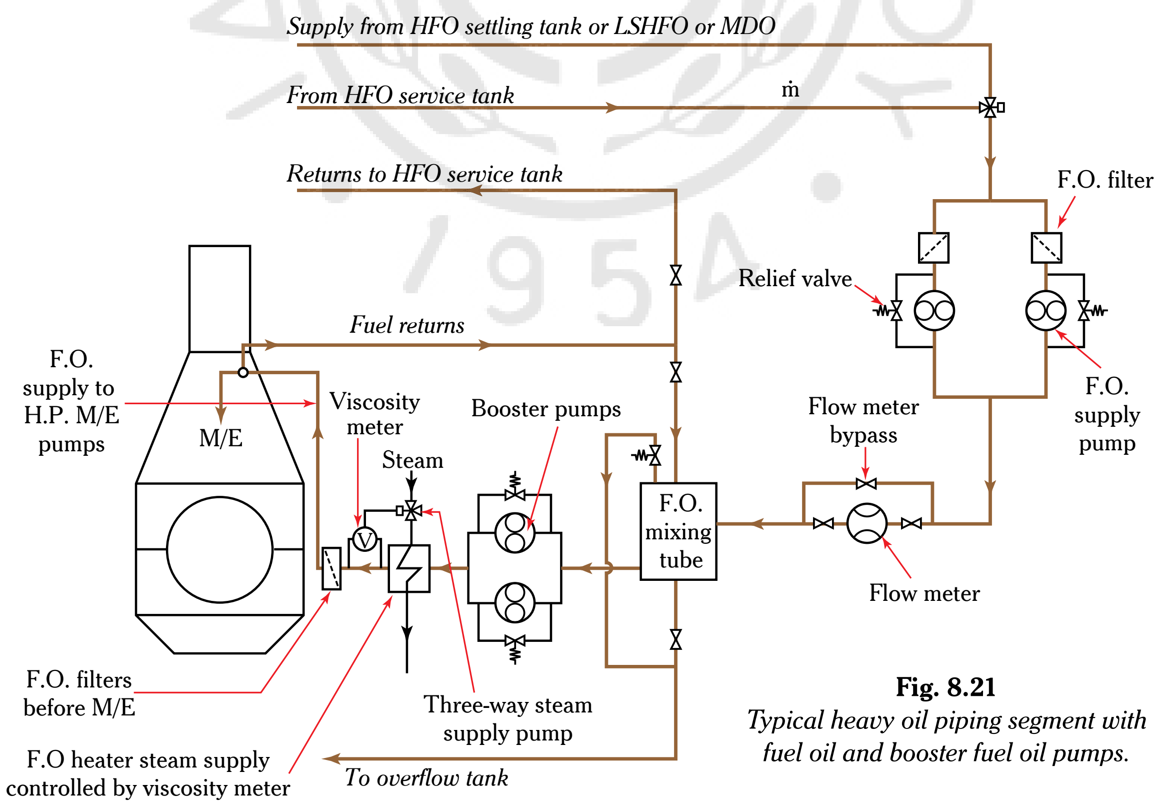


Fig. 8.21
Typical heavy oil piping segment with fuel oil and booster fuel oil pumps.

turned to the network [through special arrangement - bypass]. The sealing of these pumps is achieved by mechanical seals. At the discharge of each pump, a relief valve is installed in case of pressure increase in the pipes over the desired one. Through the valve, the fuel pressure is released from the discharge in the suction side [fig. 8.19(A)].

4) Lubricating oil pumps.

These pumps are of vertical or horizontal arrangement powered by electric motors (fig. 8.22).

Fig. 8.22
Main engine lubricating oil pump with screw rotors.

Alternatively, the pumps may receive the driving power from the engine, but this happens in medium and high speed diesel engines (figs 8.16 and 8.23). Two pumps are installed on each system, but each pump has sufficient supply for lubrication of the engine at full load. The pumps used are either centrifugal or with two or three screw rotors or geared rotors. The suction of oil is effected by the sump tank, and this is the reason they are placed low in the engine room. Usually they are located on the ***last level of the engine room*** (commonly, last plate[4]) (fig. 8.24), while they may be submerged in the sump tank.

The discharge of the lube oil by the pipes takes place on the mounting bearings of the crank shaft of the main engine, on the cross heads, on the governor control, and on the pistons (through a suitable piping system and oil passage in the piston rod when the cooling of the piston head is carried out by oil).

Otherwise, for cooling the cross heads lubrication, booster (or crosshead) pumps of smaller capacity are used in series with the main lube oil pump, discharging lubricating oil exclusively to the crossheads.

The lubrication pump is also used to supply oil to the thrust bearing of the main engine and the

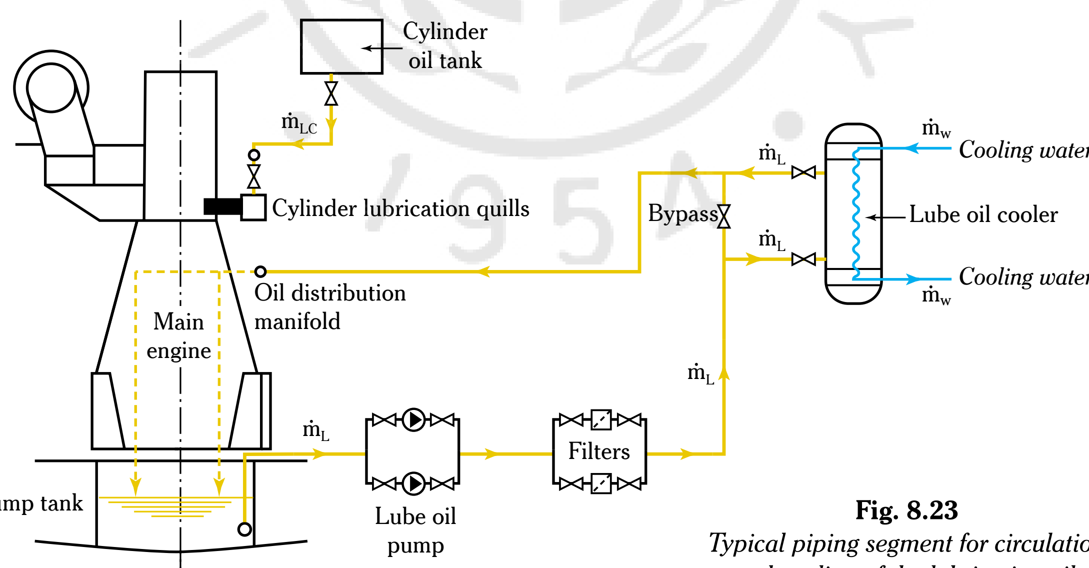


Fig. 8.23
Typical piping segment for circulation and cooling of the lubricating oil.

[4] The engine room ***last plate*** is the lower level of the small metal sheets fitted to the engine room that creates a passageway for the passage of the crew.

gears of the reduction gear at medium and high speed engines. In the last two types of engines an independent rotary pump may be used for lubricating the gear, that is not related to the network of the main lubrication pump.

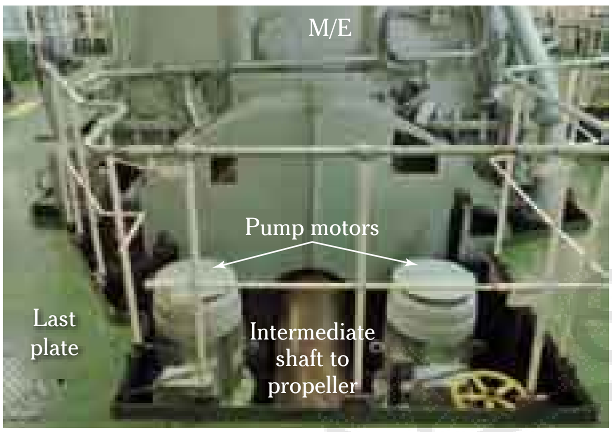


Fig. 8.24
Installed lubricating pumps (Lube oil), motor driven, next to the shaft of M/E.

8.2.3 Gas turbine propelled vessels.

For ships powered by gas turbines, the pumps used are the following:

1) Fuel oil service pumps.

These pumps are used to discharge the fuel at the nozzles of the combustion chamber of the gas turbine. They are gear-type pumps (fig. 8.25), which are driven by the gas turbine (attached on the shaft of the gas turbine through appropriate instruments) (fig. 8.26). Moreover, a pair of rotary pumps operated by electric motors is installed in series with the fuel pumps, to achieve the initial fuel injection for the operation of the gas turbine at starting. Also, they can be used as alternative fuel supply or as booster pumps of the fuel pumps that are attached on the gas turbine.

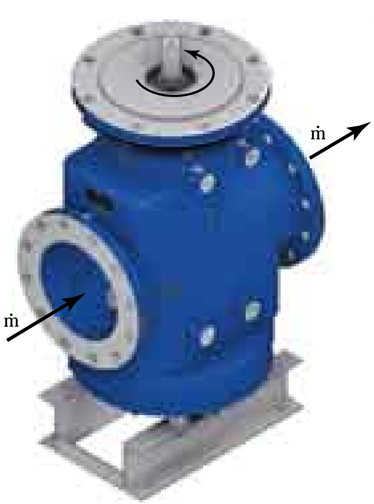


Fig. 8.25
Gear type vertical arrangement fuel pump.

2) Gas turbine lubricating oil pumps.

These pumps are gear type or centrifugal pumps driven by means of coupling on the shaft of the gas turbine (fig. 8.27). The suction of these pumps is made by a synthetic oil[5] tank exclusively used for the lubrication of the gas turbine. Their discharge takes place at the main bearings and the control system, which is used to ensure that, throughout the duration of gas turbine operation, it will provide sufficient amount of lubricant.

To provide lubricant to the turbine during starting and stopping period, where, apart from lubrication, oil is used as a means of reducing the temperature of the bearings, two electrically driven pumps are installed at the lubricating system. These pumps

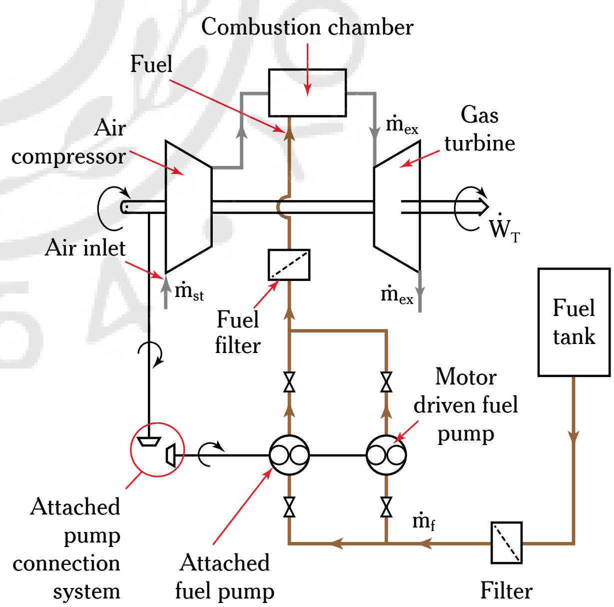


Fig. 8.26
Typical turbine piping system with attached and electric-motor driven fuel pump.

[5] The lubricants are the result of mixing base oils and chemical additives in specific proportions to ensure the requirements of the specifications of the machines they lubricate. Synthetic oils are a result of chemical composition in order to optimize the physicochemical characteristics of the final product.

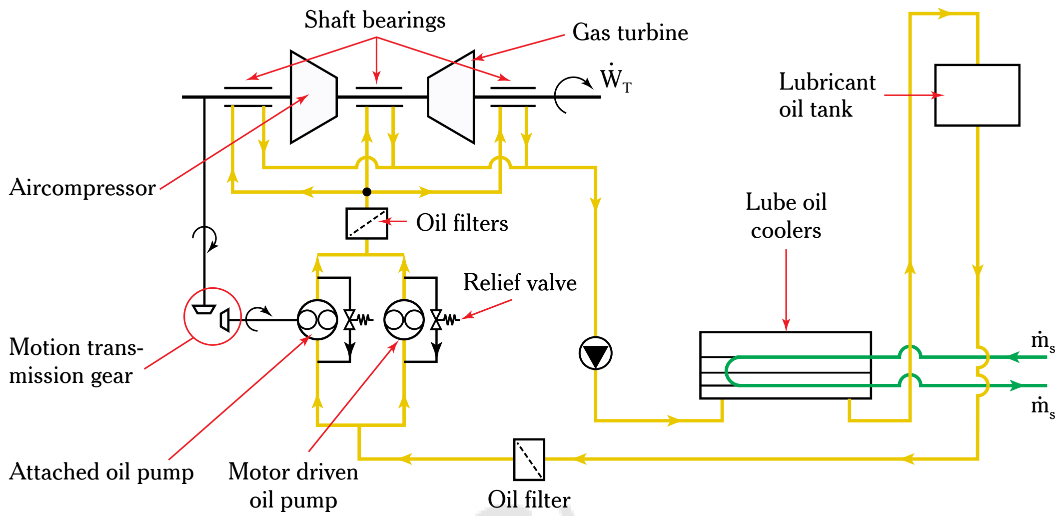


Fig. 8.27

Typical piping of lubrication system for gas turbine with attached and independent lubricating pump.

are gear type, with vanes or centrifugal. Furthermore, they provide the ability, except in periods of starting and stopping the turbine, to replace the attached lubrication pumps if for some reason (damage or repair) they are inoperative.

The safety devices that are connected to the lubrication pumps are:

1) A relief valve, connected to the network to return the oil to the collecting tank if the pressure exceeds the desired one, and

2) a control device that automatically stops the fuel pump when the lubricating oil supply pressure in the gas turbine is reduced. The pressure drop of the lubricant can be caused by damage to the pump or impurities in the oil filters that will reduce the pressure below the limit set in the lubrication control automation.

Following lubrication of the gas turbine, the lubricant is collected in a tank at the bottom and is removed by a pump exclusively used for this purpose. The pump can be centrifugal or rotary driven by an electric motor. Discharge of the pump is performed to the suction tank of lubrication pumps. For the lubrication of the reduction gears that transmit the motion by the gas turbine and their main bearings, independent multiple screw pumps are used driven by their own electric motor.

8.3 Pumps in ship electric power production systems.

The production of electric power on ships can be accomplished either by the use of a steam turbine which will give power to the generator or with an internal combustion engine (ICE). The number of the installed generators, either with steam turbine or ICE, depends mainly on the type of the ship, the propulsion system, the ship's demands for electric consumption and the available source of power which is provided to the power generator. For this reason, the electric power production systems can be distinguished into two categories; the generators that are moved by steam turbine and the generators that are moved by ICE. Depending on the motor of the generator, corresponding pumps are used. These are described below.

8.3.1 Turbo-generator pumps.

In the systems of the turbo-generators, the following pumps are being used:

1) Auxiliary condensate pumps (fig. 8.28).

As the wet steam exits from the generator steam turbine, it is condensed in a small condenser which is usually installed with both the steam turbine and the power generator, in a common system. The removal of condensate from the condenser occurs by auxiliary condensate pumps. These pumps are centrifugal, vertically arranged, with two-stage, and are also moved by electromotor.

Their operation and their characteristics, unlike auxiliary condensate pumps which have smaller size and supply, are similar to those of the condensate removal pumps from the main condenser in ships with propulsion by steam turbine.

The discharge of these pumps takes place in the DFT. In ships equipped with main condensate pumps for the main condenser, the possibility is provided for these pumps to remove, through a suitable bypass, the condensate that is collected by the steam turbine of the generator. This is an alternative in case of malfunction of the auxiliary condensate pumps. Hence, every generator with steam turbine is equipped with an auxiliary condensate pump.

2) Auxiliary seawater circulation pumps.

These are single-stage centrifugal pumps used in order to circulate the cooling seawater to the auxiliary condenser. This pump may be used not only for the supply of seawater to the lube oil cooler of the generator with a steam turbine but also in the combustion air[6] coolers that are installed in the same system.

The supply of the auxiliary pump, in case of failure in the seawater supply pumps, can alternatively be used for seawater supply to the main condenser.

3) Steam turbine lube oil pumps (fig. 8.28).

These are gear pumps with internal or external cut gear or with screws, which are moved through a specific configuration by the steam turbine shaft. (attached pump).

The suction of the pump takes place from the sump tank for the lubrication of steam turbine bearings and it is installed below the steam turbine. The lube oil, apart from the main bearings, is also discharged in the reduction gears and the steam turbine governor. So, the governor, in case of failure or deficient lubrication by the pump, will shut down the operation of the steam turbine.

Furthermore, a manual piston pump may be installed on the steam turbine lubrication system, in order to achieve lubrication of the steam turbine before or after its operation, so that the rotating parts are adequately lubricated until final standstill.

8.3.2 Pumps for power generators driven by ICE.

In these systems, the following pumps are being used:

1) Diesel generator (D/G) jacket cooling pumps.

These pumps are used in the circulation of cooling water for ICE that drives the generator (D/G). They usually are centrifugal single-stage pumps with single suction impeller that are attached to the engine (figs 8.29 and 8.30).

The water supply by these pumps cools the cylinder jackets of the engine, the cylinder heads, the turbochargers and the lube oil cooler, on condition that it is not cooled by sea. The water that cools the air cooler of the engine is being cooled by an auxiliary seawater circulation pump or through bypass from the pump of the main seawater cooling system. The decrease of the temperature of the engine cooling water, if a central cooling system (where seawater cools the freshwater) is used, is achieved by this network.

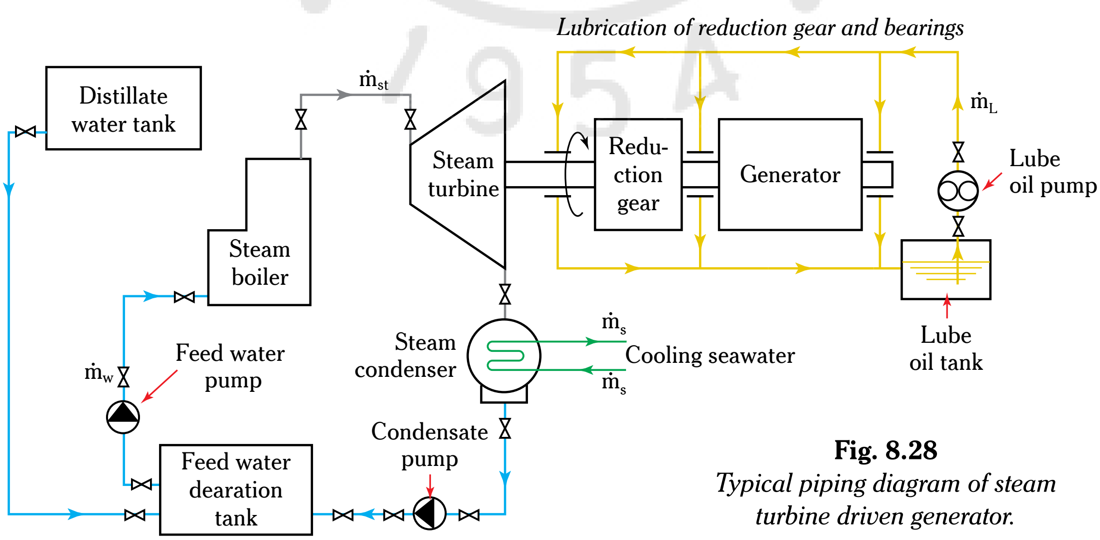


Fig. 8.28
Typical piping diagram of steam turbine driven generator.

[6] ***Combustion air*** is the air used for the combustion of fuel in heat engines.

Fig. 8.29
Attached cooling water circulation pump of cylinders on a diesel generator.

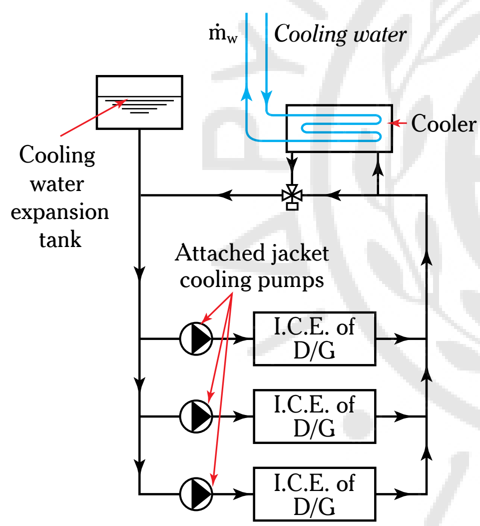


Fig. 8.30
Typical cooling water piping system in ICE generator sets (D/G).

2) D/G Fuel oil pumps.

This pump is used to supply fuel oil to the ICE of the diesel generator and may be attached to the engine or independently moved by an electric motor. The gear type pump is usually used and its motion is performed by gears. Gear motion is realized either by their internal engagement where one gear transmits the motion to the other or by engagement with external gears. The sealing of the impeller shaft is accomplished by an oil sea ring, while the lubrication of external gears is accomplished by lube oil that the user fills in the chamber where the motion transmitting gears are. The fuel oil supply pumps in diesel generators, depending on their structural characteristics, handle both types of fuel oil; heavy fuel as well as diesel fuel oil. Pump suction is performed at the daily service fuel tank (of appropriate fuel) and the discharge on the engine fuel injectors. When heavy fuel is used, the necessity of its recirculation is accommodated through an appropriate piping system. By its recirculation, the fuel can maintain its high temperature making its flow and its injection easier.

Also, if the discharge pressure exceeds the desired level where the pressure relief valve is adjusted, the recirculation system is used.

3) D/G Lube oil pumps.

The lube oil pump is attached to the internal combustion engine of D/G and the rotors consist of gears or lobes (fig. 8.31). The oil suction is realized at the

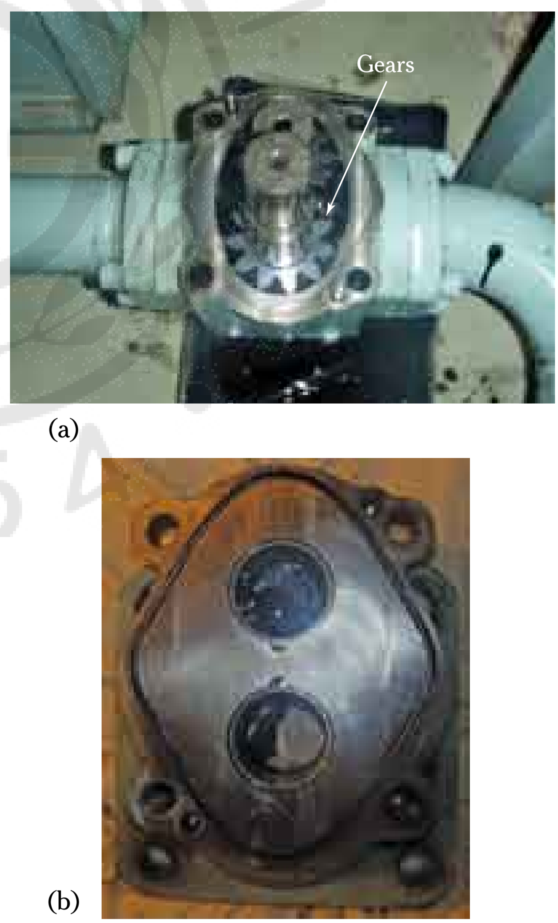


Fig. 8.31
(a) Attached lubricating oil gear pump lubrication of diesel generator. (b) The pump cover.

engine sump tank while its discharge is performed on the piston rod bearings and on the mounting bearings of the crank shaft. Then the lubricant flows out and is recollected in the sump tank. The lube oil that is transported by the pump, passes through the cooler and is cooled by seawater or distilled water when the central cooling system is used, in order to decrease its temperature (fig. 8.32).

In addition, the pressure and temperature of the lube oil are controlled by an automatic device. This device consists of specific sensors which are able to recognize the temperature values as well as the values of pressure. If these values are lower or higher than those that have been set, the engine will shut down.

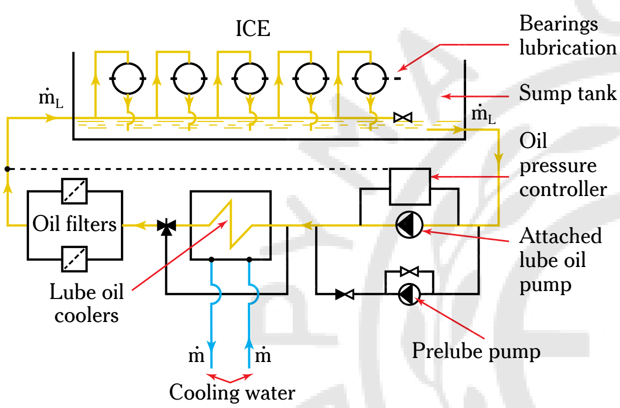


Fig. 8.32
Lubrication in ICE of diesel generator.

8.4 Pumps of auxiliary systems (auxiliary pumps).

The auxiliary system pumps serve the flow of different liquids from the auxiliary systems and for this reason they are typically found in all ships. Their number or size can differ and depends on the quantity of liquid that they handle as well as the size of the system they are installed in. Most of these pumps, as it has been mentioned, are two in every system, making the ship safer. The various types of auxiliary pumps are described below.

8.4.1 Fuel oil transfer pumps.

The fuel oil transfer pump is used for transferring the fuel oil from storage tanks to settling tanks or, if needed, from one storage tank to another (fig. 8.19). In general, the possibility is provided either for oil to be transferred from one tank to another (e.g. to concentrate remaining fuels in one storage tank in order to avoid mixing the existing fuel with new fuel deliveries) or for oil to be transferred to facilities ashore or to some other ship through the manifolds on deck (e.g., operations of prepare for the ship's dry docking).

The pumps used are arranged horizontally (fig. 8.33) or vertically (fig. 8.34), and are rotary with screw rotors or single-stage centrifugal pumps with electric motor. In some cases, when there is available steam, the steam piston type pumps can be used.

These pumps, whichever their type might be, are installed in the lower part of the ship, in order to obtain better suction conditions. In pumps with electric drive motor, the motor can be of constant speed or two speeds. In case of pumps with constant speed motor, the adjustment of flow rate is achieved:

Fig. 8.33
Fuel transfer pump in horizontal arrangement.

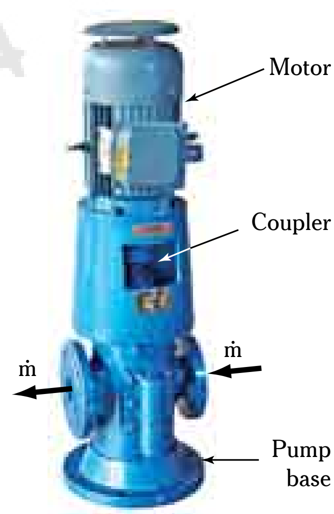


Fig. 8.34
Fuel oil transfer pump with screw rotors.

1) by limitation of the supply to the discharge valve,

2) by connecting more tanks in the discharge system of the pump. In this way the fuel supply is divided and controlled through valves, and

3) through the bypass system by an appropriate valve; this allows a part of the total supply of liquid to pass from the discharge to the pump suction, so the fuel is recirculated.

In ICE ships, for the centrifugal purification of fuels, centrifugal purifiers are being connected in the transfer system. The supply to the purifiers is achieved either by an attached gear pump or by a pump which is driven by electric motor. For discharging the pure fuel from the centrifugal purifiers in daily consumption (or service) fuel tanks, if the discharge pressure of the centrifugal purifier is not enough, a booster gear pump may be used in series with the discharge of the centrifugal purifier. So, the fuel transfer in the daily service fuel tank is facilitated.

8.4.2 Seawater service pumps.

Seawater service pumps or cooling seawater circulating pumps are used for the supply of cooling seawater to the heat exchangers which are installed in the engine room (fig. 8.35). These are the jacket coolers of M/E, the lube oil coolers of M/E, the steam condensers (when there is a steam system) and oth-

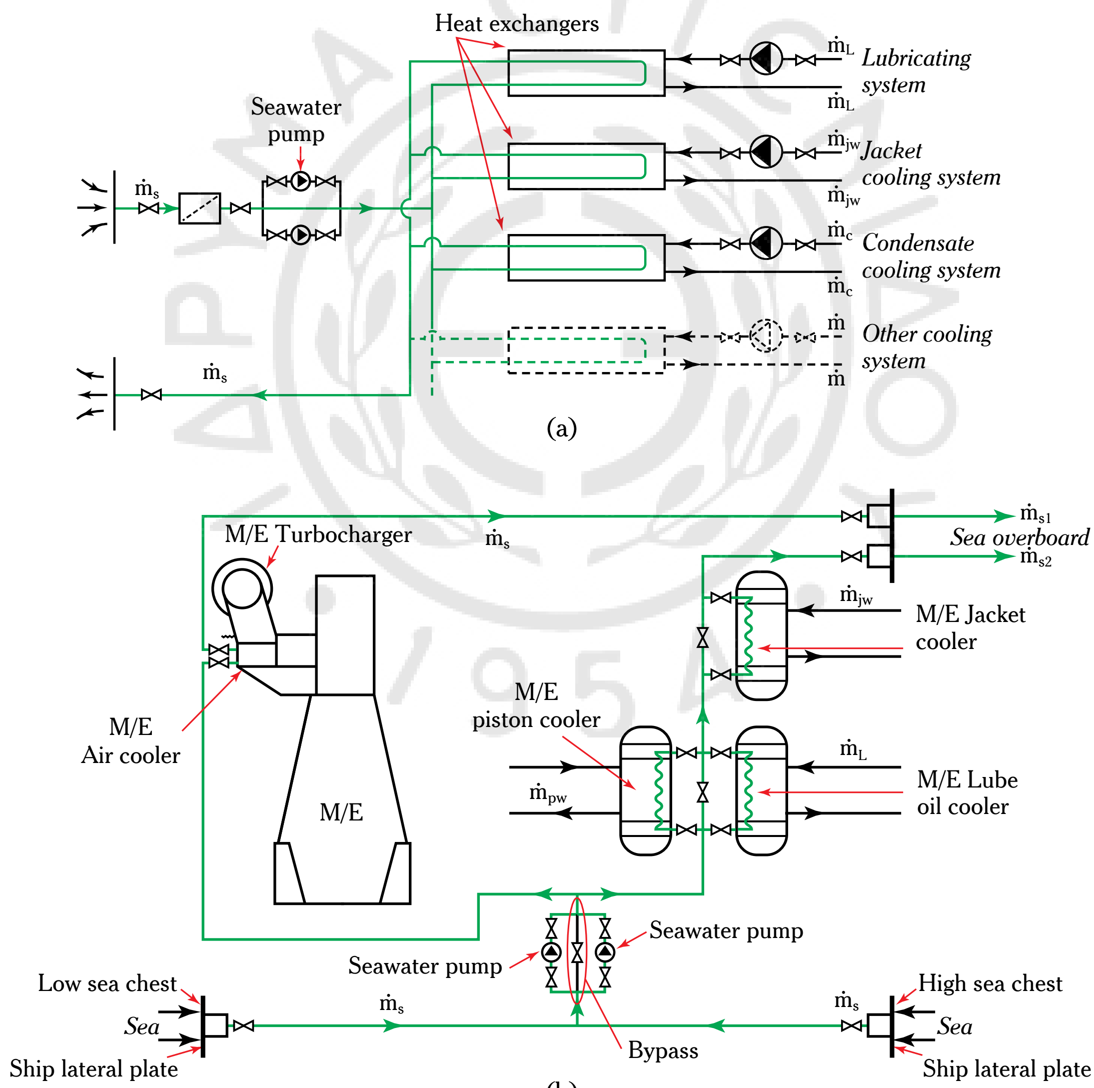


Fig. 8.35

(a) Typical piping arrangement of seawater cooling system.

(b) Typical arrangement of seawater supply pumps in ship's system.

er smaller exchangers which include the condenser of the refrigerant of the air condition plant and the compressors' air cooler.

In the systems of new-built ships, the seawater pumps may provide seawater only to a central heat exchanger, where the water is being cooled for the cooling system of the main engine generator engines and compressors (fig. 8.36). Part of the sea, which is supplied from the pump, may be used in the other part of the ship's cooling system which includes the air condition coolers, steam condensers, etc.

However in both systems, the suction of pump water is performed from the sea suctions, the high chest and low chest, which are installed on the sides of the ship. In M/T, in combination with the central heat exchangers supply, for the cooling of turbine steam returns of the pumps in the main condenser, there is a division in the seawater cooling system which sends seawater directly to it. The seawater, after its passage through various coolers, returns to the sea.

In these systems the type of pumps used are cen-

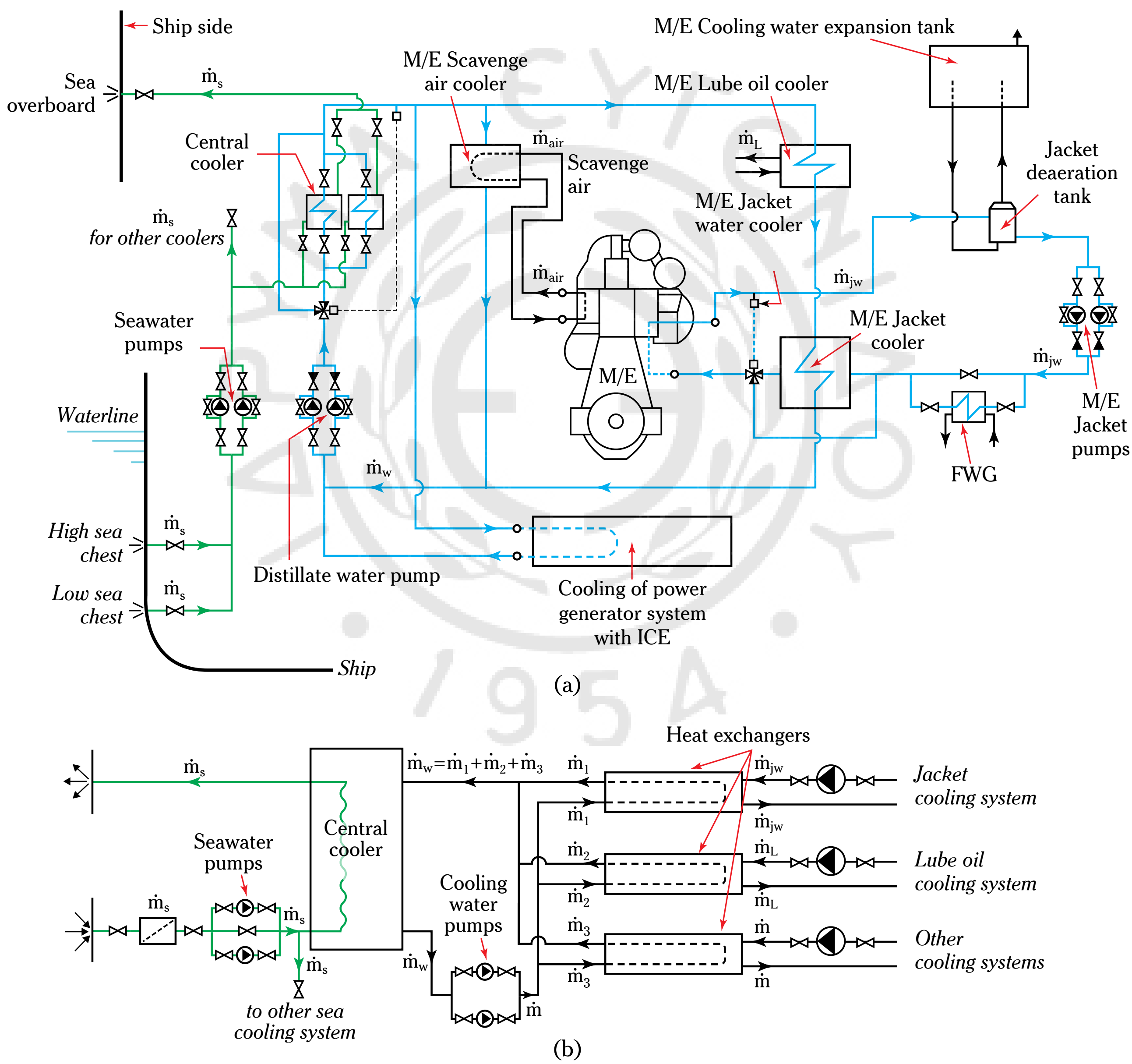


Fig. 8.36

(a) Typical arrangement of seawater pumps and cooling water circulation pumps in a ship's central cooling system.
(b) Typical arrangement of a central cooling system with seawater and freshwater pumps.

trifugal, usually in vertical layout (fig. 8.37), single stage of simple (fig. 8.38) or double (fig. 8.39) suction impeller, achieving higher flow. Packings with gland follower or mechanical seals are used for pump sealing, while filters are installed in the suction of the pumps and are used for their protection against particles that would cause damages to the impeller. For the protection of the mechanical seal against the sand which may be carried by seawater, there is a small pipe from the pump discharge that supplies clean water to the stuffing box, preserving the long and smooth operation of sealing.

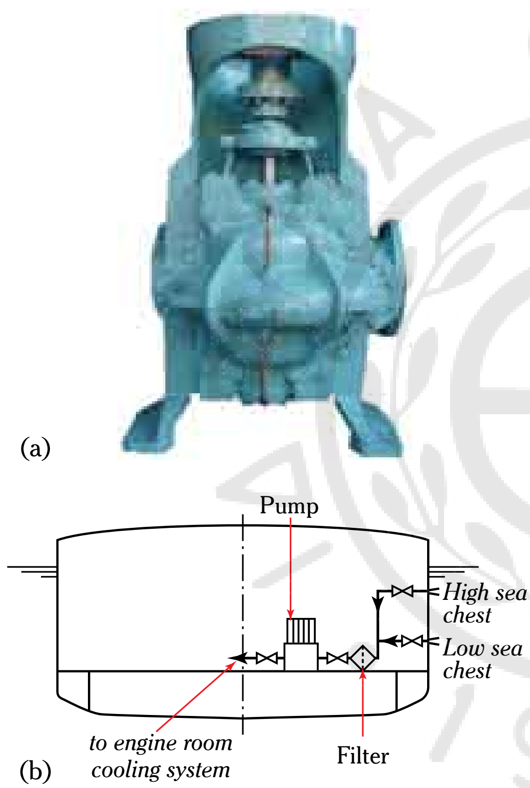


Fig. 8.37
(a) Seawater circulation pump in vertical arrangement, (b) sea chest arrangement on the side of the ship.

Pumps are installed in the lowest part of the engine room, so that the best suction conditions are achieved. Furthermore, when the ship is in loaded condition, the suction of the pump is performed through the high sea chest, whereas when the ship is in unloaded condition –empty– the suction occurs through the low sea chest. The differentiation for the suction change becomes noticed by pressure gauges at the pump discharge, which, during the transition stage from loading to unloading and vice versa, increases or decreases accordingly. The low sea chest choice ensures that air is not driven by the seawater when the ship sails in rough sea and is rolling. On the contrary, with the high sea chest choice, taking in mud from the sea bed that could be entrained by the seawater flow is avoided. Especially the choice of high sea chest helps when the ship sails in low depth areas where the entrained materials could cause not only fouling to the heat exchangers but also piping and filter blockage.

The discharge pressure adjustment of the ship system to the desirable level is achieved not only through suction selection but also by the valve (bypass valve) that is located at the bypass system

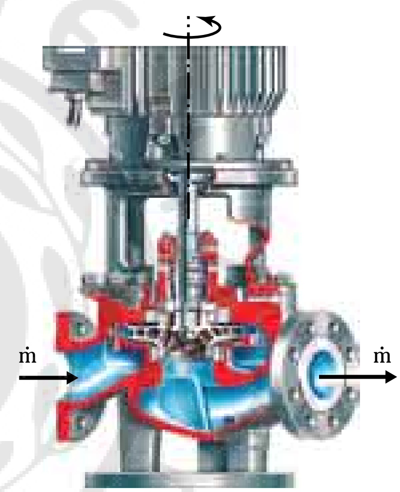


Fig. 8.38
Section of pump in vertical arrangement with single suction impeller.

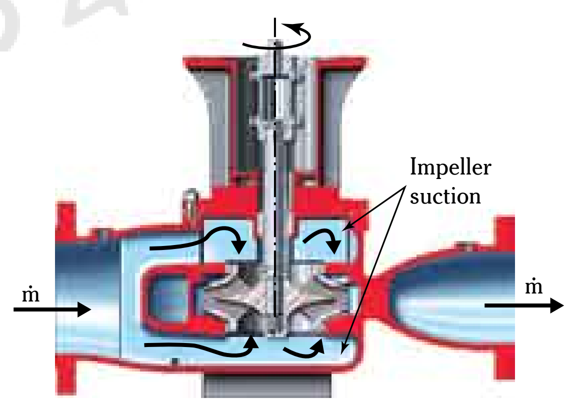


Fig. 8.39
Pump cross section in vertical arrangement with double suction impeller.

and connects suction with discharge (fig. 8.40). By opening or closing the bypass valve, the supply of water that recirculates increases or decreases, so that the pressure of seawater decreases or increases accordingly (figs 8.35 and 8.36).

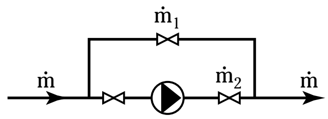


Fig. 8.40
Pump in line with bypass arrangement.

When the machinery and the exchangers' cooling is obtained by fresh water, which is later cooled by the sea in the central coolers, two independent systems are created. One is cooled with seawater and another with fresh water. In each of these, two ***seawater circulation centrifugal pumps*** and two ***fresh water circulation pumps*** driven by electromotor are installed.

The ***seawater circulation pump*** is similar to the one described in the previous paragraph and, besides the central cooler, it may circulate the seawater for the cooling returns of steam from the boiler to a cooler under atmospheric pressure. The returns derive from lubricant oil and fuel oil preheaters or from oil fuel tanks preheating systems.

The ***fresh water circulation pump*** is vertically positioned, centrifugal, single-stage with a hydraulic balanced impeller. The system's water flow is adjusted by a three-way valve. This valve either circulates the fresh water in the central cooling system of the ship or sends the water to the central heat exchanger where it is cooled by the seawater. Hence, the quantity of the water that flows through the heat exchanger depends on the flow control valve position.

The three way valve is controlled by an automation with actuator, usually pneumatic (compressed air), which receives the operation command from a temperature sensor that is located in the fresh water system. So, when the water temperature increases, the valve allows more water to pass through the central cooler, whereas when the temperature decreases, it changes position so that the water quantity that is supplied to the system through the bypass is larger.

8.4.3 Lubricating oil transfer pumps (fig. 8.41).

These pumps are used to transfer lubricants from

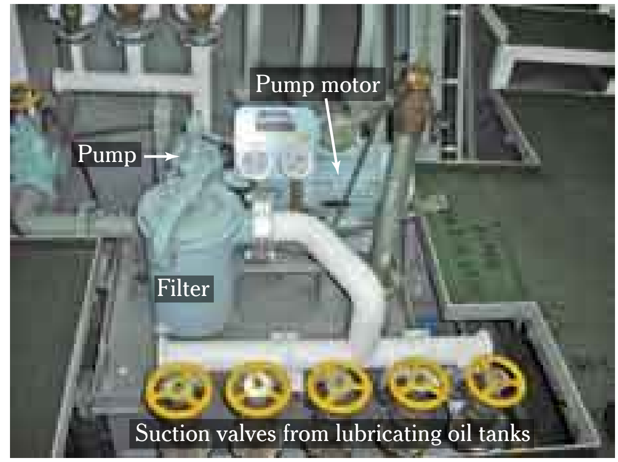


Fig. 8.41
Lubricating oil transfer pump.

storage tanks to tanks in various places of the ship, where the suction is performed by the lubrication pumps of the machineries (fig. 8.42).

They are rotary gear pumps, driven by an electric motor. One transfer oil pump is usually installed on board ships, but it is possible for more pumps to be fitted in the installation.

Alternatively, apart from using lube oil transfer pumps, the transfer of lubricant to machineries from the storage tank can be made by centrifugal purifiers through appropriate bypass in the lubrication system.

This mode of transfer is aided by a pump that may be attached or moves independently but is used to supply the lube oil to the centrifugal purifier.

Due to the wide variety of qualities of lubricants used in the different kinds of ship machinery and the risk of mixing them by mistake through incorrect handling during the transfer to the tanks, particular attention is required.

8.4.4 Stern tube lubricating oil pump.

This is an electric motor-driven rotary screw, gear, or vane pump used to circulate lubricating oil through a vessel's stern-lube bearings and seals when these components are oil lubricated. The oil suction of the pump is performed from a specific tank, the stern tube lubricating oil tank for the tail shaft, and the discharge is performed into the stern tube structure (hopper of the stern) (fig. 8.43).

The pressure on the discharge side of a stern-tube lubricating-oil pump depends on the height of the ship's draft. The "balance" of the lubricant, so

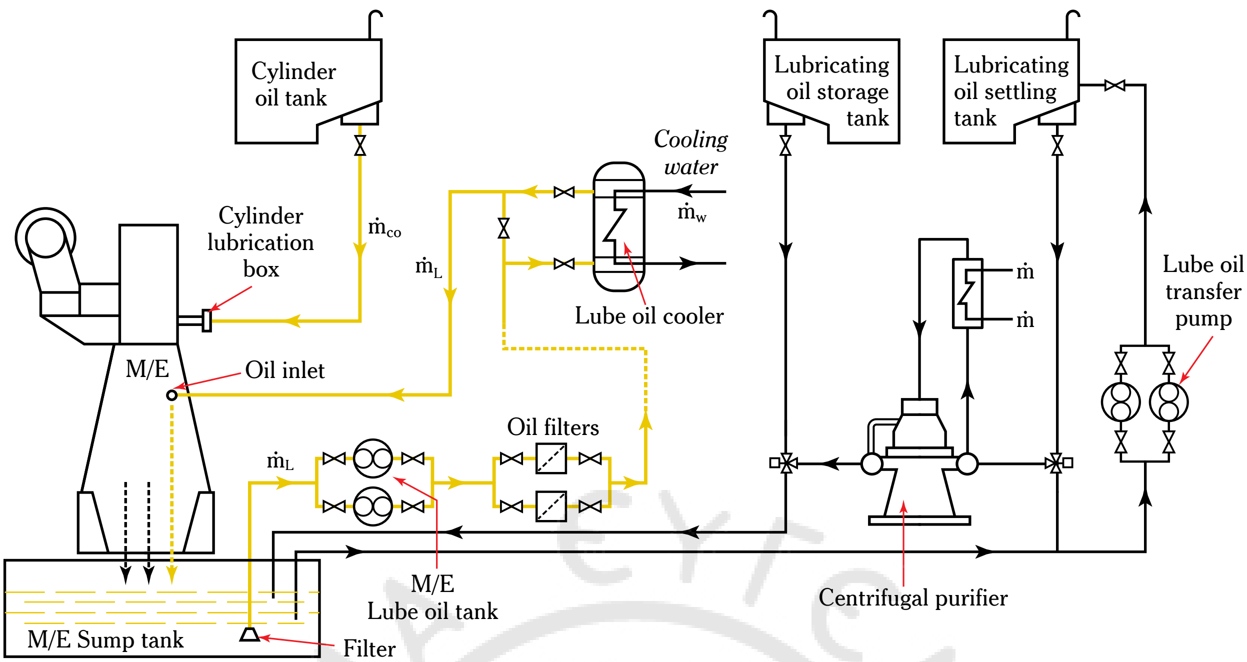


Fig. 8.42

Piping system of lubrication, clarification and transfer in the engine room of the ship.

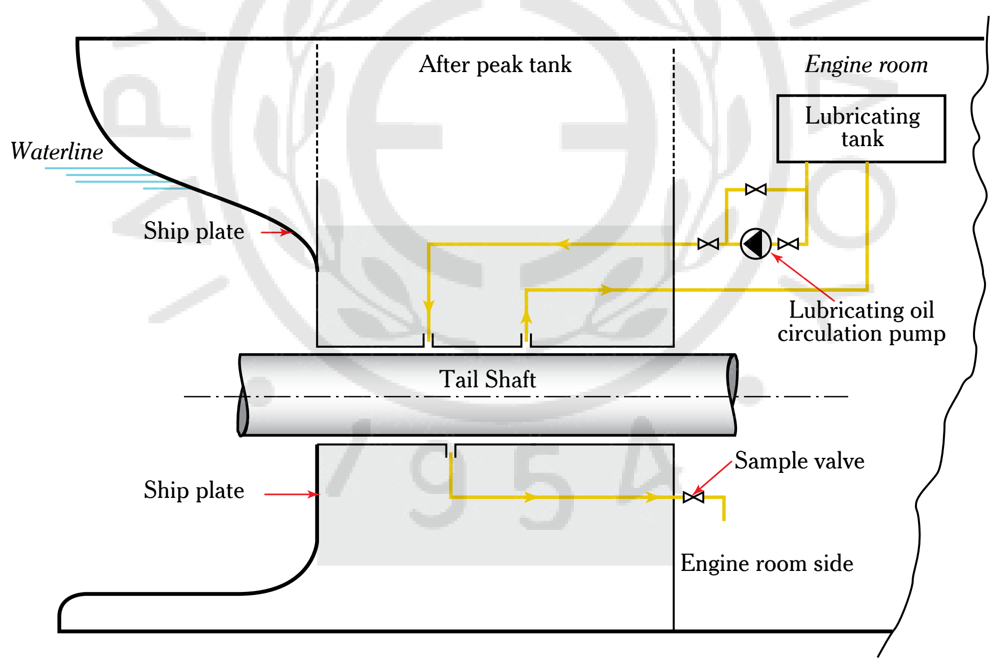


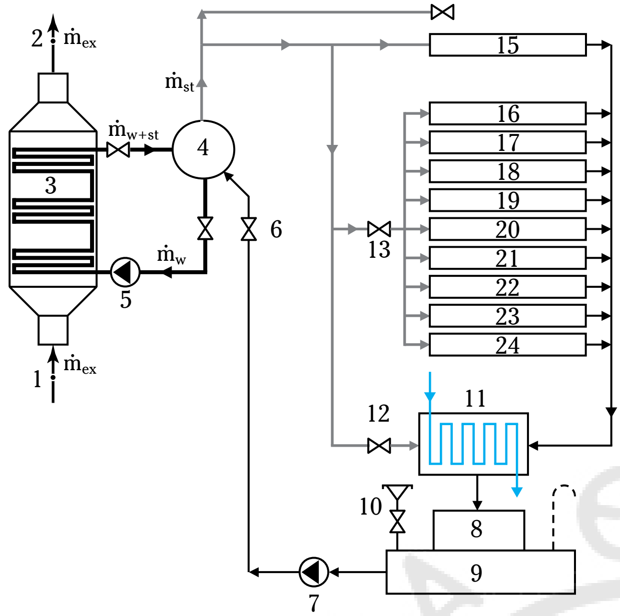
Fig. 8.43

Stern tube lubricating system and oil circulation pump.

that it does not leak into the sea, is achieved by two oil tanks established at different heights in the engine room (or, if one tank is used, by the elevation of the head tank that is connected to the system). So, the elevation depends on the loading condition of the ship. It is noted that such provision is not required on ships that have seawater-lubricated stern-tube bearings and sealing of the propeller shaft.

8.4.5 Exhaust-gas boiler water circulation pump.

In ships with ICE, exhaust-gas boilers (heat ex-



1. Exhaust from main engine.
2. Exhaust outlet.
3. Exhaust gas-boiler.
4. Steam boiler.
5. Exhaust gas boiler circulation pump.
6. Feedwater control valve.
7. Boiler feed pump.
8. Deaerating feed water tank.
9. Feedwater tank.
10. Water filling.
11. Steam condenser.
12. Excess steam valve.
13. Steam pressure regulating valve.
14. Soot blowing.
15. Fuel oil heaters.
16. Fuel oil tanks.
17. Steam tracing for fuel pipes.
18. Heaters for fuel purifiers.
19. Lubricating oil tanks.
20. Heaters for oil purifiers-clarifiers.
21. Water heaters for jacket of main engine.
22. Heating coils for settling tank.
23. Water heater.
24. Various uses.

Fig. 8.44
Simplified heat recovery system of exhaust boiler.

Fig. 8.45
Installation of two exhaust gas boiler pumps.

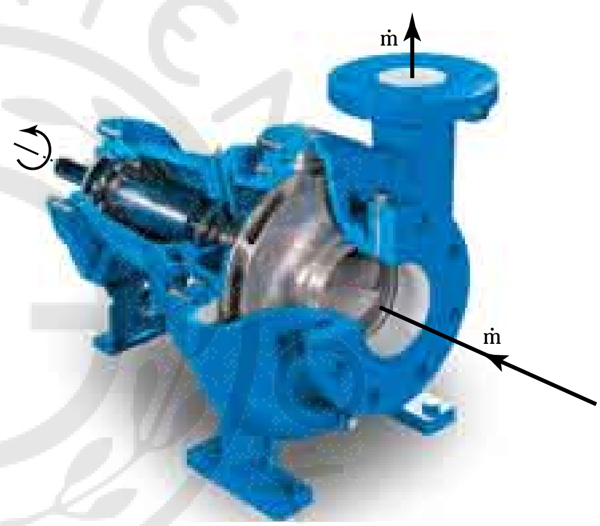


Fig. 8.46
Exhaust gas boiler circulation pump.

changers) are fitted, in which the exploitation of the exhaust heat energy (WASTE-HEAT) is utilized to generate steam for various purposes (fig. 8.44), such as:

1) heating of heavy fuel oil in tanks,
2) water heating and
3) the operation of steam turbines that drive electric generators.

Centrifugal pumps are used for the circulation of water to the exhaust gas boiler and to the steam boiler in order to generate the steam (fig. 8.45). The suction is performed from the steam boiler and the discharge into the exhaust gas boiler. The pumps consist of single-suction impeller (fig. 8.46), and are usually single stage, driven by an electric motor. The pump's sealing is achieved by mechanical seal.

8.4.6 High pressure hydraulic systems oil pumps.

Various types of ***hydraulic system pumps*** are frequently used to pressurize the circulation of hydraulic oil in order to operate the installed machinery at the vessel's hydraulic systems (fig. 8.47). Machinery which operates by circulation on high pressure hydraulic oil includes:

1) ***steering gear***,
2) ***anchor winches***,

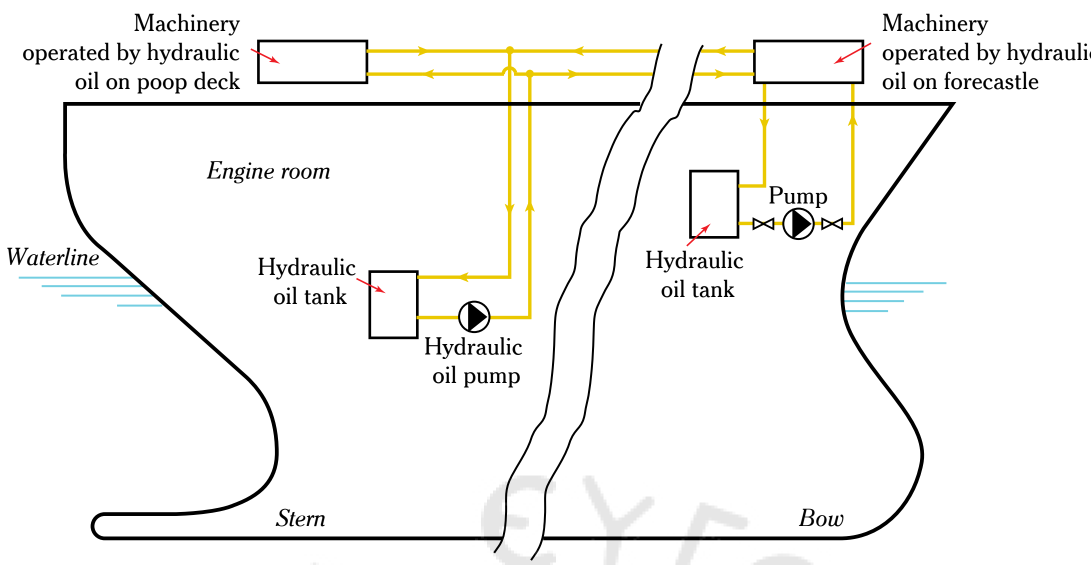


Fig. 8.47
Typical hydraulic oil system arrangement.

3) ***mooring winches***,

4) ***cargo hold cranes***,

5) ***auxiliary cranes*** for the lifting or the moving of heavy objects (for example, spare parts, provisions, etc.),

6) ***hatch covers hydraulic rams***,

7) ***valve actuators*** operating by the pressure of hydraulic oil, achieving the remote control of the valves that are installed in the handling systems of liquid cargo, sea piping systems or elsewhere, and thus the fluid flow to the relative system,

8) ***liquid cargo pumps*** driven by hydraulic oil pressure, which is developed in a central hydraulic power unit.

The power that drives the hydraulic pump is given either by electric motors or by small ICE (diesel engines). The pump types that are used in order to achieve the high pressure requirements of the fluid for the operation of machinery are the positive displacement rotary pumps. In particular, pumps that are frequently used as hydraulic pumps are the vane pumps, screw pumps and variable displacement piston pumps. In detail, at constant pressure hydraulic systems, the circulation and increase of hydraulic oil pressure is frequently achieved by one or more ***variable displacement (delivery or discharge) pumps***. By these pumps, which are controlled by an adjustable automated control system, if the system pressure exceeds the required pressure, discharge pressure is reduced. This reduction is achieved by varying the stroke of the piston of the pump, so that, by depressing less amount of oil, the pressure is brought to the desired value set by the control automation. Respectively, if a pressure drop is caused because the system requirements increase or the load of machinery already operated on the system is changed, the automated control system acts on the adjusting mechanism of the piston stroke, and achieves the pump discharge pressure increase to the desired value. By this operational mode, namely the change of the capacity of the pump cylinders, the flow of the pump is changed and is therefore adjusted to match the demand on the hydraulic system.

The pumps that operate the electro-hydraulic rudders belong to the category of variable displacement pumps. Other pumps of the same category, which are installed in other systems of the ship, operate in similar ways. Rudder pumps are used below as an example that can help us understand this type of pumps, since rudder pumps are found in all vessels, in the steering gear system.

Variable displacement pumps are used to supply hydraulic oil for the operation of the steering gear. The steering gear consists of hydraulic pistons or chambers with stable blades and vanes which are attached to the rudder stock, so, by turning it, the rudder turns to the desired position, achieving the rapid response of rudder in movements required to ***alter the ship's course***. Thus, for the supply of the hydraulic oil, pumps are used which operate continuously but under variable discharge pressure (namely, we can change the pressure of suction and discharge while the pump is rotating in the same direction with the same speed). The term "variable discharge" refers to the ability of the pump, while operating continuously, i.e., always to the same direction, to convert the suction to discharge and back by the effect of a control mechanism.

In a steering gear system, the installed variable displacement pumps are usually two, ensuring that the rudder will operate in case of failure with one of them, and that the steerage will be supported when maintenance is carried out in any of these two pumps. Also, they have the ability to operate either separately, fully serving the steerage conditions, or in parallel, increasing the supply of hydraulic oil to the system and the speed response to the commands which are given by the steering wheel on the bridge for the movement of the rudder.

In a constant-pressure hydraulic system the pumps that are used to supply the hydraulic oil at constant pressure are variable-displacement rotating piston pumps, where the alteration of discharge to the hydraulic system is achieved by the action of the control mechanism in the stroke of the pump pistons. So the amount of oil displaced by the pump can be varied from zero to maximum, while the discharging capacity is specified by its constructional characteristics. The main variable displacement pump types that are used in ships steering systems are the ***floating ring***, the ***shaw-plate*** and the ***slipper-pad***.

1) Floating ring hydraulic pumps.

In floating ring pumps, or Hele-Shaw type pumps (figs 8.48 and 8.49), the change of the suction to discharge and vice versa is controlled by pressing or pulling a control spindle (T) connected to a guide block (S) that changes the position of the moving floating ring (O) inside the pump casing (A), which is closed by two covers (B, C) (fig. 8.48). The floating ring can be moved on the vertical plane of the axis of rotation (L), in two diametrically opposite directions in the horizontal plane of the axis. In the center of the casing rotates the cylinder body (or block) (K), which is rotated by the shaft that is attached to the pump motor shaft. The body (K) consists of cylinders within which the pistons (M) reciprocate, while the entire unit rotates around

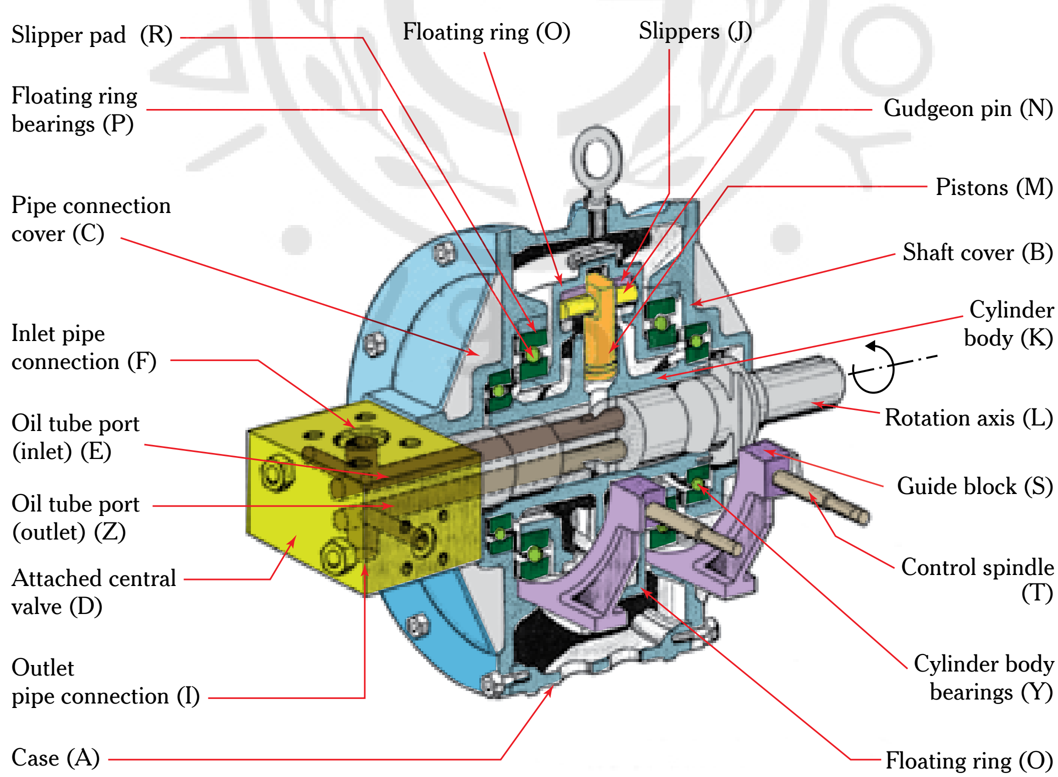


Fig. 8.48
Hele-Shaw floating ring hydraulic pump.

the central valve (D). At the edge of the pistons, towards the pump periphery, there are gudgeon pins (N) which are fastened on slipper pads (R) by means of gudgeon pins and shifted on the sliding slots (J) of the floating ring. The gudgeon pins entrain the pistons to be moving within the cylinders, along the stroke caused by the path of the gudgeon pins in the floating ring (O). The support of the cylinders body is performed by ball bearings (Y) applied on both sides of the pump casing, while that of the floating ring by other ball bearings (P). On the cover (C) is attached the central valve (D) which is connected to two oil tube ports (E and Z) into the pump center, and is a continuation for the connection of piping (F and I) of the inlet and outlet of oil. Through the pipes, which, depending on the operation of the pump, are altered either to suction or discharge pipes, the oil is supplied to the hydraulic system (e.g., steering gear system).

Hence, the pump pistons, that are arranged in a radial configuration relative to the pump rotation axis and rotate in conjunction with the body of the cylinders, reciprocate, in a way that depends on the position of the floating ring.

The movement of the floating ring is performed horizontally on the axis xy (fig. 8.49), so that:

1) When the floating ring is moving and ***maintains the same center to the valve*** [fig. 8.49(a)] where are the suction-discharge ports of oil are located, then the plungers are rotated keeping the same distance from the center of rotation and no suction or discharge occurs.

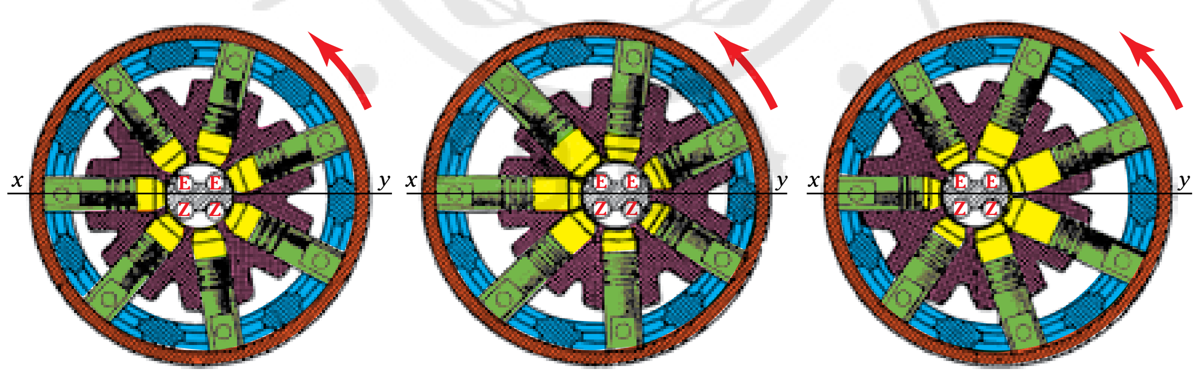
2) When the floating ring with slippers ***is shifted to the left*** [fig. 8.49(b)] on the axis xy, the cylinder block is correspondingly shifted. The pistons, under the influence of the motion of the slippers on the gudgeon pins of the floating ring, begin to reciprocate. This reciprocation has the effect of suction from the ports E and the discharge through the ports Z.

3) Accordingly, when the floating ring ***is shifted to the right*** with the same direction of rotation [fig. 8.49(c)], the reciprocation of the piston continues, but now the suction is realized from the ports Z and discharge through the ports E.

According to these movements, apart from the suction and discharge, the flow from the pump is also determined. So, the position of the floating ring from the center of rotation to the maximum distance affects the stroke of the pistons which is varied to achieve the required oil provision to the corresponding discharge of the pump.

2) Swash plate hydraulic pump.

The swash (variable inclination) plate or Waterburry type pumps (fig. 8.50) are one more type of pumps where without interrupting their operation we may change the oil provision or alternate be-



(a) (b) (c)

E: Oil inlet port Z: Oil outlet port

Cylinder *Cylinder body* *Slippers*

Pistons with gudgeon pins *Floating ring*

Fig. 8.49

Cross section of floating ring pump. (a) Floating ring moves concentric. (b) Floating ring is displaced to the left. (c) Floating ring is displaced to the right.

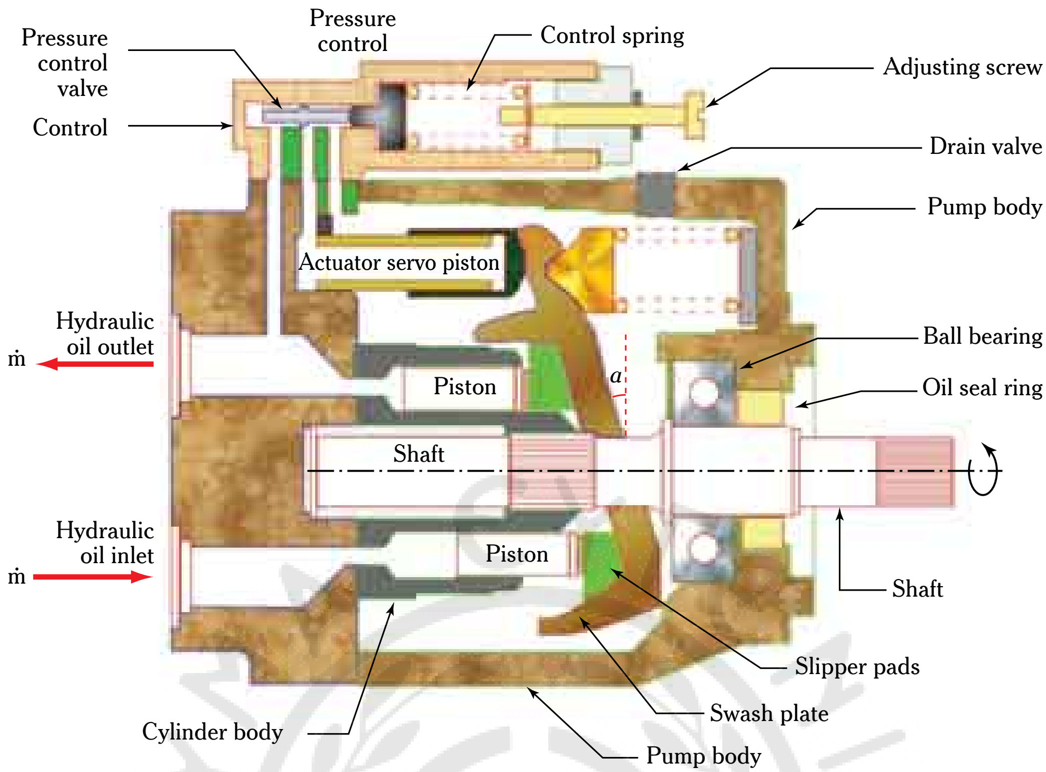


Fig. 8.50
Waterburry swash plate hydraulic pump.

tween the suction and discharge at the pump outlet ports.

The basic characteristic of these pumps is the arrangement of cylinders and pistons. Namely, the body of the cylinder, with the liners and pistons, is placed in an axial arrangement, and is rotated by the rotation shaft of the pump motor (fig. 8.51).

On these pumps, with the start of rotation and during operation, the motor rotates the cylinder block and entrains the pistons with it.

The connecting rods of the pistons are rounded on both ends, at the points of their connection, creating flexible joints. The one side of the piston rod is connected to the piston and the other on a slipper pad that slides on the swash plate. During operation the cylinders block is rotated and entrains the pistons. Because the slipper pad is connected to the piston and it slides on the swash plate surface, any change on the inclination is transmitted to the pistons and causes the axial reciprocation of the pistons relative to the pump rotation axis.

The realization of axial reciprocation of the piston is caused by an external control spindle or a servo mechanism, which is connected to the swash plate, causing changes in the tilt of the plate.

When the swash plate tilt control shaft is in mid position, the swash plate rotates vertically to its rotation shaft and no oil suction or discharge is performed at the inlet and outlet ports of the pump.

By the influence of the control mechanism that transfers the movement of the bridge steering wheel to the pump swash-plate, the swash-plate starts tilting. The tilt is transferred to the pistons as reciprocal motion, since, through the slipper pads, they follow the changes in inclination of the swash plate.

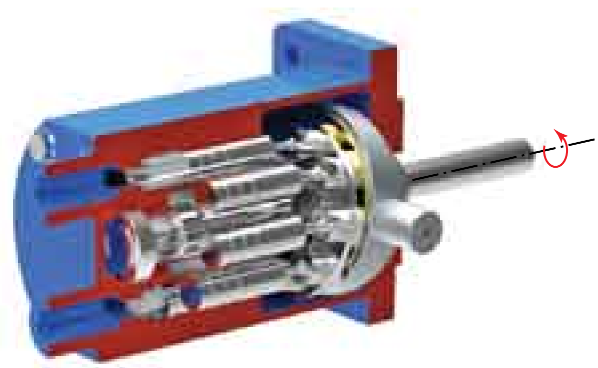


Fig. 8.51
Section of Swash plate hydraulic pump.

Hence, as the pistons reciprocate, any change in tilt of the swash plate causes variations of the pistons stroke. In this manner, the flow of hydraulic oil is alternated on the pump ports, either as suction or discharge. The ports are arranged in an arc internally of the pump casing, on the surface where the cylinder block is tangent to the cover. Therefore, the suction or discharge of each piston separately is achieved on the half of the circular path depending on the inclination of the swash plate.

3) Slipper pad hydraulic pump.

The slipper pad hydraulic pump is the evolution of the tilting swash-plate pump. Their operating principle is same, but they differ in the shape of the pistons (fig. 8.52). The pistons in this pump are elongated to form a compact body without a connecting rod. The one end of the piston is rounded and abuts on laminates which slide on the variable tilt plate so as to accomplish their free axial motion during the rotation of the pistons assembly with the cylinder body.

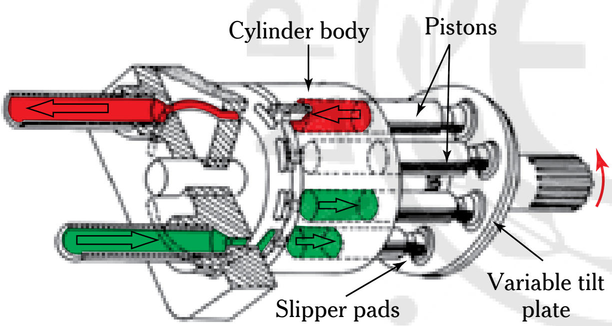


Fig. 8.52
Slipper pad hydraulic pump.

Given the compact construction of the pump, higher increase of discharge pressure of hydraulic oil is obtained, serving the functional requirements of new rudders and of hydraulic systems of stabilisers on ships.

Other types of pumps that are frequently used to provide the hydraulic oil in systems are gear pumps, vane pumps, multiple screw pumps or constant flow rotary pumps with pistons. The operation of these pumps may be continuous or interrupted, depending on the requirements and the system design.

In systems where the pump operation is continuous, in order to maintain the pressure at the desired level, the flow of hydraulic oil to the network is regulated by means of an automatic valve that is installed in the discharge of the pump. During pump operation, the suction is performed from a tank containing hydraulic oil, while the discharge is performed to the piping system. When the system pressure exceeds that required by the installed equipment, the control mechanism of the automatic valve is activated and opens to return the hydraulic oil to the suction tank. In systems with intermittent pump operation, the pump is activated when the system pressure is reduced relative to that which is set to the control mechanism, and stops when the pressure exceeds the corresponding (desired) high pressure point.

8.4.7 Fire pumps.

There are two types of fire pumps on vessels, the fire and general service pumps installed in the engine room, and the emergency fire pumps.

In more detail:

1) The ***fire pump*** takes suction from the sea chest which is installed on vessel's side and discharges seawater through a fixed fire main line to hydrants located throughout the vessel. Usually vessels have at least two fire pumps installed at the lowest point of the engine room ensuring the sufficient supply at suction (fig. 8.53), while each one is capable of delivering a minimum capacity of seawater that is required by the design features of the fire system (fig. 8.54).

Fig. 8.53
Fire and general service pump in the engine room.

Single-stage and, in some cases, two-stage centrifugal pumps may be used in service of the fire system and are generally driven by electric motors. The shaft sealing is either by mechanical seal or by packings with gland followers. Apart from the fire line, they may be used for the sea supply to other lines of the ship, like the general service line. The suction of the pumps, ***only in case of emergency***, can be performed by the engine room bilges through the bilge line, or through the ***emergency valve***. Also they may be used to provide seawater to some auxiliary machinery, including heat exchangers for preheating the washing water of the tanks in tankers.

2) ***The emergency fire pumps*** are installed outside the engine room, usually in a well at the stern the ship in the steering gear room behind the rudders, achieving optimization of the suction conditions. The emergency fire pumps are in vertical or horizontal arrangement and driven by either an electric motor or by suitable diesel engines. These pumps are centrifugal, with a single-suction impeller. In the pump housing, to ensure the initial suction, an attached prime pump is installed, which is turned off or running idle when the discharge pressure increases.

8.4.8 Bilge pumps.

Bilge pumps are used to remove the liquids that are collected inside the engine room from various small leakages (fig. 8.55). The discharge of the pump, due to the quality of liquids, which may contain, apart from water, small quantities of oil products, is performed in an ***oily waste holding tank***, or to the sea, through the ***oil water separator*** and the ***oil detection motoring system*** (ODM).

The purpose of the bilge pump discharge through the ODM is for the oil content of the discharged liquid not to exceed the limits required by MARPOL regulations, which dictate in detail the management of these fluids. Bilge pumps may be crank piston-plunger pumps with the crank driven by electric motor (fig. 8.56), rotary lobe, or centrifugal.

The bilges are positioned at the lowest point of the engine room; consequently, the suction of the pump is at higher level than the level of the liquids that are being sucked. Therefore, the suction might start by piston pumps as well as rotary pumps without the installation of a priming mechanism. But, in centrifugal pumps, a priming pump, that eliminates the air from the suction and achieves the filling of the system with liquid from the bilges, is necessary. In this case, vane pumps or liquid piston pumps are used as vacuum pumps (commonly priming) to assist the initial suction of the pump. To eliminate the need for a separate priming pump, self-priming centrifugal pumps are sometimes used in bilge service

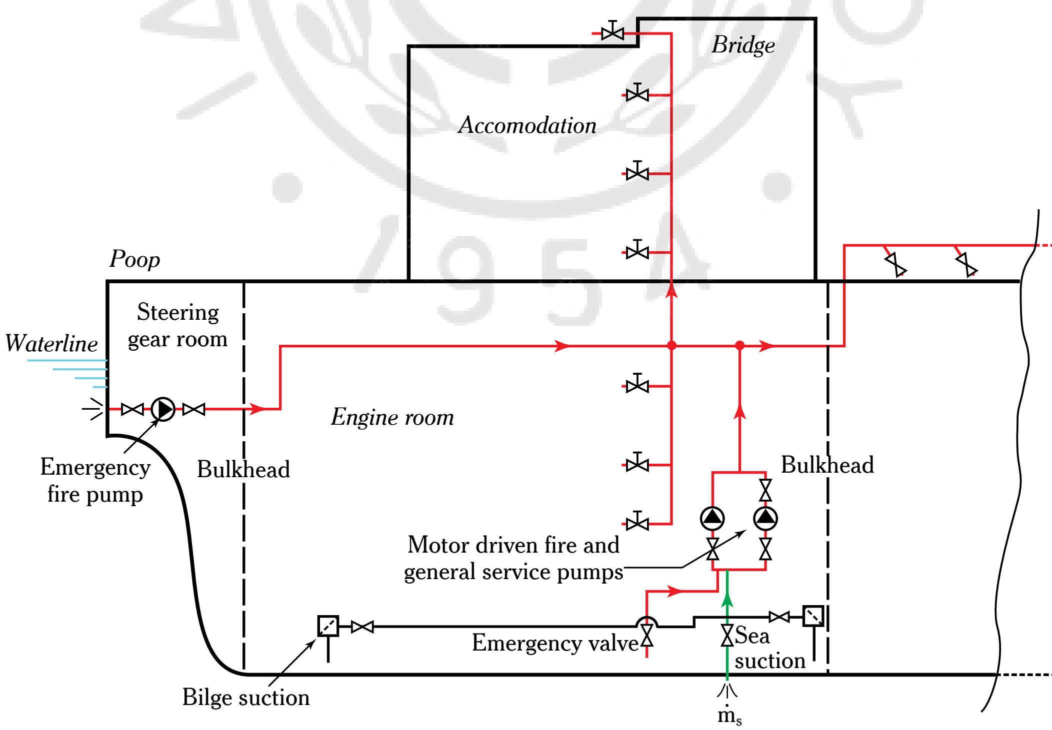


Fig. 8.54
Typical fire line in ship.

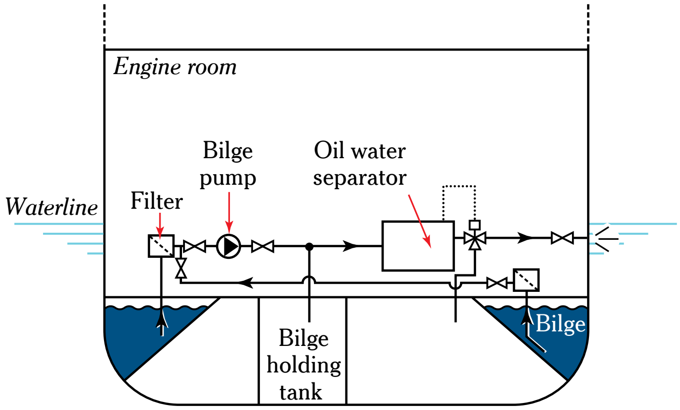


Fig. 8.55
Typical arrangement of a bilge pump.

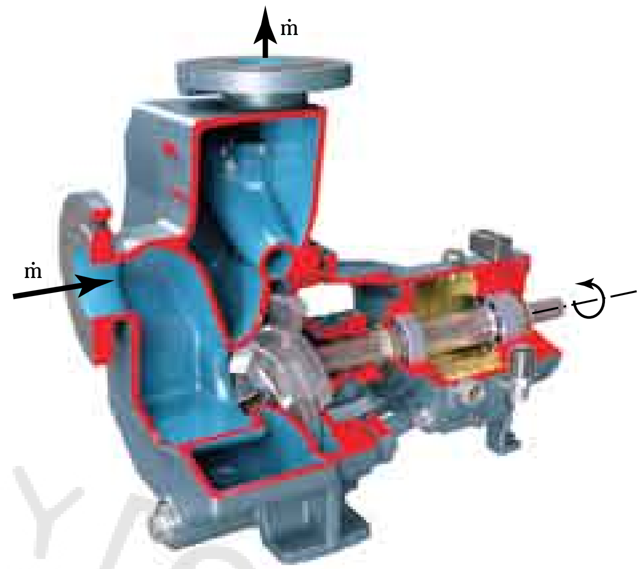


Fig. 8.57
Self priming centrifugal pump.

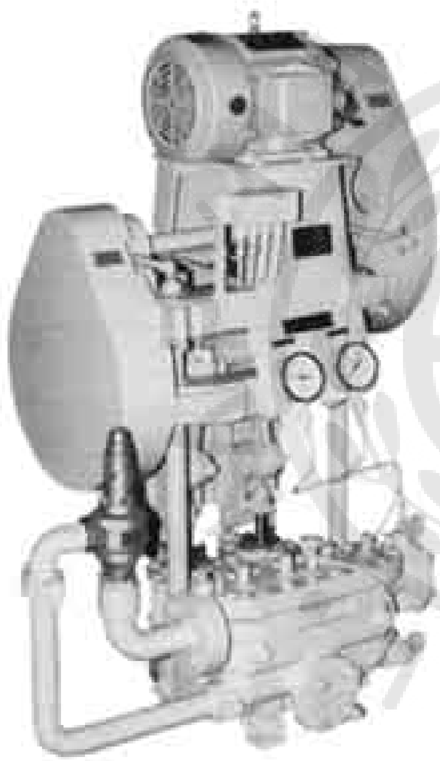


Fig. 8.56
Piston bilge pump.

(fig. 8.57). This pump has a casing with an enlarged suction chamber that retains liquid whenever the operation of the pump is stopped. Self priming centrifugal pumps are used only if the length of the line from the bilge is short. Otherwise, another type of pump or another device, which can manage larger quantities of air from the suction, could be used.

8.4.9 Waste pumps.

Waste pumps are used to manage and remove oil waste that is collected in the corresponding ship tanks. The content of the tanks derives from the discharge of the centrifugal purifiers and the bilge liquids that are discharged through the oil water separator system when the three-way valve closes to avoid discharge of oil liquids into the sea. Waste pumps must be capable to handle viscous liquids; so lobe or helical rotor pumps are used. These are in horizontal arrangement and powered by electric motor. Pump liquids are discharged in a waste tank in order to be incinerated on the ship or to be delivered onshore to reception facilities in ports.

8.4.10 Ballast pumps.

Ballast pumps are used to transfer seawater into and out of a vessel's ballast tanks. So, as the ballast water is transferred, the required ship's draft is adjusted, the ship's stability during cargo loading and discharging is maintained, and the ship's trim is controlled. Ballast pumps may also be used during a voyage to exchange water contained within ballast tanks to prevent the introduction of nonindigenous aquatic species into coastal and inland waterways.

Therefore, ballast pumps must have the capability to take suction either from a sea chest or from ballast tanks that are being emptied. Additionally, the seawater discharged by a ballast pump may be used as the motive fluid in eductors that are provided to strip ballast tanks.

Ballast pumps, due to the large capacity of the ballast tanks and hence of the amount of seawater that they have to manage, must have a high flow rate. So, the type of pumps which are used are dynamic pumps that have high flow rate and are powered either by a motor or steam turbine. The arrangement of the pumps may be vertical (fig. 8.58) or horizontal, and the flow through the pump can be centrifugal, axial or mixed flow. The type of impellers that are used depend on the flow. Hence, in centrifugal pumps we have double suction impeller, while in axial flow pumps we have propellers.

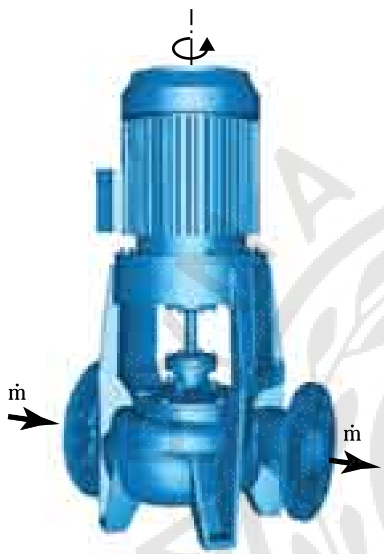


Fig. 8.58
Ballast pump in vertical arrangement.

Such type of pumps require that the installation as well as the sea suction on the ship's side must be at a low point within the ship to achieve better suction conditions. In order to prevent the loss of suction during pump operation, the low suction point may be connected to the filling system, ensuring a constant flow of fluid into the pump. If the ship ballast pumping system is motivated by hydraulic motors from a central hydraulic oil supply unit, the ballast pumps centrifugal pumps are of similar type, submerged into the tank.

8.5 Auxiliary service pumps.

Auxiliary service pumps are used in systems where the product of their functional process concerns either the general operation of the ship or service to the crew and passengers. These include the following:

8.5.1 Fresh water plant pumps[7].

In ships with ICE, a network is developed for the exploitation of the cooling water temperature of the jackets (main engine cooling water). This hot water moves from the jacket pump and provides the thermal energy for the operation of the fresh water generator. But, for the supply of seawater within the evaporator and the removal of distillated water during the production process, the seawater supply pumps are used for the operation of ejectors (or eductors) and the distiller pump.

1) Seawater supply and ejector pumps.

This pump is driven by an electric motor, and it is a single stage centrifugal pump. The suction is performed by a particular valve installed in the ship's side. This valve is used only by this pump in order to preclude the possibility of mixing the seawater supplied to the evaporator, which could cause contamination from harmful elements which are drawn by the seawater. Also, the suction through this separate valve does not affect the operation of FWG which could be affected by the alternations between high and low suction. The pump is installed low in the engine room and discharges the seawater to the ejector that creates the vacuum within the chamber of the evaporator and to the elements of the heat exchanger for the production of distillate water. The same seawater is usually used as a cooling medium at the evaporation condenser of the fresh water generator.

Usually one pump is installed at the sea supply system for the FWG, which alternatively can be replaced by the fire and general service pump, when immediate use of the FWG is needed and the seawater supply pump of the ejectors is out of order due to damage.

2) Distiller-feed pump.

This is a single stage centrifugal pump used for extracting the distillate water from the bottom of the evaporator condenser. To maintain the suction to the

[7] Fresh water generator, see: Dagkinis, I. and Glykas, A. (2015). *Βοηθητικά Μηχανήματα Πλοίων* ("Ship Auxiliary Machinery"*)*. Athens: Eugenides Foundation, 2015, Ch.12.

pump, there should be water within the condenser at a constant level. Otherwise, fluctuations in the water level or absence of water within the condenser will interrupt the flow of water and, as a result, air enters, because of the vacuum, inside evaporator. This leads to a decline in the efficiency of FWG and impairment of the needed balance in the process of distillation. The level in the condenser is achieved by controlling the pump flow to the discharge valve. The pump discharge leads distillate water to storage tanks. Depending on the type of evaporators used in the ship, that is, either flash evaporators or rising membrane evaporators, the respective centrifugal pumps are used, that have the same functional characteristics.

8.5.2 Potable water pumps.

Potable (or drink) water pumps are used to provide the potable water to faucet sinks and potable-water fixtures (coolers) located throughout the vessel (fig. 8.59).

Electric-motor-driven single-stage centrifugal pumps with single suction impeller are often used in this application. A potable-water pump typically takes suction from a vessel's potable-water or domestic tanks and discharges freshwater either directly to the piping system or to an air accumulator pressure tank, sometimes referred to as a hydrophore or a hydropneumatic tank, and, by pressurized air, maintains the potable water system under pressure. In order to maintain satisfactory pump suction conditions, it is installed at a distance from the storage tank and always as low as possible from the free water level surface inside the tank.

The air accumulators are used to prevent any short starts and stops of the pump. Through them, the fresh water piping system is maintained in constant pressure, until the point where an automatic control system is adjusted to start the pump and to restore the water pressure in the air accumulator to the desired setting point. Otherwise, when the pump is directly connected to the network, the water pressure fluctuations due to frequent starts and stops of the pump are prevented by a recirculation piping system which maintains constant flow.

8.5.3 Hot water circulation pumps.

Besides the water that is handled by potable water pumps, water is also provided through heaters to sinks, showers, and other hot-water fixtures located throughout a vessel. The supply of hot water is achieved by single-stage centrifugal pumps driven by motor. The hot water pumps are running continuously and through a recirculation system they recirculate unused water contained in the hot-water distribution piping through the heater so that the water remains hot.

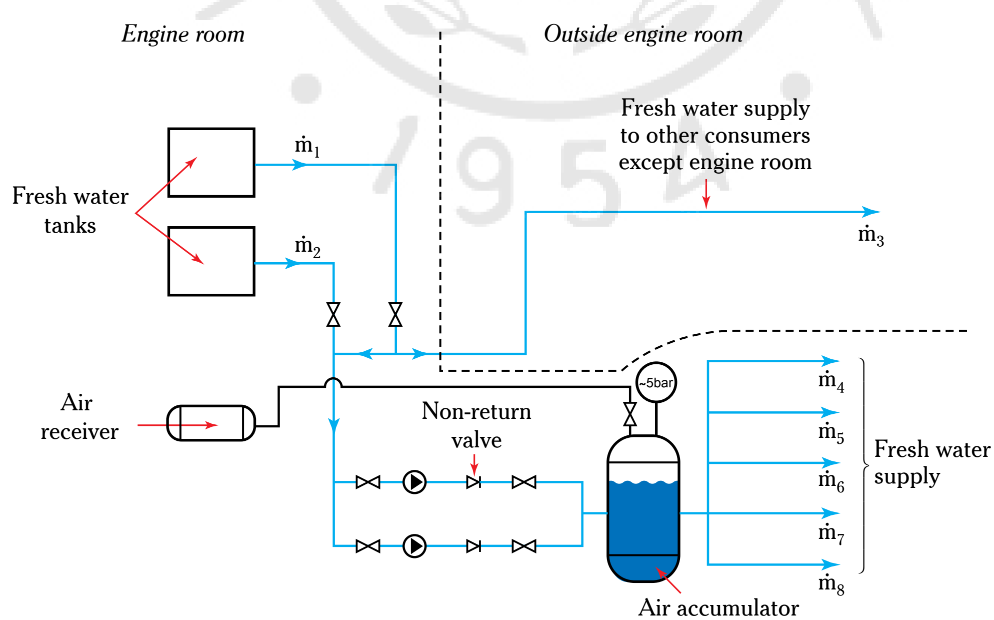


Fig. 8.59
Typical freshwater supply system.

8.5.4 Sanitary pumps.

Sanitary pumps are used for water supply to toilets on board. The water is either seawater or fresh water from the ship's storage tanks, depending on the structure and operation of the network. The type of pumps used is single stage pumps with a motor. To avoid frequent starts of the pump, air accumulators are used with air pressure, like in the network of drinking water pumps.

8.5.5 Sewage pumps.

The management of sewage on board is carried out either by collecting organic waste in tanks, or, as in modern constructions, by processing it with appropriate systems.

In the first case, the organic waste is collected in a tank and returned to the sea away from the coasts (as defined by applicable regulations) or delivered to shore reception facilities for further processing. The handling of wastewater is performed by sewage pumps that are of the centrifugal single stage type, driven by an electric motor. In some plants, the pumps are submerged in the tank and discharging wastewater through a pipe installed on their discharge.

In the second case, that of the use of marine sanitation devices on the ship, the wastewater collected is subjected to treatment. Depending on the processing system used, wastewater may be slurried or agitated in the presence of excess air and finally discharged to the sea. The pumps installed in these systems are single stage centrifugal pumps with an electric motor. The impeller of the pump is properly constructed to manage the suspended solids entrained in the effluent and it disintegrates them into small pieces for easier processing [e.g. filtering, impregnating (commonly, soaking), etc.].

8.5.6 Air conditioning chilled water pumps.

In some vessels the circulation of fresh water is used as a secondary refrigerant means for air conditioning installation. In these installations a centrifugal air conditioning cooling conservation pump circulates the fresh water in heat exchangers where that water is cooled by the primary coolant circuit of the air conditioning installation. Fresh water is then led through insulated pipes to local devices, which provide hot or cold air, or to particular devices that are placed inside the airways that provide air conditioning to the vessel sites.

To maintain a minimum pressure at the pump suction, a tank is installed that contains water and, on its surface, controlled pressure is applied by compressed air (pressure vessel).

8.6 Cargo pumps and related systems.

Cargo pumps are used for the transfer of liquid cargoes carried by tankers (M/T). M/T types include ultra large crude carriers (***ULCCs)*** and very large crude oil carriers (***VLCCs***), distillate petroleum product tankers, chemical cargo tankers and tankers for liquefied petroleum gas (LPG). On these ships, apart from the pumping system, other networks with installed pumps are used, associated with the safe handling of the cargo or the cleaning of tanks. For this reason, the cargo pumps and related systems pumps are discussed in the following paragraphs.

During unloading, as the height of the free surface of the liquid in the cargo tank is reduced and because a lot of cargoes due to their volatility have a high vapor tendency, the cargo pumps have to operate at relatively low values of the net positive suction head available (NPSHa). Also, during the emptying of tanks, vortices may often be formed on the liquid surface, through which the cargo gases or inert gases are transported into the pump suction.

To reduce the cavitation due to low NPSHa in the pump impeller suction, as well as the reduction of vortex formation that transfers gases in the suction, the discharge rate of the tanks is controlled either by pump speed setting or through valves of piping system at the latter stages of the unloading process. These valves may be controlled manually and/or by an automated arrangement located in the ship's cargo control room.

The material used for the manufacture of components of these pumps must be compatible with the cargoes handled, to prevent the creation of sparks that pose fire hazards due to cargo flammability. Also, the possibility of seawater pumping should be considered that is used for the washing of tanks. In this case, the pump must be constructed of corrosion-resistant materials.

The following paragraphs describe the main types of pumps used in M/T.

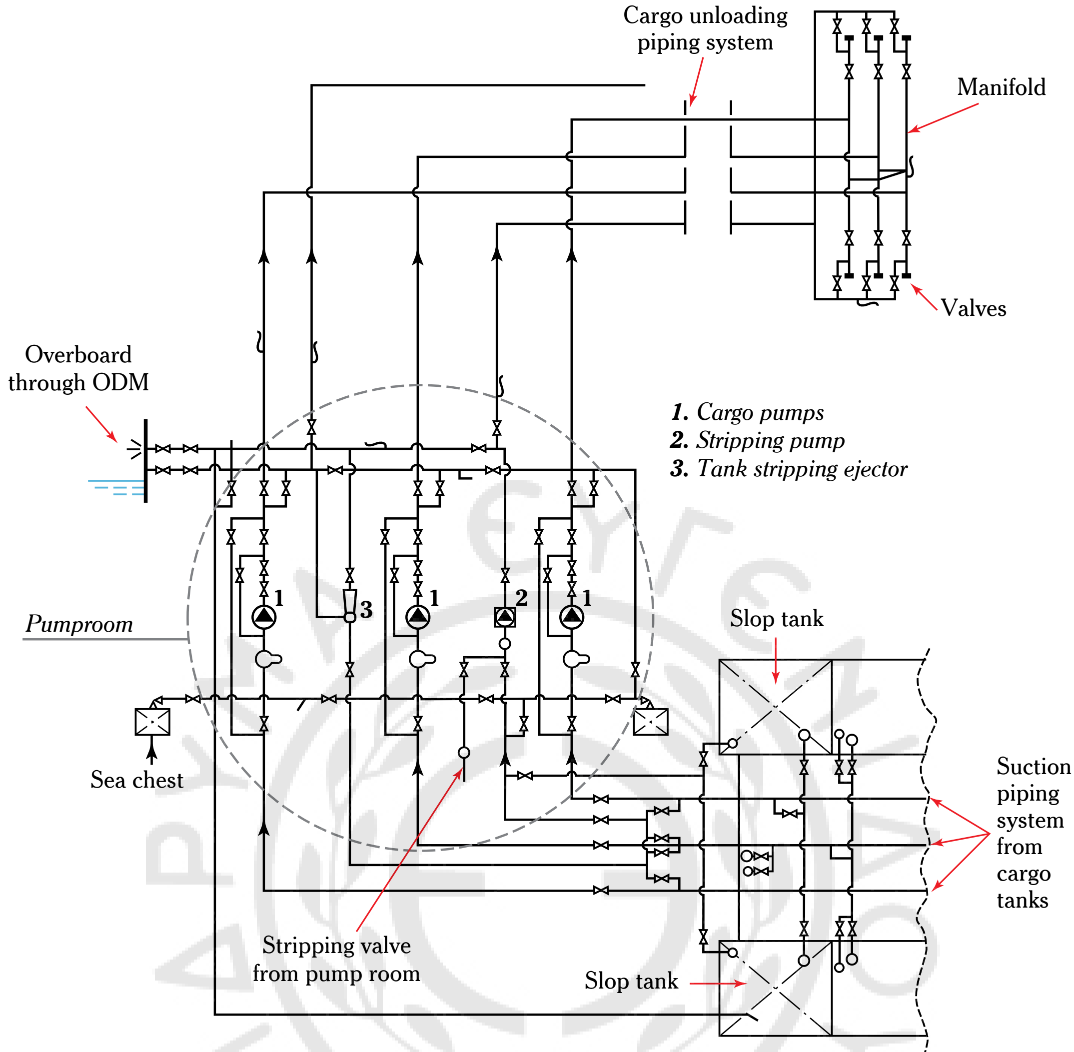


Fig. 8.60
Typical cargo pump and cargo piping arrangement on a M/T.

8.6.1 Centrifugal cargo pumps.

Centrifugal cargo pumps are single-stage and are frequent encountered in ships for transporting a limited number of liquid cargoes including crude oil and a few distillate products such as diesel, kerosene, etc.

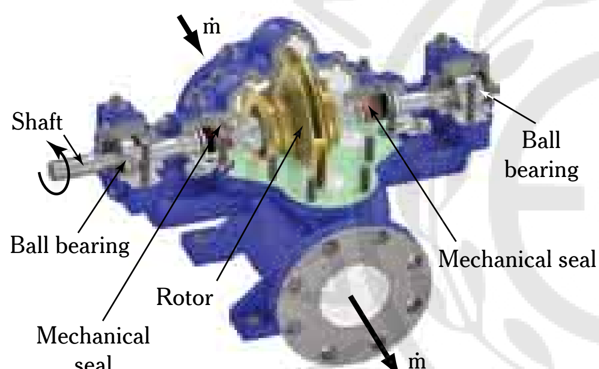
Three or four pumps are used, which are installed as low as possible into the vessel's pump room (fig. 8.60). Through piping interfaces, each pump is able to suck from various tanks, and discharges the cargo onshore through a pipe that corresponds to each pump and is connected to the manifold connection on deck.

The pumps are installed vertically (fig. 8.61) or horizontally, and the type of impeller used (to

Fig. 8.61
Centrifugal cargo pump in pump room.

achieve high flow rate) is double suction impeller [fig. 8.62(a)]. To optimize the damping of radial thrusts, because of the pump shaft length, there might be two support points with ball bearings located on both outputs on the shell. The support with ball bearings on both outputs of the axis on the shell is performed in vertical as well as in horizontal pumps, which are identified as "between bearings pumps" (fig. 8.62).

The pump bearings shell is frequently equipped with a sensor, allowing the operator to perceive the temperature increase. The increase of the temperature indicates abnormality during pump operation and enables the operator to take appropriate action before serious damage occurs. Possible causes of temperature increase of the bearings are inadequate lubrication, high rotational speed, wear of bearings, excessive increase of load, high temperature of transferred cargo, the misalignment or the improper



(a)

(b)

Fig. 8.62
Between bearings pump impeller, in (a) horizontal and (b) vertical arrangement.

assembly of the pump after maintenance-repair, exceeding the wear limits of the rings that are placed on rotating portions between the impeller and the casing of the pump.

The sealing of the shaft, between bearings in pumps where two outlets on the pump casing exist or in pumps with one shaft outlet from the casing, is achieved by mechanical seal.

In order to improve the suction conditions (i.e., the net positive suction height), apart from installing the pumps at the lowest point in the pump room, one or more vent systems are used. These are installed on the side of suction for removing air and cargo gases during pump operation.

These pumps are powered either by steam turbines or by electric motors. In both cases, the drive machine is installed in the engine room and the transmission of movement is accomplished by a connection axis (***jackshaft***), which passes through the bulkhead separating the engine from the pump room (fig. 8.63). The installation of the drive machine out of the pump room achieves the insulation of the explosive cargo vapors that are inevitably generated in the pump room and pose risk of explosion and fire. The opening in the pump-room bulkhead (for a horizontal pump) or overhead [for a vertical pump, (fig. 8.64)] through which the jackshaft passes is ordinarily sealed with a gas-tight stuffing box to prevent explosive vapors, originating in the pump room, from entering the machinery space. The weight of the jackshaft as well as the thrusts from the rotation are received by a thrust bearing that is established in the shaft crossing point at the bulkhead (fig. 8.64).

During pump operation, as tanks are emptied, the net suction discharge head is respectively decreased, which is improved by the provision of inert gas in the tank, by the appropriate system. Inevitably, though, the level of the cargo inside the tank approaches the inlet to the suction tailpipe, so the centrifugal pump loses suction before the cargo tank is emptied. Therefore, the remaining cargo in the tank is pumped by a positive displacement piston pump, the stripping pump, which typically delivers much lower capacities than large centrifugal cargo pumps.

8.6.2 Vertical deep-well cargo pumps.

Deep-well cargo pumps are used to pump cargo from tanks in multi-product and chemical carriers carrying different quality and quantity of petroleum products. Therefore, a pump of this type should usually be suitable for pumping a wide range of liquids having different specific gravities, vapor pressures, viscosities and temperatures. Some cargoes, such as lubricating oils, and other viscous cargoes, may be heated to preserve their fluidity and improve pumpability. In addition, deep-well pumps are sometimes used to pump cryogenic cargoes, ***in LPG carriers***, such as liquefied petroleum gas.

A deep-well cargo pump can normally be driven by a vertical electric or hydraulic motor mounted on top of the pump discharge head above the out-

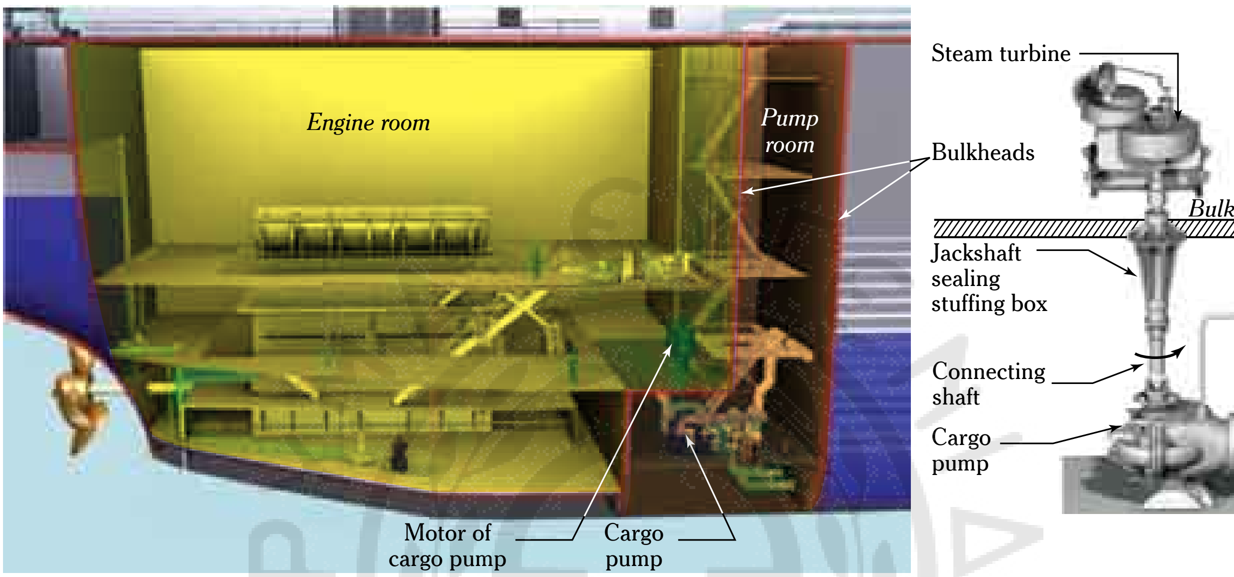


Fig. 8.63
Typical arrangement of pump room in a M/T.

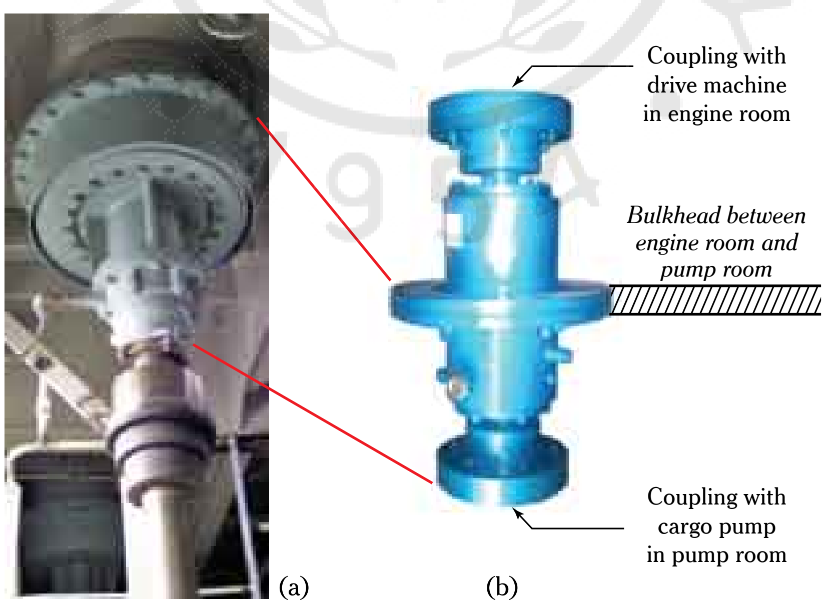


Fig. 8.64
(a) Arrangement of stuffing box of the jackshaft on bulkhead to the side of the pump room, (b) connection of the pump shaft through the bulkhead.

flow of the discharge tube to the deck. The motor drives the pump installed near the bottom of the tank through a long length axis (fig. 8.65). In some cases where the pumps are driven by hydraulic motors, the hydraulic oil is supplied by a central hydraulic power unit. The pump hydraulic motor is installed on the bottom of a well and, the shaft passes through a bulkhead, and is connected to the pump (fig. 8.66). Alternatively, deep-well pumps may be driven through angular motion gear transmission by horizontal drive motor, steam turbine or diesel engine. The driver gear motion is converted to rotational movement of the vertical shaft.

While the discharge of these pumps is typically located on the deck, in some cases they are located lower in the pumping room. One such case is the deep-well cargo pump with horizontal driver which is located in an adjacent space and is coupled to the pump right-angle gear with a jackshaft that passes through a bulkhead separating the pump discharge head from the driver. In this case, the opening for the jackshaft on the bulkhead between the two spaces is ordinarily sealed with a gas-tight stuffing box so the driver can be isolated from any explosive vapor that may be emitted from the pump to the space where the motor is.

The deep-well pumps are centrifugal and the casing is usually constructed as a multi-stage assembly, wherein at each stage there is a single-suction impeller (fig. 8.67). Over the shell, at its connection point with the motion transmission axis there is a short addendum (***spool***), which facilitates the works during inspection or repair of the pump.

In pumping of liquids having high vaporization tendency or low viscosity, an induction rotor is installed prior of the first suction stage impeller on the end of the rotation shaft (***inducer***, which has a specific helix forming shape) (fig. 8.68). As the

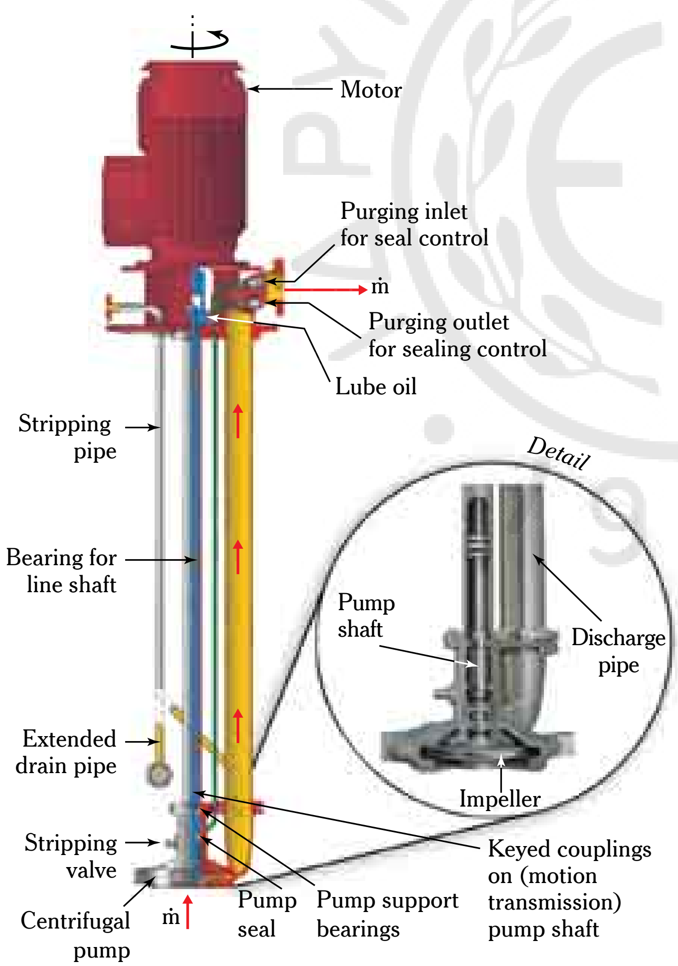


Fig. 8.65
Electric-motor driven centrifugal pump of M/T.

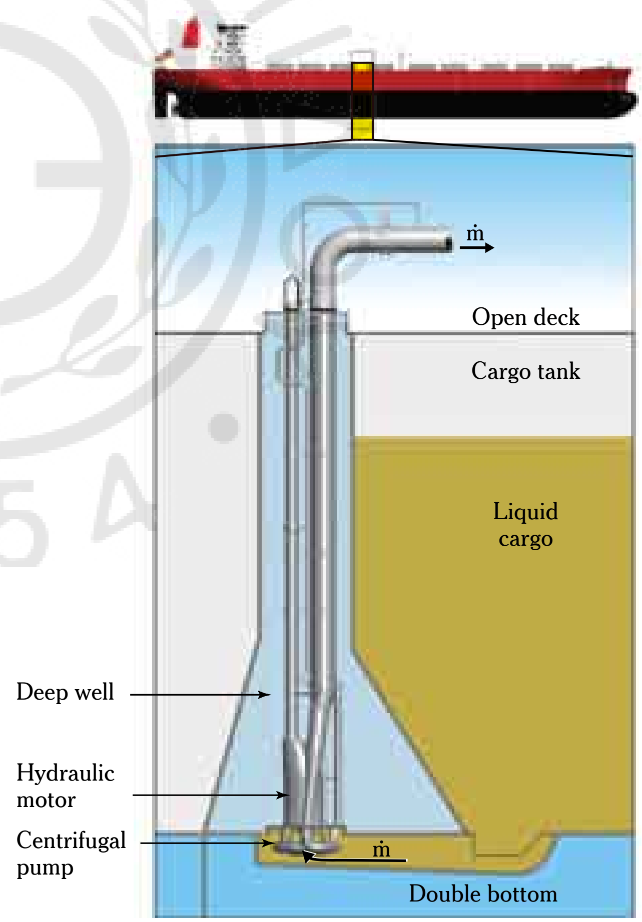


Fig. 8.66
Deep-well cargo pump in vertical arrangement, with hydraulic motor and discharge pipe outflow on deck.

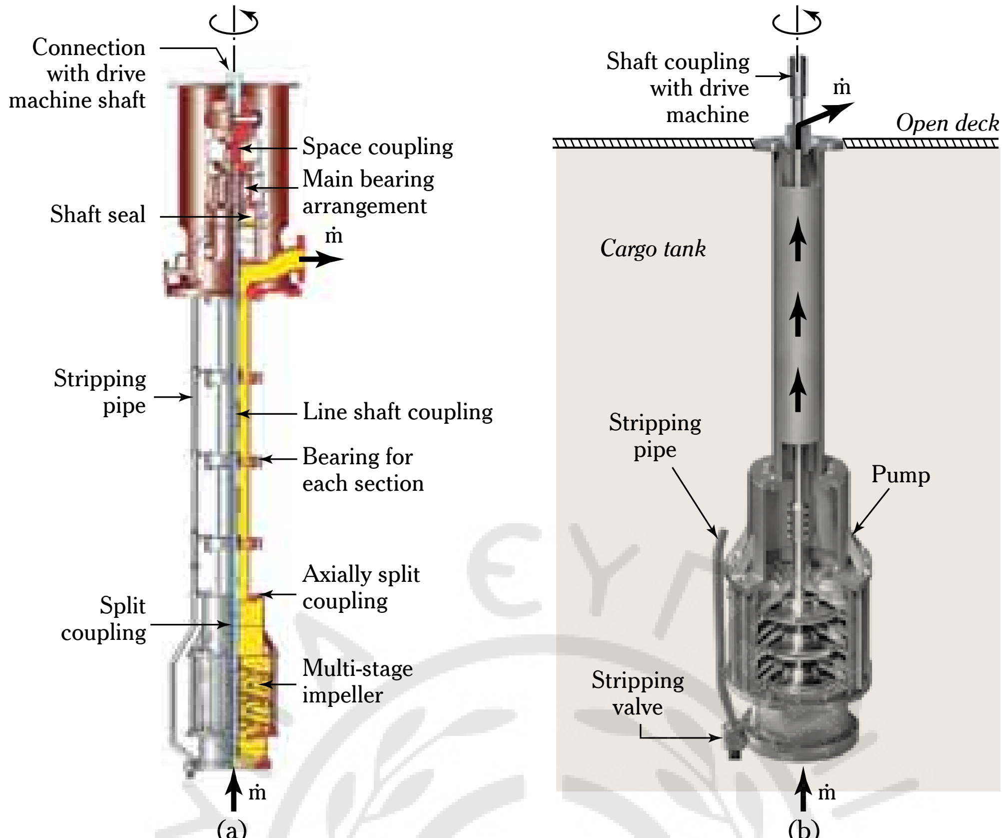


Fig. 8.67
(a) Multi-stage deep-well cargo pump. (b) Image of multi-stage deep-well pump with pipe and stripping valve.

inducer rotates, it provides liquid to the inlet of the first stage impeller, lowering the required net positive suction head of the pump suction. Thereby, the suction and the performance of the pump are improved. Alternatively, when an inducer is not used, the first stage impeller is designed specifically to operate under conditions of low net positive suction head, or the first impeller is double-suction type.

When single suction impellers are used during operation, this leads to increase of axial thrusts, because it creates hydraulic "imbalance" in the pump. It can be prevented by a suitable balancing arrangement such as the balancing drum[8] or the wear rings that are placed on the shell on both sides of the impeller or by a thrust bearing acting on the pump shaft and absorbing the development of thrusts. Correspondingly, the thrust bearings in horizontal arrangement motors, where the motion transmission is performed by angular gear arrange-

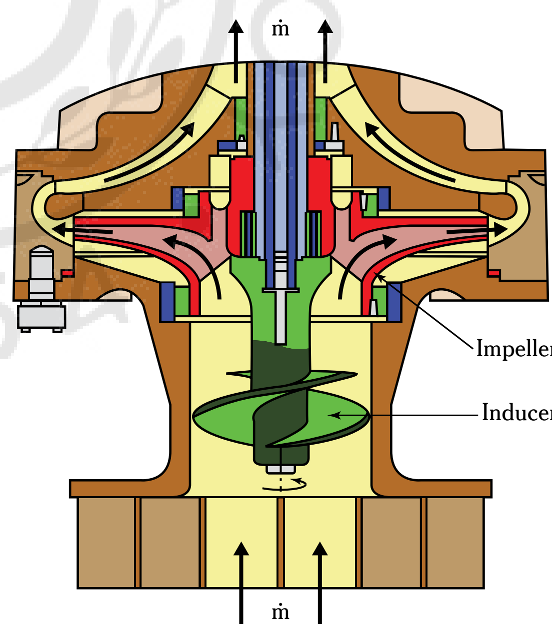


Fig. 8.68
Pump with induction rotor (inducer).

[8] The ***balancing drum*** is an element used in pumps with impellers in series, in order to cope with the forces generated during pump operation.

ment, are placed near the outlet of the discharge tube of the pump on top of the motion transmission axis. In addition, a nut is installed at the supporting point of the shaft to adjust its position, and thus the relative position of impellers to the pump housing.

It is important, in any type of pump support, to maintain the relative gaps, paying particular attention during pump assembly or during shaft position adjustment to prevent any contact of the rotating parts when the pump is activated.

During deep-well pump operation, the cargo is transferred to the shore facilities from the bottom of the tank, where the suction is located, and discharged through the riser pipe located overhead of the pump to the manifolds on deck.

Because of the long distance that exists between the drive machine and the pump, located at the bottom of the tank, the motion transmission shaft is comprised by sections associated with thread or with keyed couplings. Also, because of the long length of the shaft, radial bearings that support the line shaft are frequently mounted on brackets, sometimes referred to as spiders, that are sandwiched between mating sections of the column pipe (fig. 8.67).

Impeller and line-shaft bearings are often lubricated by the pumped cargo. Consequently, bearing materials must typically be compatible with all the fluids that are transported by the ship. The bearings should also be able to tolerate operation with loss of suction if the deep-well pump is used for stripping or during tank cleaning. So, despite the fact that bronze bearings are commonly used, bearings constructed from carbon, polytetrafluoroethylene (PTFE) compounds, and various composites and plastics can also be used.

The shaft seal at the point passing through the bulkhead glands is achieved by gland packings or mechanical seal when the load has high volatility.

When the vessel has a double hull, a ***small well*** is created around the deep-well pump suction opening, which facilitates suction (and pumping of remaining cargo) since it is located below the bottom of the tank and the liquid is collected in it. This well helps particularly during the cargo stripping process of the tank.

The non-return valve which is mounted on the pump suction is another device that helps the extraction of the cargo and stripping, especially due to the long length of the discharge pipe. During pumping, as the fluid flows in one direction, the valve is open and it closes automatically when the pumping is stopped or the discharging of the tank is completed. However, after the discharge cycle has been completed, a quantity of cargo remains within the vertical discharge pipe that can not reach the deck. So, a smaller pipe is used (stripping or bypass pipe) (fig. 8.67), which is connected to the vertical discharge pipe and enables the discharge of remaining cargo. The execution of this process is achieved after the deep-well pump driver has stopped and the pump above-deck discharge valve is closed. Then, compressed air or inert gas is injected into the pump through a connection in the discharge head. The gas forces the cargo contained within the deep-well pump and pipe out through the bypass line connected to the lower portion of the bowl assembly and into the vessel's piping.

Moreover, initial suction valves are installed in the discharge, in order to ensure smooth pump operation and secure suction. Through these valves, in case of suction interruption, the return of liquid is allowed from the discharge pipe to the pump inlet, in order to circulate the fluid and regain the required conditions of successful suction. The initial suction valves are particularly useful for highly volatile liquids.

8.6.3 Submersible cargo pumps moving by hydraulic motor.

Hydraulic motor driven submersible cargo pumps are used for pumping the cargo in chemical M/T, as well as in product carriers, whereas they can also be installed in crude oil tankers. In each tank usually one or two pumps are installed. Thereby the suction between the tanks is avoided and the probability of cargo mixing is minimized, because each pump is usually connected to an independent discharge network set up on deck.

The increasing of the pressure of the hydraulic oil which is used to operate the hydraulic motor of the pump is performed by a central hydraulic power unit, installed in the engine room. The ***hydraulic power unit*** serves the cargo pumps, but it may also provide hydraulic oil to other auxiliary equipment, such as cranes, mooring machinery or the hydraulic motor of the bow thruster (figs 8.69 and 8.70).

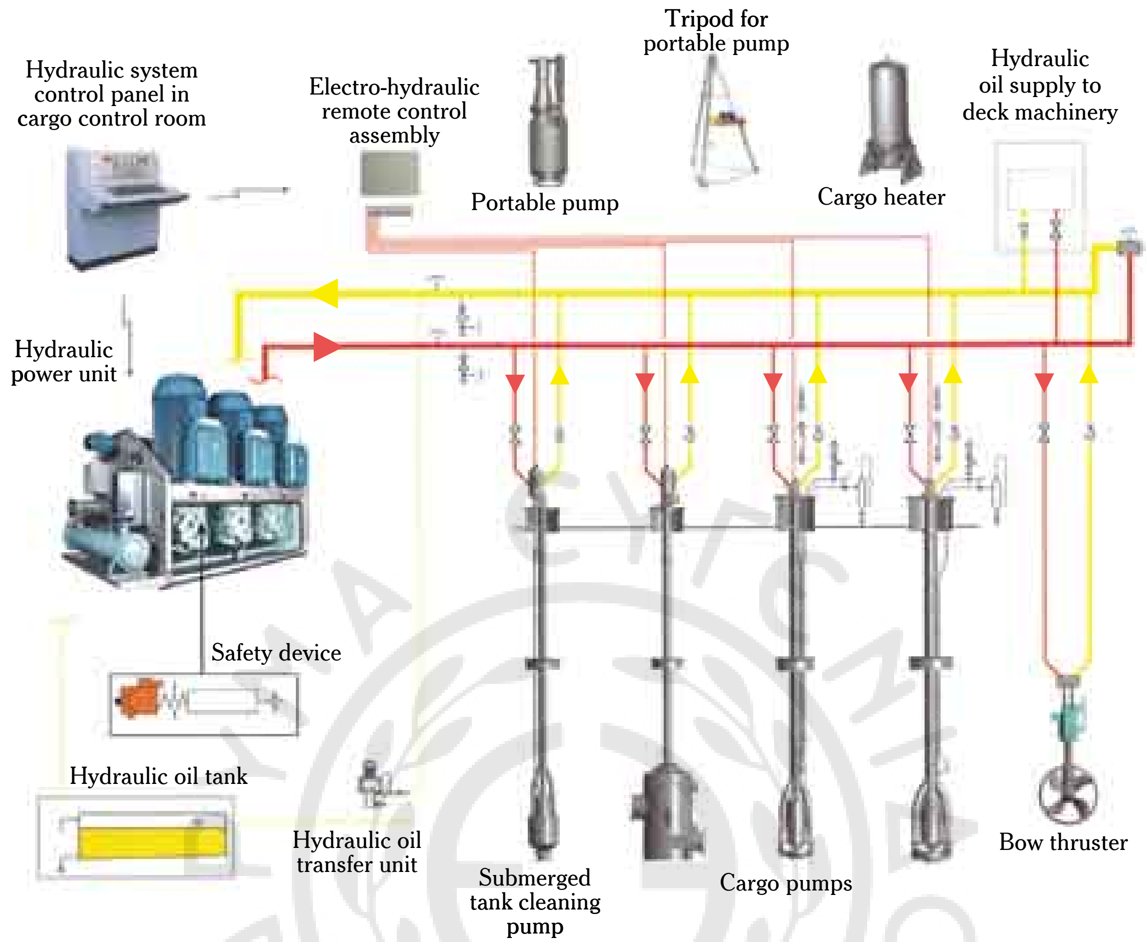


Fig. 8.69
Typical hydraulic oil supply system configuration from a central power unit.

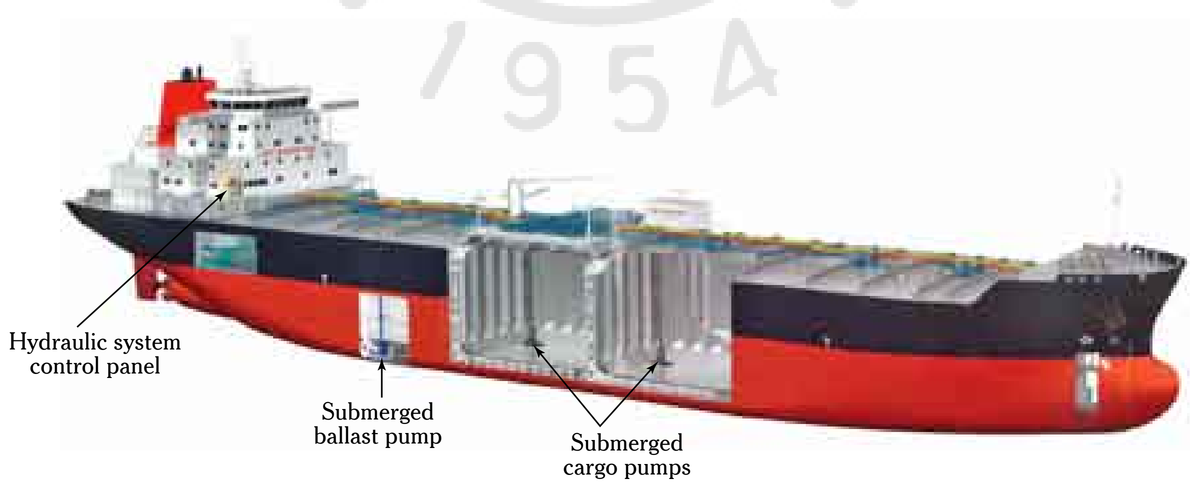


Fig. 8.70
Typical configuration of hydraulic oil supply system on deck, the central power unit and the pumps.

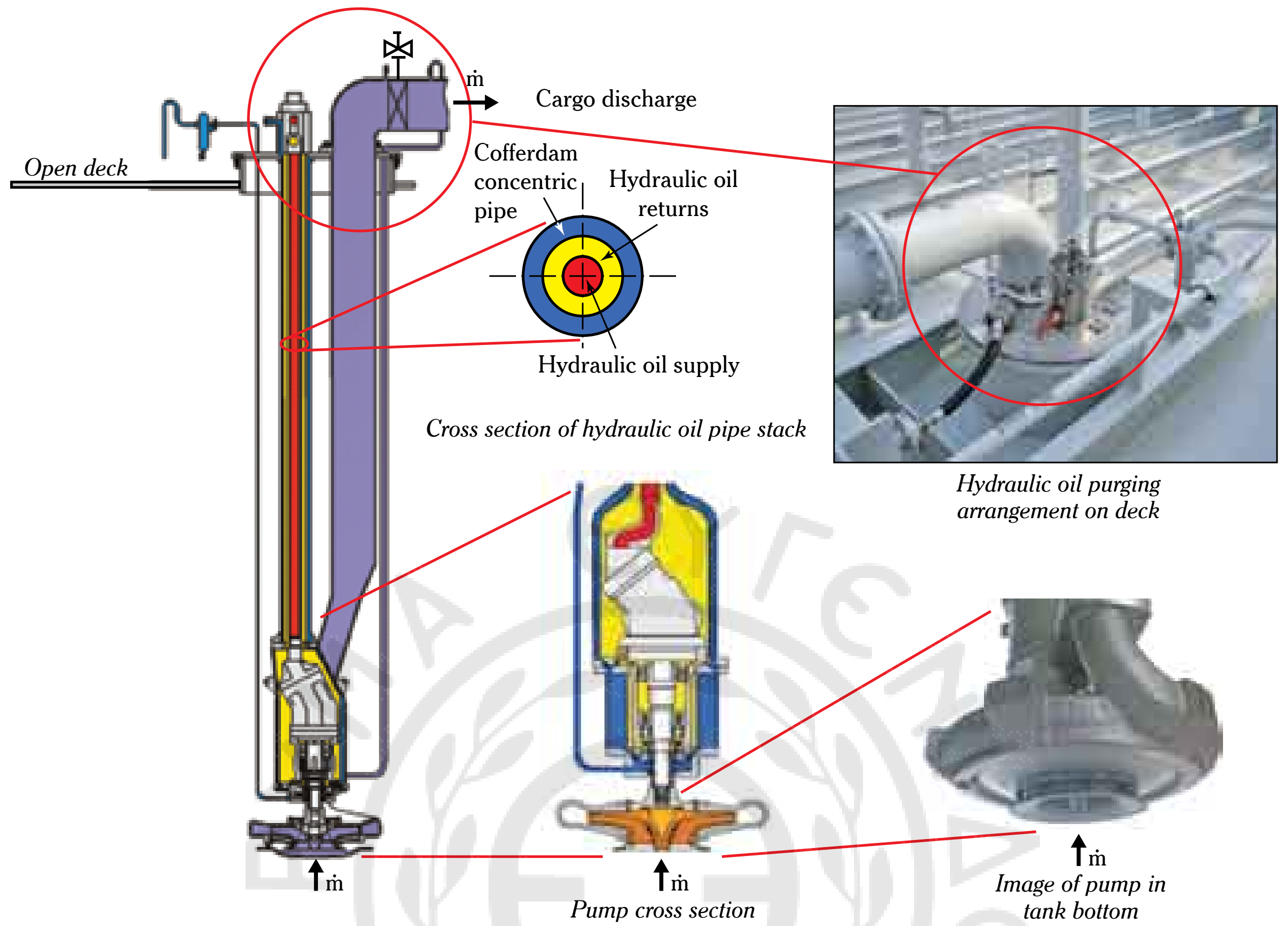


Fig. 8.71
Typical pump configuration, the hydraulic oil supply pipe and the hydraulic oil control system on deck.

The submersible hydraulic motor pumps are single stage vertical arrangement pumps, and consist of a centrifugal single suction impeller (fig. 8.71). The hydraulic motor pump is mounted on the lower end of a vertical pipe, and its upper end is connected to a top plate slightly raised from the vessel's main deck. The vertical pipe has a triple wall so as to form three concentric pipes.

In the central pipe passes the hydraulic oil that is pressed by the power unit, while through the second wall passes the returning hydraulic oil towards the power unit. Inside the third wall passes air or inert gas, creating a leakproof space between the hydraulic oil and the cargo, in case of a leak or a pipe failure.

The rotor shaft is short and is connected to the hydraulic motor by a coupling that may have wedge-shaped configuration or an appropriately shaped slot in which a pump coupling key is placed. The shaft support is achieved with ball bearings, which are lubricated by the hydraulic oil returning from the hydraulic motor to the power unit. The seal of the circulation pipe of the hydraulic oil with the pump shaft and the outer seal pipe is performed by particular ***lip type seals*** or with mechanical seals. Thus mixing of the cargo with the hydraulic oil is prevented.

The control for any leakage of hydraulic oil is achieved by supplying air or inert gas under appropriate pressure inside the sealing tube. The discharge is performed through a small diameter bypass line, ending in a collector of leakage on the deck near the extraction point of the discharge pipe [fig. 8.72(b)]. The control is carried out by measuring the amount of leakages that is collected in a given operating time period.

Above the top plate of the vertical pipe for the supply of operational oil to the pump, the hydraulic oil supply valve is installed, by which the speed of rotation of the pump is controlled.

The liquid from the volute casing of the pump is discharged through a pipe that is established in parallel to the hydraulic oil pipe. The discharge pipe reaches up to the deck, where the discharge valve is installed, which is used to check the flow towards the ship's discharging piping system.

The drainage of the pump and the vertical discharge pipe after completion of tank discharging is realized by compressed air or inert gas. The injection of air or inert gas is achieved through a small valve appropriately connected on the discharge pipe on deck. So, the stripping of the discharge pipe is performed through the bypass pipe whose one end is connected to the bottom of the discharge pipe, and the other after the discharge valve [fig. 8.72(a)]. Because there is no non-return valve installed at the suction of the submersible pumps, the discharge pipe stripping process must be done before the pump shutdown.

The pump's installation onto the piping system is performed so that it can easily be removed in case of damage or repair without requiring disassembly of the hydraulic oil pipe and the discharge pipe. Moreover, one or two portable submersible pumps are available on vessels and are inserted from a suitable opening on the deck into the tank (using a tripod or winch) for pumping the cargo when the main pump of the tank is inoperative due to damage. The hydraulic oil pipes for the operation of the portable pump as well as flexible discharge pipes for connection to the discharging piping system are available on the ship.

The lack of a non-return valve on the pump suction allows the discharge system to be used for the

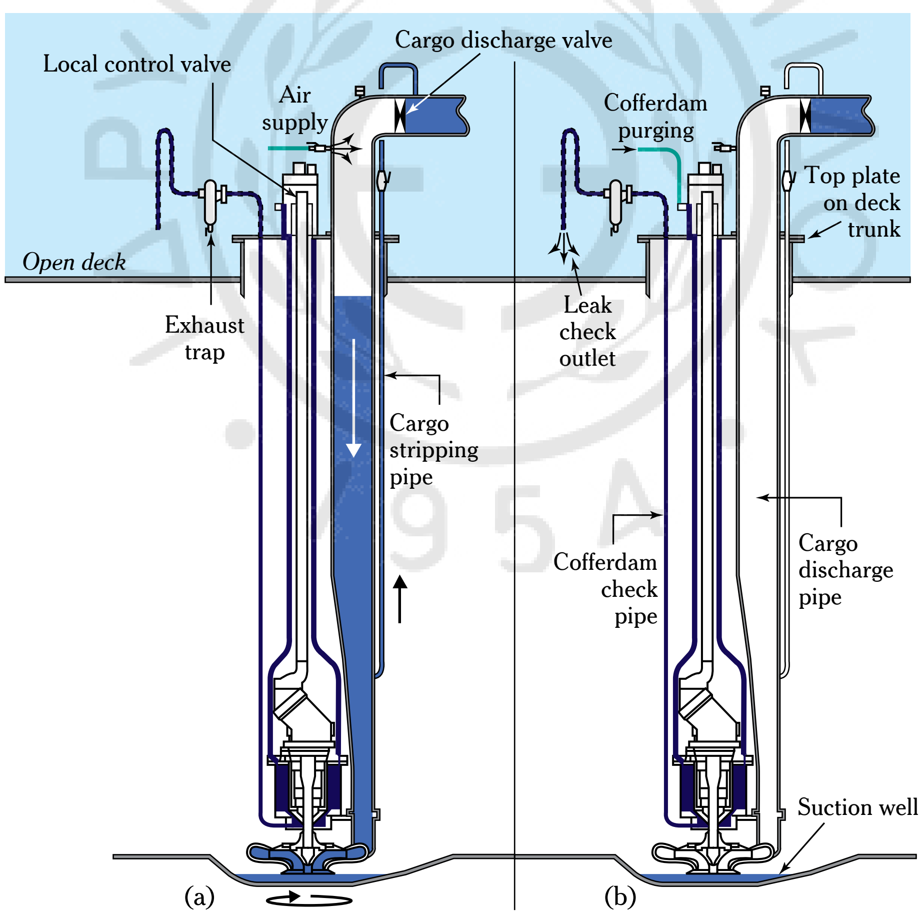


Fig. 8.72
Stripping and check of sealing of the pump pipes.
(a) discharge pipe stripping procedure, (b) leak test procedure.

loading process of the tanks. In this case, an arrangement in the pump prevents the rotor to rotate backwards. The loading speed is reduced due to resistances in flow by the impeller.

When submersible pumps are used for ballast seawater management, one or two pumps in appropriate arrangement are usually installed to tanks towards the stern of the ship. From the portion where the pump is installed, the ballast pipes system extends to the rest of the tanks (fig. 8.73).

The ballast management pump operation is similar to the cargo pumps, except that the rotor of these pumps is designed for seawater management and it must always be immersed in water to facilitate initial suction.

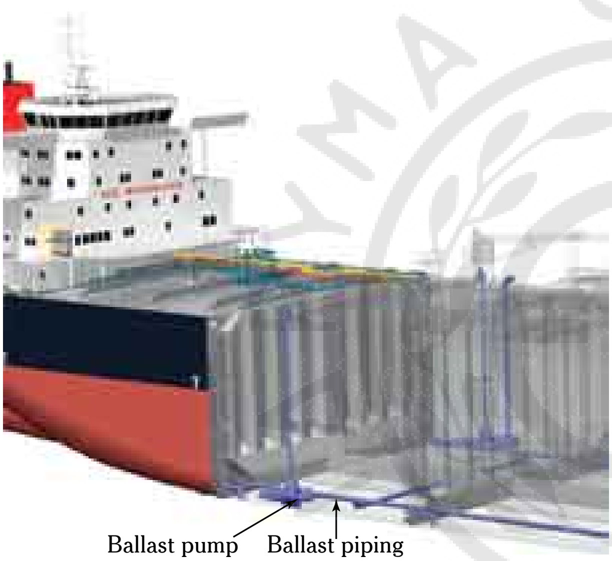


Fig. 8.73
Ballast piping system configuration with submersible pumps.

8.6.4 Electric-motor-driven submersible cargo pumps.

Electric-motor-driven submersible cargo pumps are used to discharge cargo from liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers (fig. 8.74). Their application for such cargoes is related to the very low cargo temperatures (cryogenic cargoes) (approximately −162°C for LNG and below −50°C for many LPG cargoes). In those temperatures the hydraulic oil that is used to operate the hydraulic-motor-driven submersible cargo pump is unsuitable.

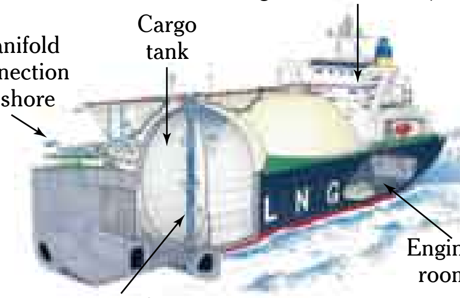


Fig. 8.74
Configuration of cargo pump installed in a tank of LNG carrier.

Each motor-driven submersible unit typically consists of a vertical centrifugal pump with single or double stage. One or two impellers are mounted on the lower end of an electric motor shaft. Furthermore, an ***inducer*** (auxiliary propeller-type impeller) is often installed at the inlet to the first stage impeller (the only impeller in a single-stage pump) to reduce the pump net positive suction head requirements.

The submersible pump with electric motor is an integrated unit (fig. 8.75) and a pump is placed in each tank. The liquid cargo enters at the bottom of the pump from the suction opening of the rotors and as it is discharged it enters an annular passage formed between the motor frame and an outer casing that surrounds the motor.

A portion of pumped liquid usually passes through openings in the motor frame and flows through the motor. In addition to cooling the motor, this bypass flow lubricates the ball bearings that support the common pump and motor shaft, where the pump impeller is installed. After leaving the top of the annular passage, the cargo that has been discharged by the pump passes through a connection located above the motor and enters a vertical pipe that leads to the discharging piping system on the main deck.

The motors of the pumps are specially designed to operate without risk of causing an explosion (***explosion-protected, Ex-e***) following the strict regulations set by the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC Code).

In addition to the main cargo discharging pumps, smaller spray or cool-down pumps are often installed on an LNG carrier. The pumps installation serves the preparation of cargo tanks during the voyage

Fig. 8.75
Electric-motor-driven submersible cargo pump.

back to the terminal for loading, since they circulate LNG through a cool-down header and spray nozzles in each tank.

8.6.5 Rotary cargo pumps.

Rotary cargo pumps are used as main unloading pumps on vessels that carry high-viscosity cargoes. They are multiple-screw or lobe-type rotary pumps (see par. 2.7). Furthermore, in vessels that have centrifugal type main cargo pumps, it is sometimes possible to also have lower-capacity screw, lobe, or sliding-vane pumps that are used to strip cargo tanks.

When the rotary pumps are used as the main cargo pumps, they may be located either in the pump room and a suction and discharge network is developed in the ship through which the cargo passes, or as deep-well pumps typically by installing the pumps in the bottom of each tank.

The electric motor that is used to operate the pumps in the pump room is installed in the engine room and the drive shaft of the pump passes through the bulkhead between the pump room and the engine room. In this way the driver can be isolated from explosive vapor in the pump room, because it is installed in an adjacent space and is coupled to the pump with a jackshaft that passes through a bulkhead stuffing box.

In deep-well pumps, where a pump is installed in each tank, that pump is driven by an electric motor installed on the deck, and the transmission of motion is carried out by a line shaft that reaches the bottom of the tank where the pump is installed. The discharge of the pump is performed by a vertical pipe above the pump that reaches up to the deck. In many pumps the drive line shaft is enclosed within a vertical column pipe. Hence the cargo that is discharged by the deep-well rotary pump may pass through the column, or it may pass through a separate vertical pipe mounted adjacent to the column. Bearings that support the line shaft, together with the pump bearings and timing gears, when used, are sometimes lubricated by the pumped fluid. The pump flow may be adjusted either by changing the motor rotation speed, or by a hydraulic device that alters the supply of discharge of the pump (i.e., limits the supply by using a valve which creates a pressure drop).

8.6.6 Reciprocating cargo pumps.

These pumps are direct-acting ***reciprocating cargo pumps*** and used when steam is available on board. These units typically have double-acting pistons in both the liquid as well as the drive steam-cylinders and function similar to the piston pumps described in paragraph 2.6.

Direct-acting reciprocating pumps are used as stripping pumps of tanks on vessels that have centrifugal turbine-driven main cargo pumps. Reciprocating stripping pumps have less capacity than the centrifugal cargo pumps and are mounted in the pump room. Also, they are connected by appropriate bypasses through the cargo suction and discharge pumping system of the main centrifugal pumps to the vessel's cargo tanks.

8.6.7 Inert gas system pumps[9].

To eliminate the risk of explosion and fire that could be caused by flammable gases generated by the properties of oil liquid cargoes in cargo tanks of tanker vessels, cargo tanks must be kept filled with inert gas through the inert gas production system. For the proper operation of such systems, seawater supply pumps are used.

In vessels where flue gas from a fossil-fueled steam boiler is used, a pump is usually required to deliver seawater to a ***scrubber*** (tower for cooling and cleaning of exhaust). In the scrubber the water is used to cool, clean, and desulfurize the inert gas. The scrubber pumps used are frequently single-stage electric-motor-driven, single suction impeller centrifugal pumps in vertical arrangement. These are installed at the lower level in the engine room and discharge seawater through appropriate nozzles to the scrubber.

Additionally, on some vessels flue gas is used from a special oil-fired inert-gas generator where the inert gas for cargo tanks is produced. There, the seawater is provided by similar scrubber pumps.

A separate centrifugal pump is also frequently required at inert gas systems to supply seawater to a wet-type deck seal that is used to prevent vapor in the vessel's cargo tanks from flowing backwards through the inert-gas supply piping and into the machinery spaces. It is installed in the engine room and is named ***deck seal seawater pump***, because it is used to send the seawater in the wet-type deck device.

8.6.8 Tank cleaning pumps.

On some vessels, either hot seawater is used to clean cargo tanks of residue that remains in the tanks after the liquid cargo has been discharged (tank cleaning), or the cargo itself is used during discharging to clean cargo tanks (cow-crude oil washing).

A tank-cleaning or tank-washing pump takes suction from a sea chest and discharges seawater through a header in tank washing machines. This pump may be a separate pump or the fire and general service pump. Hence single and two-stage centrifugal pumps that are driven by steam turbines, electric motors, or hydraulic motors may be used in this application.

In the crude oil washing process, the cargo pump that is used during cargo discharging takes suction by the same cargo tank and discharges in tank washing machines. So, for each vessel, the pump characteristics for the crude oil washing are those that apply for the cargo pumps.

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[9] For inert gas system see: Dagkinis, I. and Glykas, A. (2015). *Βοηθητικά Μηχανήματα Πλοίων*, ("Ship Auxiliary Machinery"*)*. Athens, Eugenides Foundation, Ch. 9.

CHAPTER NINE
Sealing, bearings, starting and operation of pumps

9.1 Sealing of pumps.

Pumps consist of various components which are connected to each other for the creation of a single machine. Therefore, the presence of an element is obligatory between connected metal surfaces, in order to cover the small imperfections and to prevent leakages of liquid to the environment. The prevention of ***internal leakages*** and the ***isolation of inner spaces*** can be achieved:

1) By piston rings, in piston pumps,

2) by the liquid itself that is transferred in positive displacement rotary pumps, which creates the sealing between stable and moving parts, while allowing rotation of the rotor, and

3) by friction (wear) rings between rotor and casing in dynamic pumps.

To prevent ***external leakages***, the sealing material depends on factors such as the following: the place where it is placed, the liquid that is transferred (taking into account its corrosive and chemical properties), the pressure that develops within the pump, the liquid temperature, the type of the pump and the frictions, if any. So, suitable sealing means with appropriate designs are used, in order to fit precisely on the surfaces, and succeed in getting the desired result.

Sealing means, according to their application, can be classified into the following two categories:

1) ***Static sealing means***, applied for sealing the cylinder heads with liners, the valves to the cylinder heads, in connections of pumps with split shell, in connections of flanges to the piping system, as well as for attaching of components that are adjacent to the pump, such as the relief valves or the air extraction pumps.

The static sealing means are called ***gaskets***, and they are the "mechanical sealing" which fills the space between two or more surfaces that are connected. Gaskets are intended to prevent leakages from the pressure developed within the pump, or to prevent air inlet to the chamber or chambers where the liquid passes. The gaskets are usually made by cutting special sheets which are commercially available in various dimensions (e.g., $1\,m \times 1\,m$ or $1\,m \times 1.5\,m$) and various thicknesses (e.g., 1 mm, 2 mm etc.). Also, they are readily available for use by pump manufacturers at the required dimensions for each application. The cutting of the gaskets (when done by the ship's crew) shall be carried out with precision and must cover the entire surface of the linking point; otherwise the effectiveness of the seal is reduced.

The materials used in the production of gaskets are paper[1], simple permanite of aramid[2] (without asbestos due to the health risks), and permanite containing pure graphite. All these materials are available in various thicknesses and sizes, but, particularly for the sealing of the pump casing, the thickness should be very small. Otherwise, clearances between moving and stationary parts of the pump are affected, resulting in malfunction, and a high risk of serious harm is caused. Also, sealing rings of durable elastic may be positioned in suitable grooves.

2) ***Moving parts sealing means***, applied for sealing the rods of reciprocating pistons and the shafts of impellers.

By the use of this type of gaskets, leakages are prevented when the level of sucked liquid surface is over the pump and enters the pump by gravity, or, the pump suction is located higher than the level of

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1 ***Gasket paper*** is a paper used for making gaskets of various dimensions and thickness, and composed of cellulose fibers, vegetable fiber, vulcanized fiber or other materials impregnated with glycerol adhesive composition.

2 ***Aramid fibers*** are a class of strong synthetic fibers, resistant to heat and high pressure.

sucked liquid, so that pumping is achieved by developing vacuum. In these cases, the sealing means prevent the entrance of air to the pump or the leakage of water to the environment, which cause difficulties in starting and in operation accordingly.

The sealing means of the moving parts are as follows:

a) ***Gland packings*** are placed in a stuffing box that is created on the pump housing or on the cylinder block, at the point where the impeller shaft or the piston rod passes, respectively. The soft construction material they are made of allows their compression by the gland follower. The conventional gland follower (fig. 9.1) is a ring with a flange and two diametrically opposite holes, where full threaded studs are passing through, on which the nuts are applied. The full threaded studs are mounted on the pump body. So, as they pass through the holes of the gland follower, by tightening or by loosening the nuts, adjustment is possible, as the packings are compressed by the gland follower. The compression of the glands causes their expansion into the stuffing box and minimizes the gap between shaft and packings. Thus, leakages from the pump are prevented, while the passing shaft rotates or reciprocates easily. In order to ensure sealing, with conventional gland packings, the positioning of 4 to 6 packings in the stuffing box is sufficient (fig. 9.1).

Various kinds of packings are commercially available, each of which is manufactured of materials suitable for the fluid that is handled by the pump, the developed pressure and the temperature. Moreover, when they are installed in centrifugal pumps, the rotation speed of the shaft is taken into account.

The right choice of packings to be used ensures not only the sealing, but also the protection of the shafts by corrosion. According to their construction material, they could generally be classified into several types.

Some indicative types of packing are the following:

- ***Flax impregnated with teflon***, that are suitable for use in ship shafts, hydraulic systems and other shipbuilding applications. The operating conditions appropriate for these gaskets are pressures up to 80 bar, temperatures up to 150 °C, pumps with shaft rotational speeds up to ***5 m/s***, and fluids with alkalinity index between ***ph 5-10.***
- ***Cotton fibers impregnated with teflon*** and specific lubricant, which ensure shaft minimum wear. These packings are generally used in nautical applications, and the operating conditions are appropriate for are pressures up to 200 bar, temperatures up to 100 °C, pumps with shaft rotational speeds up to ***10 m/s*** and fluids with ***ph 2-10***.
- ***Lubricants of high temperature***. They are suitable for use in rotary and reciprocating pumps, which generally handle chemical oils,

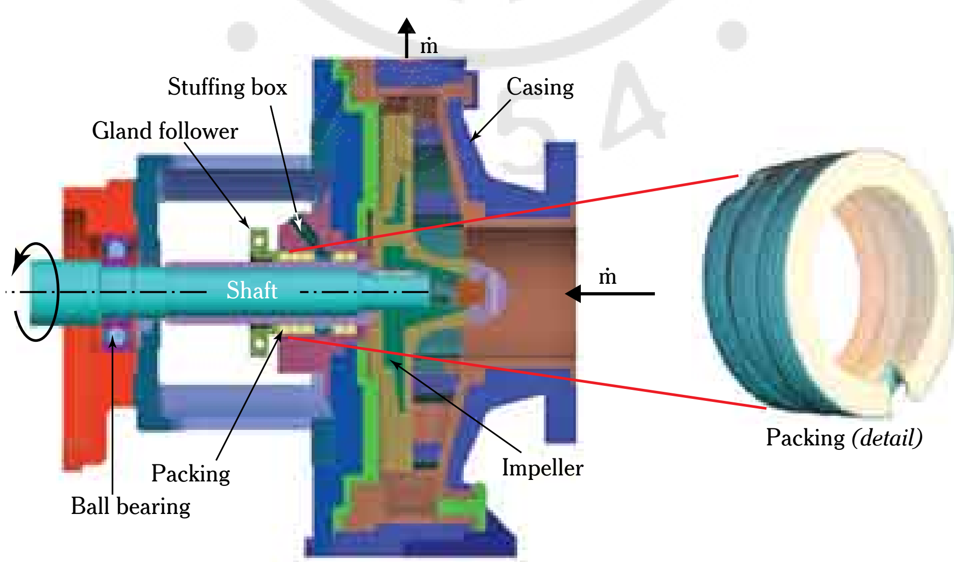

Fig. 9.1
Stuffing box and packings.

solvents, light acids, etc. The operating conditions they meet are pressures up to 140 bar, temperatures up to +260 °C, pumps with shaft rotational speeds up to ***10 m/s*** and fluids with ***ph 3-12***.

- ***Pure graphite fibers*** 98%, by adding Inconel particles. The operating conditions met by these types of gaskets are pressures up to 300 bar, temperatures −190 °C to +650 °C, while in non-oxidizing environment may be −190 °C to ***3000 °C***, and fluids with ***ph 0-14***.
- ***Inorganic materials***, by adding mineral oil and graphite. These packings are suitable for pumps which handle liquids containing water, alkalis, oils and chemicals, and generally respond well to pressures up to 140 bar, temperatures up to +480 °C, shaft speeds up to ***15 m/s*** and fluids with ***ph 3-12***.

The packings that are available on ships are either prefabricated rings, or are available in rolls, from which the required length of packings may be cut. The correct length can be measured by wrapping the packing around the shaft, or by measuring the circumference of the shaft and then the packing can be cut to the relevant length, accordingly.

In order to cover the sides of the packings, when applying them into the staffing box, the cutting at the ends of the packing pieces must be diagonal, so that, as they are wrapped around the shaft, both ends are covered, providing a better seal. The splices during the installation of the packings shall not coincide, that is why they must be placed at a difference of 90° from the point of splice. The tightening, during the installation, is performed by pressing the gland follower at a pressure 1.5 to 2 times the operating pressure.

b) ***Mechanical seals*** are used for sealing the shafts of centrifugal pumps (fig. 9.2).

Due to their yields, they have replaced the gland packings. Also, they are installed at a point with a specific configuration on the shell of the pump, from which the shaft of the rotor passes. They consist of rings made of metal and soft material (e.g. coal and graphite alloy) with a special configuration. One of the metal rings is fixed on the shaft, and maintains its position by bolts, so to rotate with the shaft. Inside this ring, an elastic sealing ring is applied circumferentially, preventing the leakage of fluid.

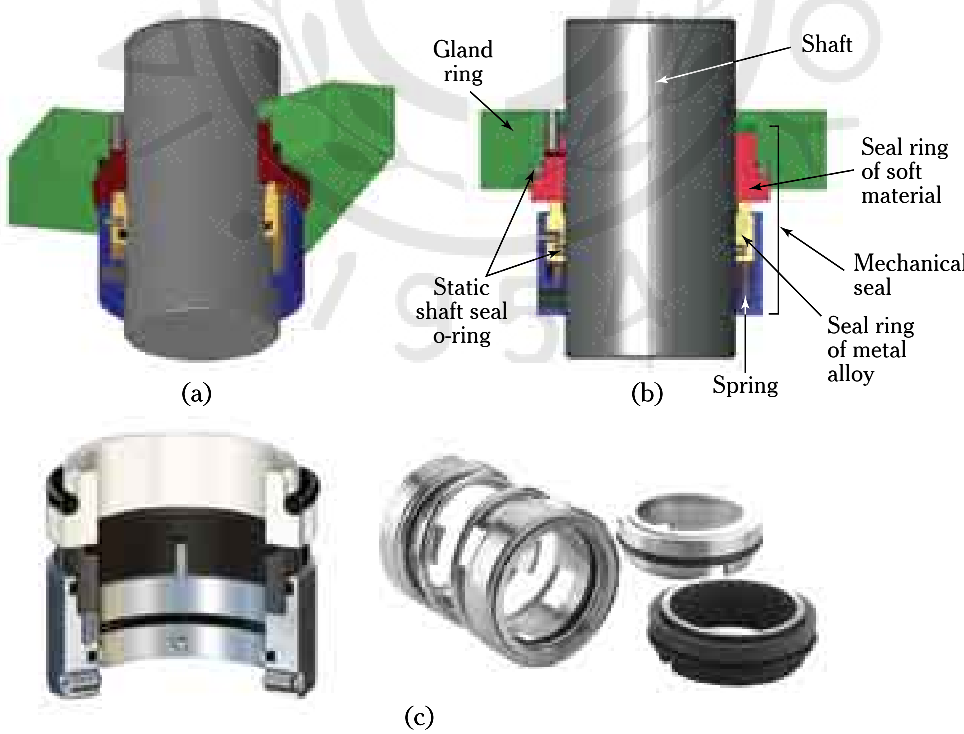

Fig. 9.2
Mechanical seal.
(a) Installation on the shaft. (b) Section of mechanical seal. (c) Image of mechanical seal.

The ring made of soft material is either applied on the pump casing, and fixed on it by a sealing rubber ring, or is fixed with a pin on the moving part and entrained by it. The other metal ring that the mechanical seal consists of is called ***friction ring*** and its type depends on the type of mechanical seal. This ring is positioned opposite to the position of the ring made by soft material, either on the pump casing or on the rotating ring. In this way, the interface of the two rings is achieved, i.e. the soft ring made of carbon-graphite and the friction ring, and, in interface areas, the sealing of the pump is realized, preventing leakage, while at the same time permitting rotation of the shaft.

During operation, as the shaft rotates, constant contact of the two rings, which are used to seal the pump, is achieved by spring tension. The spring is located under one of the intermediate reinforcement rings, which biases the ring made of soft material. When the soft material ring is stably positioned on the shell, the spring biases the friction ring in the same mode.

Mechanical seals can be used for all kinds of liquids that are transfered through pumps, particularly when the pump is operating at high speed, when the liquids are corrosive, as well as in high temperatures, since these seals are resistant up to 260°C.

Mechanical seals, compared to gland packings, have the following ***advantages***:

- They minimize power loss, because friction developed between the sealing surfaces is small.
- They maintain their tightness when there are small displacements and vibrations of the shaft.
- The shaft is not rubbing on the seal, as in gland packings, thereby reducing the repair cost because shaft wear is eliminated.
- After installation, adjustment for sealing is not required.

They are at a ***disadvantage*** in terms of:

- The sensitivity of the rings made of soft material, which are fragile and get easily damaged from incorrect handling and tightening.
- The high cost, especially for those used in pumps that handle oil product liquids and highly corrosive fluids, and
- no adjustment is possible after installation, since the disassembly of the pump is required.

c) ***Shaft seals***[3] are used to seal rotating shafts which transmit the movement to the rotor. They are placed at the point where the shaft enters into the pump housing. They are manufactured in "U" shapes and the construction material may be either:

- elastomer (rubber) internally at the point that comes into contact with the shaft and metal externally at the outer ring, for fastening extremally at the point of application (fig. 9.3), or
- solely elastomeric material (fig. 9.4). In this case, more than one seals are required, while, to maintain their point of application, a support ring is used.

The seals are installed as illustrated in figure 9.5, with the inner side of the «U» shape towards the internal part of the pump. The pressure they can respond to reaches 200 to 300 bar (depending on type) and a temperature of about 105°C.

The contact point of the seal with the shaft forms two angles (or lips). One lip is towards the outer side and is in contact with the air, while the other is in contact with the liquid and faces the pressure developed inside the pump (fig. 9.5).

In the first type of lip seal, the one with the metal

Fig. 9.3
Lip seal of elastomeric material with a metal ring and spring.

Fig. 9.4
Lip seals solely of elastomeric material.

[3] Sealing with shaft seals is also known as lip seal or radial oil seal.

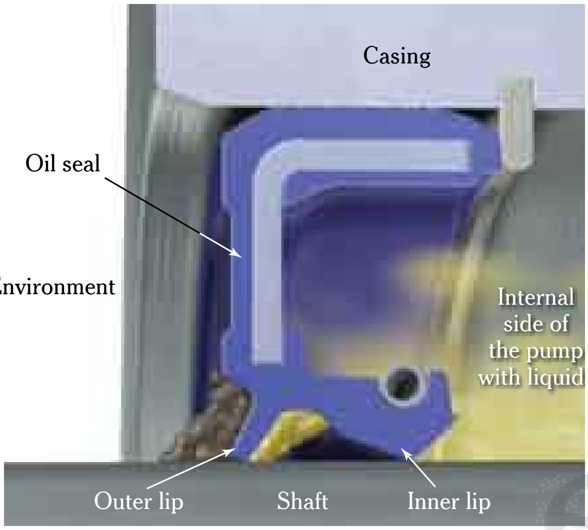

Fig 9.5
Installation of Lip seal.

support (which is actually the most common), an internal spring retains the contact between the lips and the shaft. The pressure of the spring compensates the functional wear, caused by the rubbing of the rubber with the metal shaft. In order to achieve the desired sealing, only one gasket of such type is used.

In the second type, entirely made of elastomer, the sealing is achieved by the pressure of the retaining ring, which is an adjusting means of the seal. The seals are usually positioned two or three in a series, and are held together by a specific connector ***(cartridge design)***.

The seals are commonly found in rotary positive displacement pumps, such as lobe pumps, gear, screw and sliding vanes.

Apart from elastomeric material, they may be constructed from teflon, and in this case they are used at higher pressures and temperatures, while being suitable for pumps that operate at high speeds.

9.1.1 Sealing of piston pumps.

In positive displacement piston pumps, seal (fig. 9.6) is usually required: at the area where the cover is attached on the cylinder wherein the piston reciprocates; at the cover in the portion where the suction and discharge valves are positioned as well as on the portion from which the piston rod passes into the cylinder chamber; at the flanges of the cylinder where it is connected to the suction and discharge piping system.

A suitable sealing means or material is used for sealing each portion of the pump. Thus, at the connecting area between the body of the cylinder and the cover, at the area of the sealing of the valves and at the connecting flanges, vegetable paper or small thickness permanite are usually used. However, in certain types of pumps, a groove is formed on the surface of these attaching areas, wherein an elastic ring (o-ring) is placed.

At the area where the piston rod comes through the body of the cylinder, there is a stuffing box where packings are placed and they are pressed by a gland follower in such a manner as to eliminate the leakages from the interior of pump.

The existence of humidity or drops of liquid on the gland packing is desired, and it is not a malfunction or poor sealing, because the liquid acts as a coolant of the packings. Note that the humidity or the presence of liquid droplets involves only the handling of salt or fresh water by the pump, otherwise any leakage by volatile and corrosive fluids must be prevented.

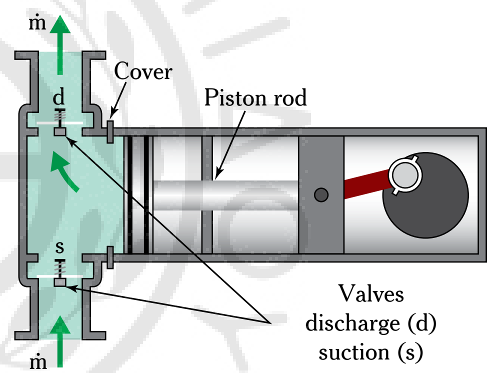

Fig. 9.6
Indicative sealing points on piston pumps.

In most pumps, the adjustment of gland packings can be performed during the operation of the pump. This work should be done with great care, because the possible excessive expansion of the packings [par. 9.1(1)] will cause the interruption of the free stroke of the piston, resulting in fatigue of the drive machine, in heat development in the packings, and in abnormalities in the operation of the pump.

9.1.2 Sealing of rotary pumps.

The sealing of rotary pumps is required: in order to connect the covers to the body or housing in which the rotors rotate; for the suction and discharge valves according to the type of the pump; on the connection of the flanges to the piping system; for the portion wherein the rotation shaft of the rotors enters into the pump chamber (fig. 9.7).

In order to seal the pump body with the covers, the appropriate medium is used depending on the type of the pump. Thus, in some pump types vegetable paper may be used, while in certain types, such as lobe pumps, sliding vane pumps, gear pumps, seal material is not used. In this case the metal surfaces are abutting and only a ***liquid sealant*** is added, which creates a very thin film. In other pumps of these types, grooves are formed on the surfaces of contact points of the covers, where an elastic ring (o-ring) is placed.

The shaft sealing of the rotors is accomplished either by mechanical seals or by lip seals, while in the suction and discharge valves rubber rings are usually used.

In piping connection flanges, we use permanite or rubber gaskets, depending on the fluid handled by the pump.

9.1.3 Sealing of centrifugal pumps.

In centrifugal and generally in dynamic pumps the entrance of air as well as the leakages of liquid to the environment are a cause of yield reduction. Particularly, due to their high rotational speed, the presence of air will increase the risk of cavitation as well as interruption of the flow within the pump. Therefore, proper sealing contributes significantly to the improvement of pump operating conditions.

The seal of centrifugal pumps in the connection points of the dividing axially casings [(fig. 9.8(a)] is performed by vegetable paper or thin permanite.

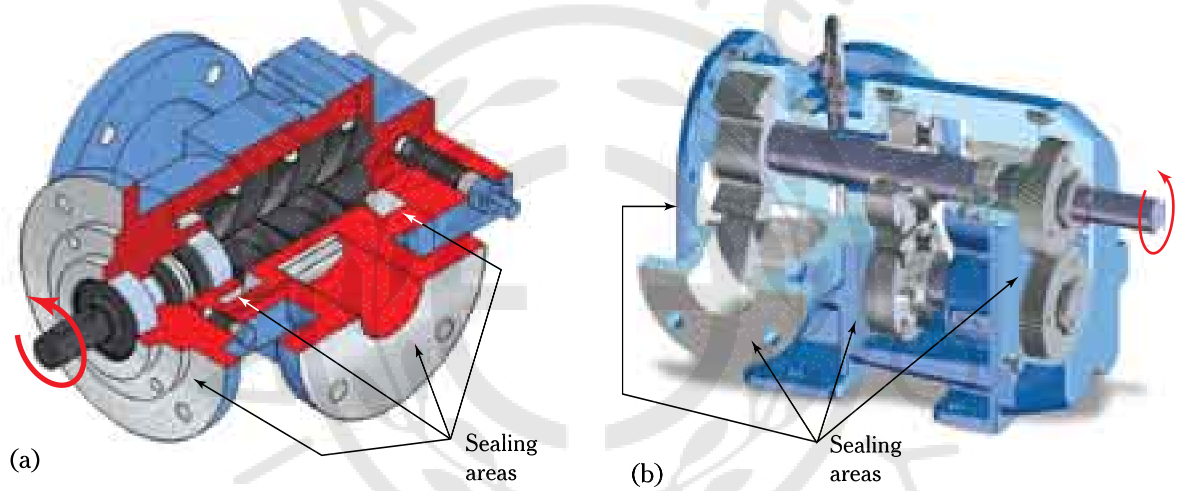

Fig. 9.7
Sealing points on pumps with (a) screws, (b) lobes.

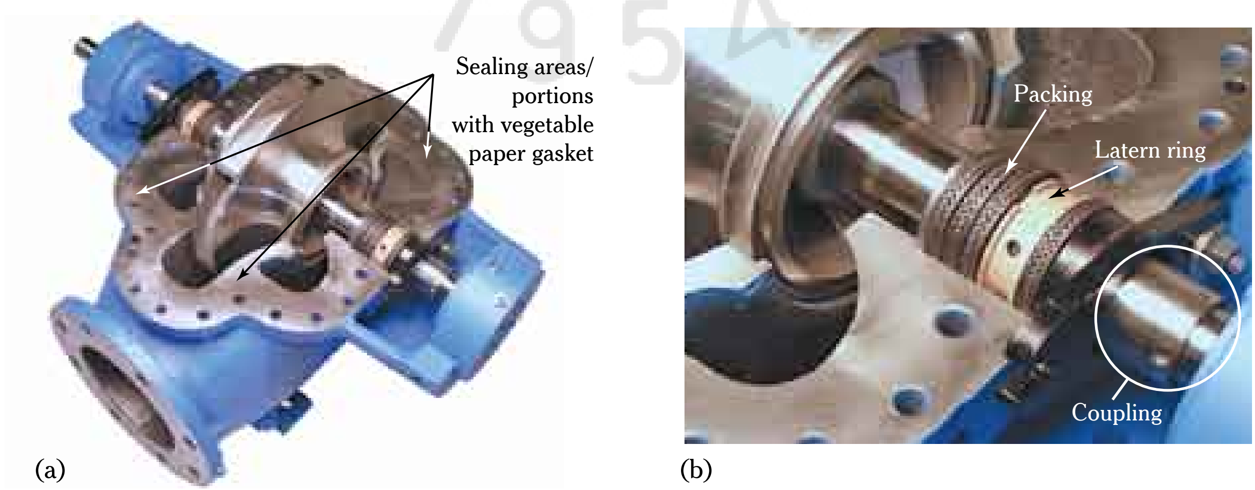

Fig. 9.8
Pump (a) with axial split casing (b) with installed packings.

Also, the seal may be achieved with o-ring, when the casing is closed with a cover on the top side.

At the point of the casing where the rotor shaft passes through, the seal is achieved by packings located in a specially designed stuffing box [fig. 9.8(b)] or by mechanical seals (fig. 9.9).

When packings are used, the significant friction resulting between the shaft and the packings causes temperature increase, which is prevented by adjusting the packings expansion, so that there is little leakage of the liquid. Hence, the small leakage of the liquid from the stuffing box acts as a coolant and as a lubricant to the space between the shaft and the sealant material in the stuffing box. Furthermore, for better cooling, packings may be divided into two groups by a ring with peripheral holes ***(lantern ring)*** [fig. 9.8(b)]. The cooling liquid, when the pumps convey water or non-volatile liquids, is provided by the pump itself through a suitable pipe which is connected to the discharge side of the pump, or to the second stage in volute casing pumps, and leads the cooling liquid to the stuffing box. Otherwise, the cooling liquid is provided by an independent external source.

In normal operating conditions, leakages should not be displayed when mechanical seals are used (fig. 9.9). Their cooling is accomplished either by liquid, as it is handled by the pump, or by a small pipe providing liquid linked to the discharge of the pump.

9.2 Pump bearings.

In both types of positive displacement pumps, either reciprocating or rotary, as well as in dynamic rotary pumps, thrusts and friction are developed during the reciprocation or the rotation of the moving parts. Therefore, depending on the type of pump, appropriate bearings are used, in the following way:

1) In ***reciprocating positive displacement pumps***, bearings are placed on the connections between the pistons and the crank shaft, in order to face the thrusts of the moving parts. The bearings (or bushings), as in all machineries with a crank shaft, are constructed of suitable material, while the lubrication of abutting surfaces is realized by lubricating oil.

2) In ***positive displacement rotary pumps***, bearings are located on the housing of the covers, and constitute a means of support and alignment of the rotor shaft. The bearings may be of the rolling elements type, as well as sleeve-type slide bearings (oil -sliding contact bearings). The shaft rotates as it is connected on the inner ring on these bearings, which are stably installed on the cover.

3) In ***dynamic rotary pumps***, the placement of bearings aims to maintain the correct alignment of the shaft or of the rotor with the pump fixed parts and to counter the axial and radial thrust that is developed. Even for pumps with the same basic design, the type of bearings that could be used may differ, since it depends on the operating conditions of the pump and the buyer's preferences. However, in

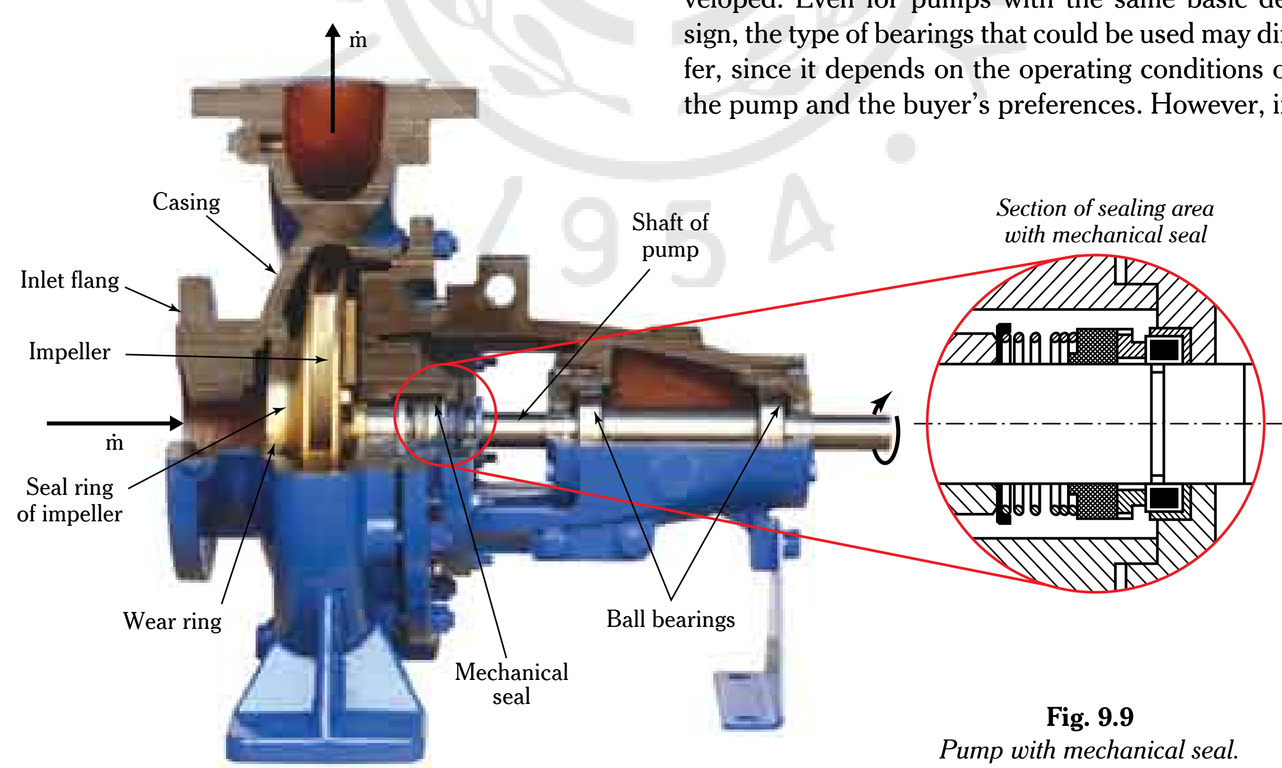

Fig. 9.9
Pump with mechanical seal.

most cases the bearings that are encountered today are either rolling element or sliding contact bearings. The bearings of dynamic rotary pumps, due to their particularity, are presented in the following paragraphs.

– *Bearings of dynamic rotary pumps.*

Although theoretically the impellers of dynamic rotary pumps are hydraulic balanced, in practice this can be rarely achieved. So, even in pumps with double suction impellers or with impellers in opposite arrangement, as in multi-stage pumps, it is necessary to use bearings to damp the thrusts caused by:

1) Vortices caused by the flow of liquid inside the pump, despite the fact that the casting, used in the construction of the casing, possibly reduces defects on flow surfaces.

2) Unequal wear of the rings between the impeller and the casing and

3) unequal axial and radial thrust caused by the flow of liquid from the impeller, due to incorrect suction pipe positioning.

So, bearings are used that are designed to face the radial and axial thrusts which are developed by the pump. When these are used to maintain the radial position of the rotor, they are called radial bearings or alignment bearings (***lining bearings***), while the bearings facing axial thrusts are called ***thrust bearings***. Thrust bearings may, in most cases, facilitate both the axial and radial thrusts.

In horizontal arrangement pumps (fig. 9.10) with bearings at each end of the shaft, the bearings which are placed between the housing and the coupling of the drive machine are characterized as internal, whereas those placed at the other end of the shaft as external.

The installation of bearings is performed on a portion of the pump casing or on the outer shell with a separate support base (this is typically found in larger pumps) (fig. 9.11). The support casing constitutes the chamber for the lubricant used for lubricating and cooling the bearing (fig. 9.12).

The lubricant may be grease or oil. The oil may be supplied through an external device, providing continuous flow, or contained in a chamber between the pump and the drive machine, entrained by the bearings during operation. When a chamber with lubricant is used for the lubrication, it is necessary to maintain the oil level within the limits specified on the sight glass, otherwise the smooth functioning of the bearings is affected.

As a rule, if the lubricant and coolant for the bearings is grease, the filling of the housing containing the bearing, should be up to one third of the housing. Greater amount of grease does not allow its circulation, resulting to insufficient cooling of the

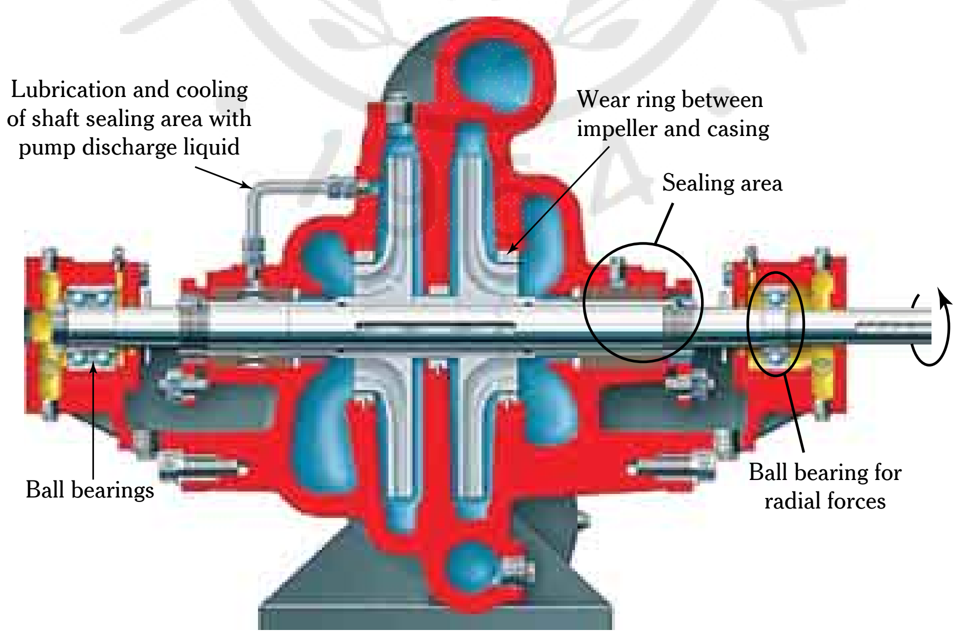

Fig. 9.10
Impeller between bearings in horizontal arrangement pump (section).

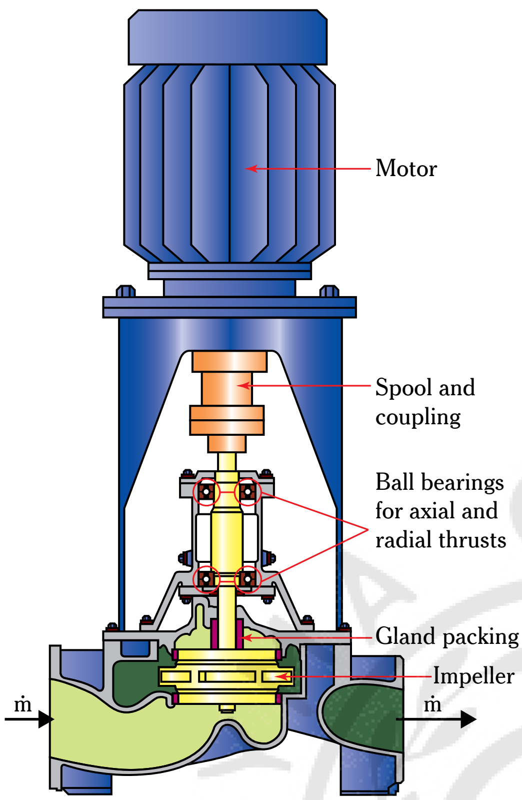

Fig. 9.11
Bearings on one side of the axis, between the rotor and the drive motor in vertical arrangement pump (section).

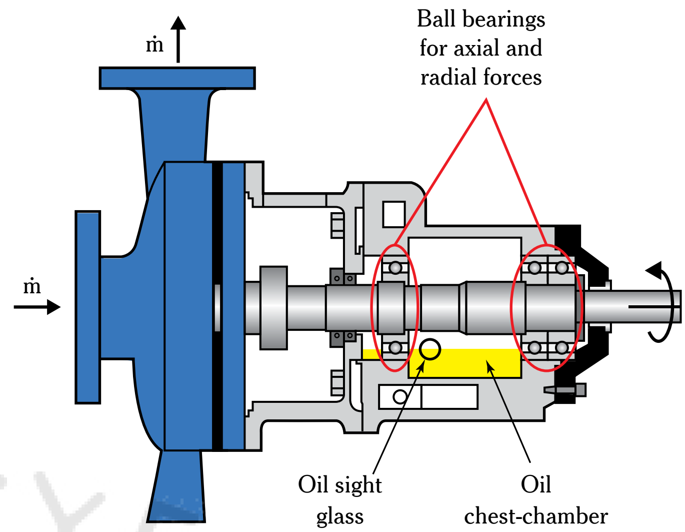

Fig. 9.12
Horizontal pump with chamber for the lubricating oil.

bearing. Furthermore, insufficient cooling will lead to increase in the temperature of the grease, and rapid wear of the bearings, because the grease will be liquefied and will leak from the sealing points where the shaft passes the chamber.

The types of bearings which are used are the following:

1) ***Ball bearings*** (with spherical rolling elements) (fig. 9.13), which are commonly used in pumps, because they can face both radial and axial loads, the only limitation being that the load of the supported system should not be high. The bearings are manufactured in two types; the open type, where, for the installation, initial lubrication is essential, and the closed type, where the lubricant is contained within the bearing by the manufacturer.

2) ***Roller bearings*** (with cylindrical rolling elements) (fig. 9.14), which are designed to face radial loads with heavy weight, because the rollers allow the distribution of the weight exerted by the system on a larger surface. These types of bearings are not suitable for facing the axial thrusts.

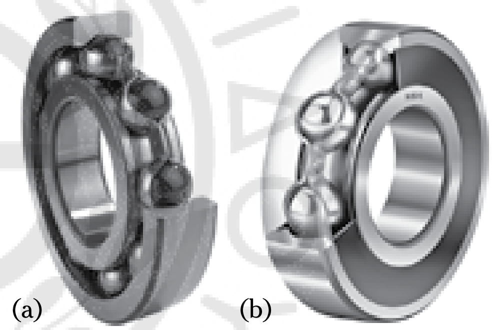

Fig. 9.13
Bearing with spherical rolling elements (ball bearing) (a) open and (b) close type.

Fig. 9.14
Bearing with cylindrical rolling elements (roller bearing).

3) ***Self-alignment taper roller thrust bearings*** (self-adjusting bearings with sliding conical elements) (fig. 9.15), which are commonly used in vertical arrangement pumps and have the capacity to handle bigger axial loads of a relatively heavy-weight pump rotor, while being able to align the shaft in case of small deviations.

Fig. 9.15
Self-alignment taper roller thrust bearing.

4) ***Sliding contact bearings*** (sleeve-type sliding bearings) (fig. 9.16), which are used when the operational conditions (such as speed), the axial and the radial thrust no longer allow the use of bearings with rolling elements, or when, due to pump construction, it is necessary to use pumped fluid to lubricate the bearings.

The operation of all these types of bearings is based on their hydrodynamic design as well as on the circulating fluid viscosity. These two factors influence their ability to manage the applied loads. Under normal operating conditions, the lubrication is achieved by the circulating liquid itself, with the shaft and the bearings being completely separated by the lubricant membrane. Also, during the manufacture of the bearings the corrosive properties of the liquid should be taken into account as well as the fact that the contact of the shaft with the bearing leads to rapid wear. The construction material is rubber, carbon, metal alloy such as copper, brass, cast steel, phosphor bronze or ceramic, with several longitudinal or helical grooves.

Fig. 9.16
Sliding contact bearings.

The various types of bearings (fig. 9.17) are:

1) The sleeve type; it is the simplest and is used at low rotation speed pumps.

2) The counter-dividing type (countervailing split sleeve bearings); the inner surface is hydrodynamically designed so, upon rotation of the shaft, a lubrication wedge is created that assists in the alignment of the shaft.

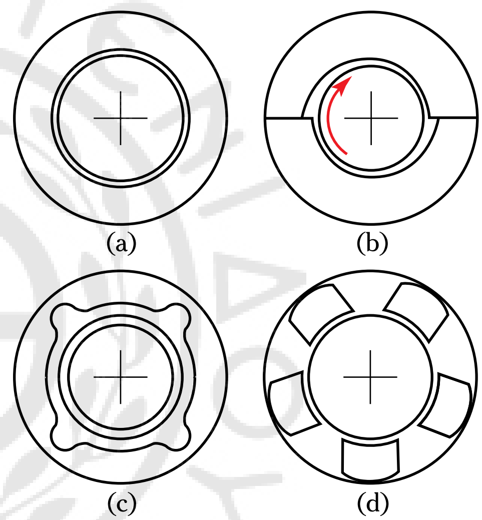

Fig. 9.17
Sliding bearings: (a) sleeve bearings, (b) counter-dividing split bearings, (c) multi-lobe bearings, (d) tilting arc-shaped pad bearings.

3) The multi-lobe type; it is commonly used in high speed pumps that have the ability to manage low loads. This kind of bearing is often found in ship's pumps, and its interior may be coated with synthetic material.

4) The tilting pad type (tilting pad bearings with an arc layout); it has the ability to vary its tilt by exerting load. This is achieved because, as the shaft rotates, the lubricant pressure increases, creating oil wedges.

9.3 Drive transmission on pumps.

One of the pump installation elements is the component that is used for the transmission of motion by the drive machine. The type of this driveline component depends mainly on the type of pump, the motor and the yield-speed relationship between motor and pump. The major ***coupling modes*** for transmitting motion from the shaft of the drive machine to the rotating shaft of the pump are:

1) The ***direct coupling*** of the motor to the pump through a rigid (fixed) or flexible link. These are used in dynamic and rotary positive displacement pumps, which operate at high speed, enabling their coupling with motors, such as electric motors and diesel engines. But, in order to achieve direct coupling, the following prerequisites must apply:

a) The pump and the motor must be designed to operate at the same speed and the same direction of rotation.

b) The axial lines of the rotation shafts should coincide, whether the connection is made in short or in long shafts. The particularity of links in direct coupling shaft and their frequent use in ships' pumps is the reason they will be further analysed on page 209.

2) The ***gear coupling*** (with geared wheels or gears), which is used in the transmission of motion when:

a) Speed reduction is required, as occurs during transmission from the motor to a reciprocating pump with crank or from a steam turbine to a centrifugal pump.

b) The drive shaft and the pump rotation axis do not coincide, as happens in the transmission of motion from a horizontally installed engine to a vertical arrangement pump.

The gears may be surrounded by a housing or, rarely, are located outside the housing when low-speed rotation shafts are used. When gears are placed inside a housing, lubricant also exists within it, and is used to lubricate and reduce the temperature that develops during operation. Alternatively, particularly in high speed engines, where the degree of speed reduction attributed to the pump is relatively high, the circulation of lubricant to the gears is performed by an attached or independent oil supply pump, such as when steam turbines are used as drive machines, and the reduction of the rotational speed of the pump is achieved by gears.

3) The ***magnetic coupling***. In pumps with conventional coupling between the rotating shaft of the impeller and the stable casing, complete sealing is impossible. This is due to the small leakages that are present even in cases where mechanical seals are used. However, even small leakages may cause problems to safety and pose a risk to the environment or to health when toxic fluids or hydrocarbons are transferred. So, the requirements of regulations which are adopted for the prevention and limitation of potential leaks from pumps (SOLAS Requirement II-2 Regulation 4) can be met by the magnetic shaft coupling system.

The magnetic coupling system (fig. 9.18) con-

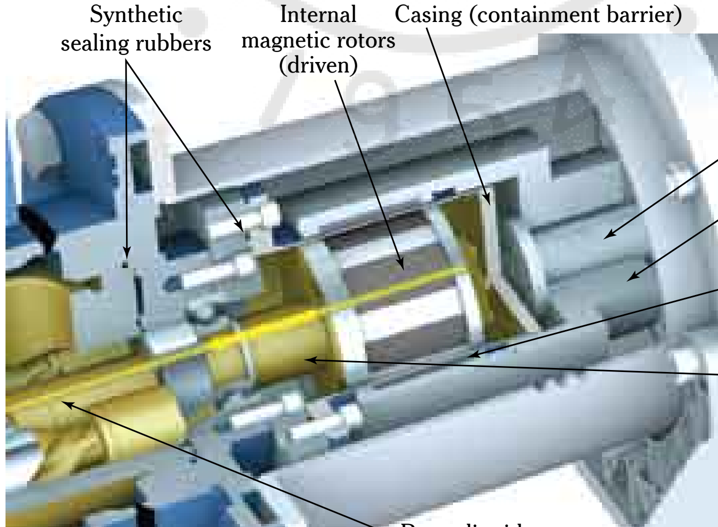

Fig. 9.18

Magnetic coupling system cross section.

sists of an outer rotor driven by a motor, and on the inner periphery of the outer rotor permanent magnets are installed. Respectively, an inner rotor with permanent peripheral magnets is installed on the shaft of the pump impeller. This rotor is surrounded by a metal shell which is sealed by synthetic o-rings (rubbers) placed on the pump housing. Round the shell that surrounds the inner rotor, the outer rotor with permanent magnets is rotated, driven by the motor. This creates the most important structural advantage of the magnetic shaft coupling, one that allows it to address the leakages. The liquid does not come into contact with the external environment at the point where the shaft crosses the shell. That is, the pump shaft and the inner rotor installed in it (the inner rotor with permanent attached magnets) rotate in an integrated shell (containment barrier) inside the pump.

The torque of the pump is accomplished by the rotating magnetic field generated by the permanent magnets of the outer rotor, which is connected to the motor. The torque causes the rotation of the shaft as it is transmitted to the permanent magnets of the inner rotor and thus the rotation of the liquid handling rotor is obtained. The fluid handling rotor can be of an impeller or screw (fig. 9.19) or gears type, satisfying the applicable low viscosity fluid handling requirements (up to 1.4 cSt) [ISO 8217: 2005] and low sulphur fluid handling requirements (0.1%) [EU / SECA 2005-33-EC].

For the cooling and the lubrication of the inner rotor with the permanent magnets, a small proportion of the liquid that is handled flows into the shell of the magnetic coupling, and then returns to the pump pipeline suction through the shaft (figs. 9.19 and 9.20). Also, in the magnetic coupling system located inside the pump, there are electrically insulated rings in order to absorb the internal pressures which are generated during pump operation.

Thus, the magnetic coupling system is advantageous compared to other systems because, in addition to preventing leakages of liquid or gas, it has minimal maintenance requirements and the alignment of the system is easier, since there is no contact between the parts that are used for motion transmission.

4) The ***chain drive movement***, by which reduction of speed is achieved, but it is rarely used in ship pump systems.

5) The coupling with ***drive belts*** (transmission belts), where the motion is transmitted smoothly by means of suitable pulleys that are applied to the shaft ends. With these, it is possible to decrease or increase speed (by selecting the appropriate diameter relation between driving and moving pulley), as well as the possibility of changing the direction of rotation or the transmission of motion when an angle is formed between the drive shaft and the pump shaft. The drive belts provide flexibility as regards the motion transmission, allowing small deviations between the axial positions of the shafts, easy fitting with satisfying efficiency and low construction and maintenance costs compared to direct transmission couplings and gears.

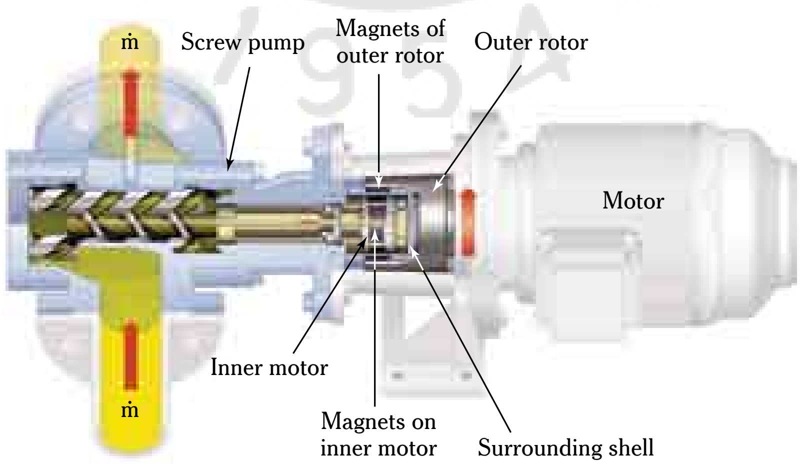

Fig. 9.19
Magnetic coupling in a screw pump and the liquid flow of lubrication and cooling of the magnetic coupling system.

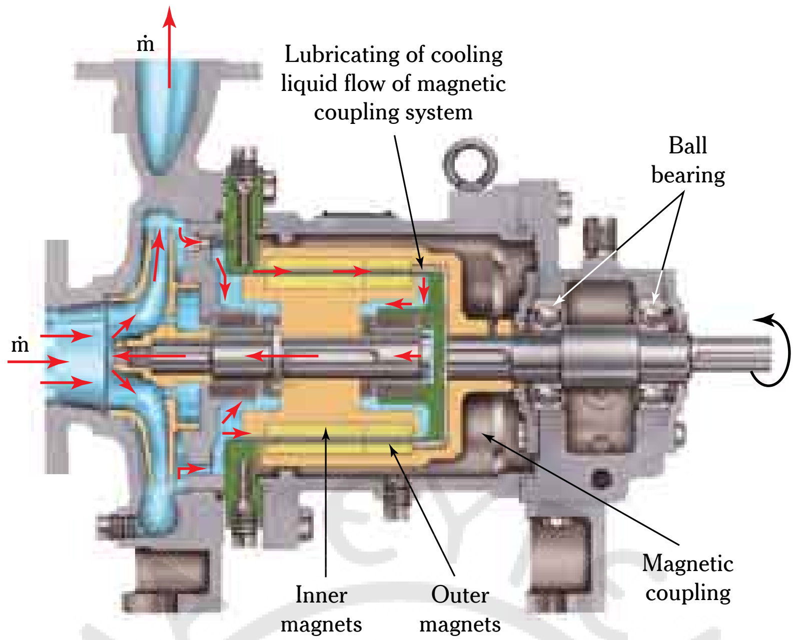

Fig. 9.20
Magnetic coupling in a centrifugal pump and liquid flow lubrication and cooling of the magnetic coupling system.

9.3.1 Direct shaft couplings.

Drive couplings with direct connection are used, as previously mentioned, to transmit the torque load between two shafts in line, namely with their axial lines coinciding. With the coupling, the motor and the pump are designed to rotate at the same speed. Another function of direct coupling application is the required connection of the two shafts, which are not in perfect alignment to each other, since any existing minor shaft misalignment can potentially be absorbed. Therefore, there are different types of couplings which display different functional and constructional characteristics. Based on these, they can be classified into rigid and flexible coupling. In more detail:

1) ***Rigid coupling*** is mainly used in the connection of shafts with ***perfect alignment***. The slightest divergence results in the appearance of significant tension in the connection and the shafts due to the stable contact of the sections that constitute the coupling. The coupling consists of two metallic components or sections which often have the form of a flange (fig. 9.21). The internal diameter of each connecting section is approximately as much as the external diameter of the shaft, in order to fit perfectly on the pump shaft and the motor. Internally, a groove is formed where a pin is placed, while the same groove with a pin exists respectively in the shaft, in order to ensure that the connection section will rotate with the shaft without the risk of being moved. The stability of the coupling is further reinforced by a screw which functions as safety and prevents the axial displacement of the coupling after assembly. Alternatively, in flange connections, a metal spacer or spool is used, which connects the

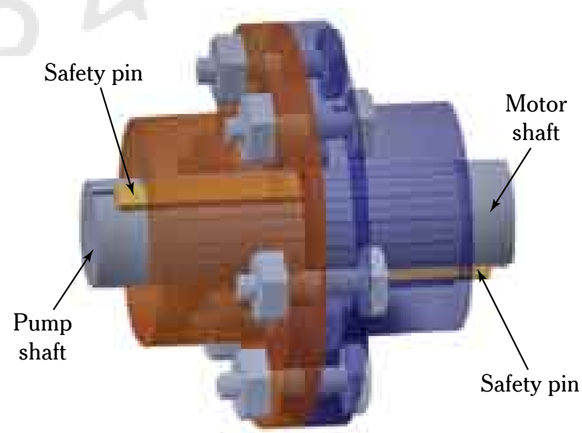

Fig. 9.21
Direct coupling (flange type).

two coupling flange sections of the shafts (fig. 9.22). Besides the connection, the spacer offers flexibility during the disassembly of the pump, which is done without it being necessary for the housing and the pump motor to be shifted from their base.

The connection of coupling sections is done with the contact of metal on metal, while their contact is accomplished by screws, which are fitted into holes on the body of the flanges and nuts.

Besides the flanged couplings, there are also other types of rigid couplings, such as ***sleeve coupling*** [fig. 9.23(a)], ***mull coupling*** [fig. 9.23(b)] and ***split type sleeve coupling*** [fig. 9.23(c)].

2) ***Flexible couplings*** consist of metal parts in the form of flanges that combine elastomeric segments. During the coupling parts connection, no slip is presented, despite the use of elastomeric parts. Also, flexible couplings enable the absorption of small angular and radial inclinations as well as of axial displacement between prime mover and driven shafts. The presence of flexible (elastomeric) elements which are used in the coupling of metal sections significantly contributes to the damping of impact loads which appear during pump starting but also during pump operation.

The flexible drive couplings can be ***rings*** which are placed on the attachment screws of the metal flanges [fig. 9.24(a)], ***rubber spider couplings*** (or elastomeric jaw couplings) which alternate between the teeth created by the flange connections [fig. 9.24(b)], or ***flexible shaft couplings*** which surround the two flanges at their contact point [fig. 9.24(c)], respectively creating three different types of flexible connections. One more type of flexible connector is the gear coupling [fig. 9.24(d)]. It consists of two metal components with peripheral teeth, which are placed one on each axis and are surrounded by a single or split ring with recesses involved in the respective teeth. Depending on the manufacturer, there may possibly be other types of flexible connectors, such as those with a grid (***grid couplings***) [fig. 9.25(a)], or a chain (***chain couplings***) [fig. 9.25(b)] at the connection of the two sections.

9.3.2 Alignment of movement transmission couplings.

To operate a pump without the presence of fatigue and load caused by the connection of the driven shaft to the drive shaft, it should be aligned. As regards the rigid couplings, the alignment must almost be perfect, while as regards the flexible couplings, there might be a small deviation, without it

Fig. 9.22
Spool (or spacer) between couplings which facilitates the disassembly of the pump.

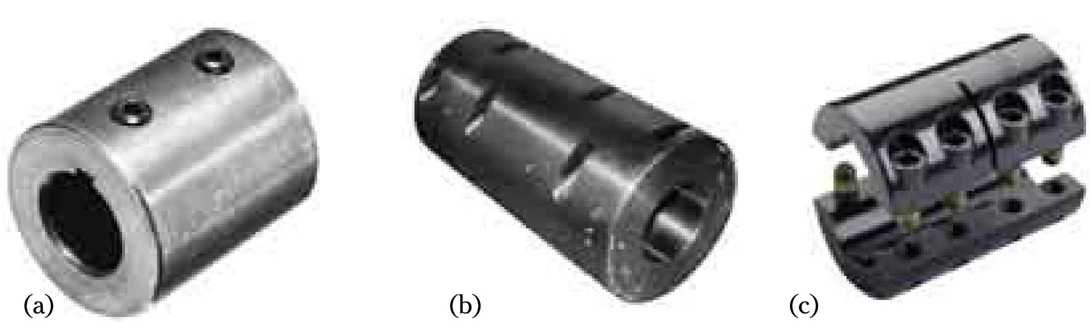

Fig. 9.23
Types of rigid couplings.

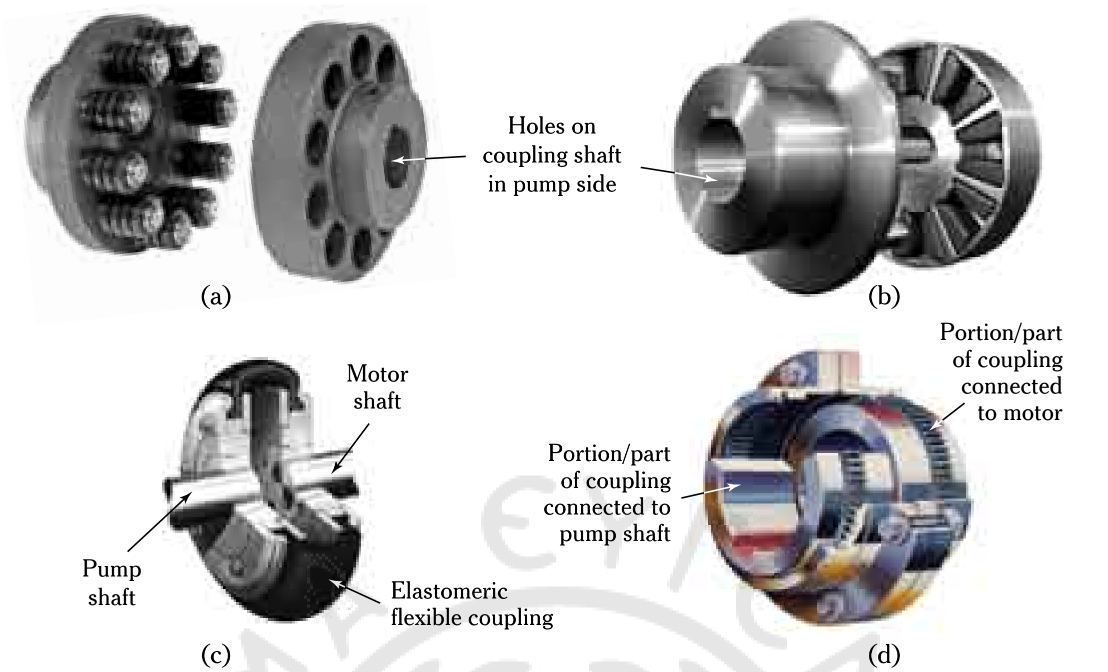

Fig. 9.24

(a) Coupling with flexible rings, (b) rubber - elastomeric jaw coupling which alternate between teeth, (c) flexible shaft coupling which surrounds the two flanges, (d) gear coupling with peripheral teeth.

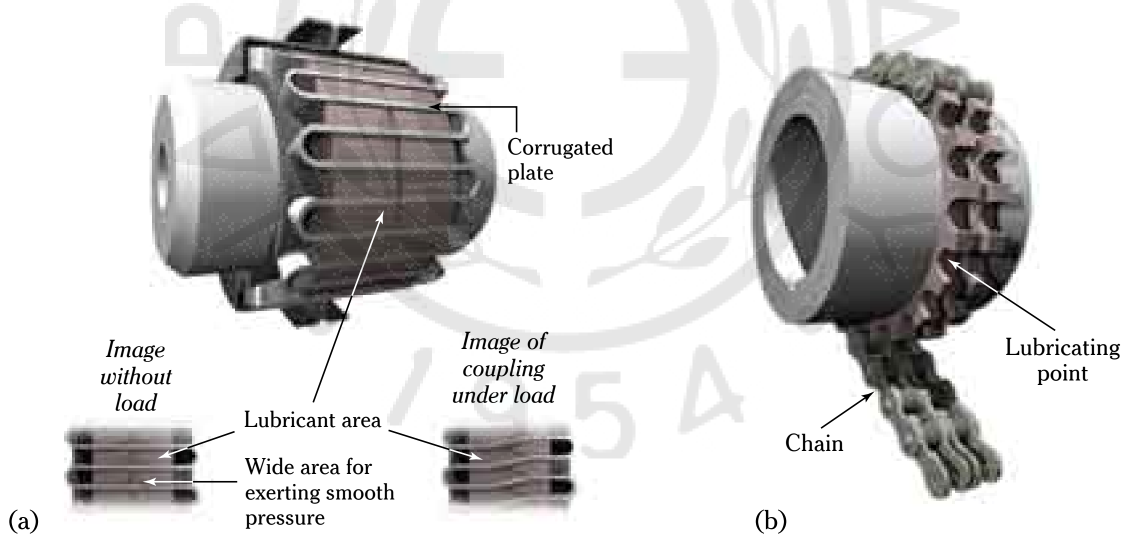

Fig. 9.25

Couplings (a) grid coupling (with corrugated plate), and (b) chain coupling.

meaning that the use of flexible couplings eliminates the need for alignment. With the use of flexible couplings, the deviation is usually caused by deformation of the flexible elements, and, as a result, the life span of these elements depends on the degree of alignment deviation (or misalignment).

There are three main misalignments which are caused by the shifts of the shafts:

1) ***Axial misalignment*** (x), where the main axle lines are aligned at the common centreline, yet the axial position is incorrect, making the axial movement possible during pump operation [fig. 9.26(a)].

2) ***Parallel misalignment*** (or radial) (y), where the axles are parallel but they are not at a common centreline [fig. 9.26(b)].

3) ***Angular misalignment*** (a), where the cen-

trelines of the respective axles are not parallel but form an angle [fig. 9.26(c)].

The shift may constitute the combination of all three cases, so we might have a combined misalignment x, y, α [fig. 9.26(d)], while individual misalignments may appear in changes of operating conditions. A typical cause of misalignment is the temperature increase of the operating environment or the transmission of heat from the liquid flowing into the pump and the piping system. The changes, which may be caused during operation, must be taken into consideration both by the manufacturer as well as by the pump operators when an alignment, after a repair or after a planned inspection, is conducted.

The coupling manufacturers provide information concerning the maximum permitted deviations for every kind of deviation. Therefore, it is important to be aware of the maximum permitted value of alignment deviation and how it is affected by the speed and the torque transmitted. For example, when flexible couplings are used on pumps powered by an electric motor, the maximum permissible radial deviation is 0,1 mm. The maximum permissible angular alignment deviation is usually the one which causes changes to the distance between both parts of the coupling by 0,05 mm, when this measurement is done on one perimeter point of the coupling during manual rotation of the shaft.

The inspection and the alignment of the shaft and the coupling can be done in different ways, like with an alignment ruler, and a feeler gauge[4] (fig. 9.27(a), with a specific instrument called dial gauge [fig. 9.27(b)] for greater accuracy or by using Laser

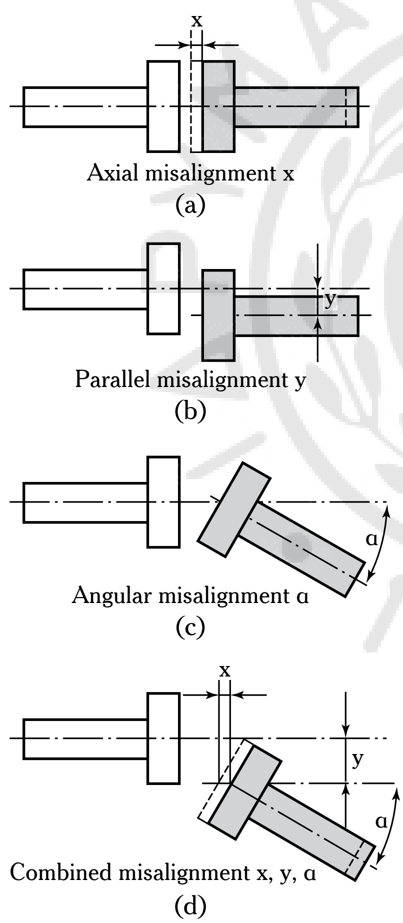

Fig. 9.26
Positions of shaft misalignment, which, in very small scales may be offset by the tolerance of the flexible couplings.

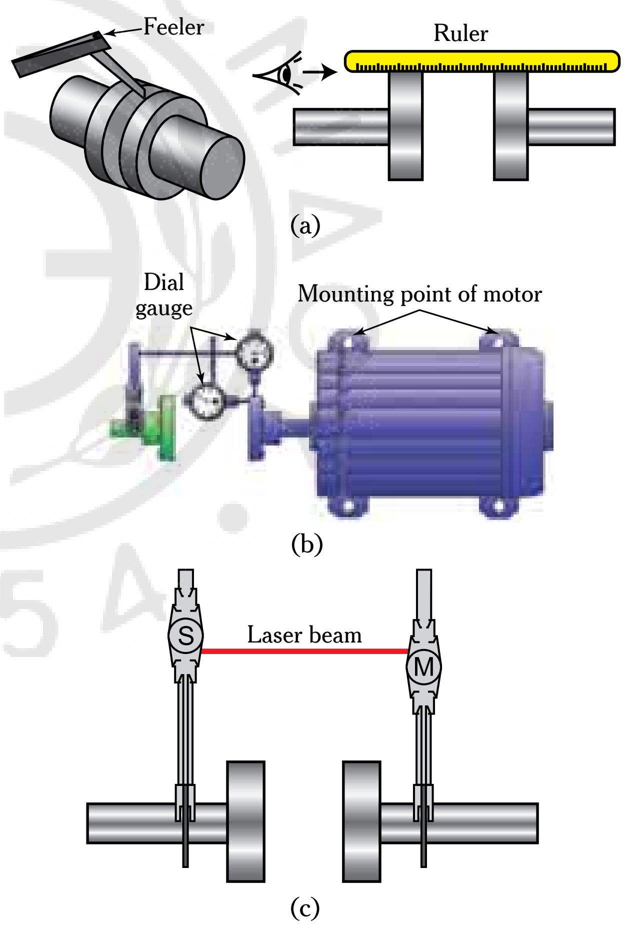

Fig. 9.27
Shaft and transmission coupling alignment methods.

[4] ***Feeler*** gauge is a tool with calibrated thickness metal foils used in engineering to measure the clearance between two parts. Also feelers are used either one by one or in combination, and the sum gives the distance between the two coupling sections.

gauge [fig. 9.27(c)]. It must be noted that alignment is necessary at the horizontal arrangement pumps and rarely at vertical pumps, because the support bearings on the vertical arrangement of the shafts and the correct assembly of the movement transmission coupling at the same points before disassembly do not create misalignment.

To achieve horizontal alignment in the pump device, at the four points of support in the motor base [fig. 9.27(b)], shims (or spacers) by thin metal sheets are used for correcting the pump position. By adding or removing shim and small motor displacements, in combination with controling the coupling, the alignment process can be summarized in the following steps:

1) Initially the motor support bolts are loosened and an optical alignment with the pump takes place, to verify that the coupling is not crushed in any way, while the screws that join the two parts of the coupling must be removed.

2) The measurement device(s) is attached. When a dial gauge is used, the indicator(s) is checked, making sure it can move freely in the measured area. When a ruler is used, it is placed at the interface of the two coupling sections.

3) A possible distortion of the motor mounting is checked, by tightening and loosening the support screws separately.

4) The dial gauge is adjusted, so that the indicator(s) is set to zero, and if a straight ruler is used, it must abut both parts of the coupling.

5) The shaft is rotated 180°, about half a turn, and the position of the motor is corrected in order to repeat the inspection of the angular and radial.

6) By controlling the distance of the coupling parts from the dial indicator or the feeler, the axial deviation is corrected, until the reading comes within maximum allowable variation for that coupling style.

7) The steps 5 and 6 are repeated until the coupling is aligned, while gradually the screws of the motor support are tightened with continuous monitoring of alignment.

8) A transverse alignment is performed in the same manner, as in the vertical plane.

9) The final inspection of the alignment is carried out in both vertical and lateral directions, and the angular deviation as well as the conditions at the time of alignment are noted, for example cold or warm pump motor and recorded for future reference.

By correct alignment, the optimum operating conditions of the pump can be achieved, which include decrease in axles and bearings fatigue, decrease in movement transmission couplings fatigue, decrease in vibrations and maintaining the normal temperature of the machinery (fig. 9.28).

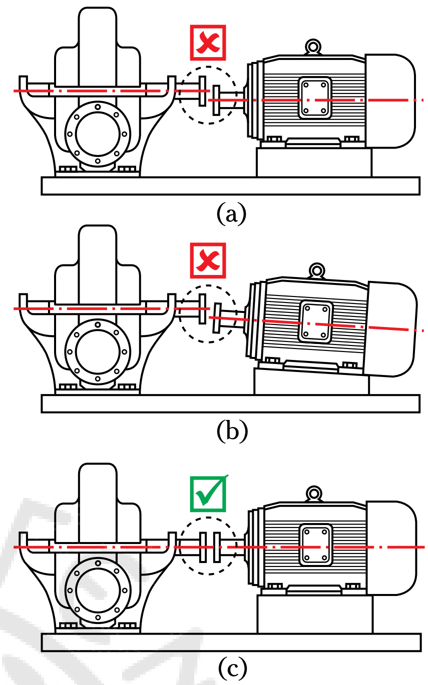

Fig. 9.28
(a), (b) Misalignment (incorrect-arrangement).
(c) Shaft aligment (suitable arrangement).

9.4 Pump wear.

Pump wear is a result of erosion and mechanical stress, and constitutes a high economic burden on the ship's budget. Furthermore, in order for a pump to respond to the particular environmental operating conditions, it is necessary for it to be made of durable material suitable for the fluid that is handled. But despite the strength of the construction material, continuous operation causes damages in different areas, depending on the pump type, affecting its performance. Thus, for each type of pump one can mention that:

1) In ***piston pumps***, wear appears at the piston rings, at gland packings that seal the piston rod and at the valves. The wear in the glands can easily be detected due to the external leakages, so appropriate maintenance and repair actions should be taken by the ship's crew. However, a leak due to the wear of the piston rings and the valves creates internal

leakages which can only be detected by crew's attention, since the pump flow and the pressure that develops inside the cylinder are decreased. To address the wear, maintenance jobs are performed, which include replacement of the piston rings, the valves or the sealing elements of the valves. These parts may be constructed by copper alloy or elastomeric o-rings.

2) In ***positive displacement rotary pumps***, the wear appears in the leakages between the rotating parts and the casing. It is essentially a function of the pressure exerted on the surfaces of the pump components and the frictions developed among them, or a combination of both. Moreover, the heat which is developed per surface unit depends on the pressure and the friction due to the number of rotations and contributes to the wear of the components. For these reasons, the impellers, especially for pumps which transfer ***liquid with abrasive properties***[5], are either constructed by durable material or function at lower rotation speeds. If the reduction on rotations still does not limit the wear conditions, then the surface pressure at the pump components is reduced. Exception to these ways of addressing the wear is the use of flexible materials for the construction of liquid transfer rotors which prevent the leakages. The transport of liquid in peristaltic pumps constitutes an example.

3) In ***dynamic pumps***, the wear appears in areas between the impeller and the pump housing (where its suction space is separated from the discharge) and in the staffing box (where the rotating shaft of the impeller passes through). The gap[6] between the casing and the impeller is quite small, to prevent leakages from high to low pressure, but, at the same time, to allow the free rotation of the pump. In order to lower the repair cost of the pump from wear in the housing, ***wearing rings*** are installed, which are periodically replaced. The wear of gland packing is due either to the quality of liquid, e.g. when solid particles or sand are entrained by the liquid and result in the destruction of the surfaces which come into contact at the sealing point, or due to normal wear after a long period of operation which will lead to an external leak. That, depending on the mechanism used, is dealt with the replacement of the mechanical seal or of the packing rings.

9.5 Starting and pump operation in general.

The installation and operation of each pump depends on its type and is described in the manufacturer's instruction book. Furthermore, the starting and performance is related to other factors, such as the position of installation in relation to the system that it serves, the number of pumps used in the same pumping system, e.g., if two pumps in one pumping system operate in parallel to each other or in series. Generally, regarding the starting and operation of pumps, one can mention that:

1) The pump installation must be as close as possible to the suction source of the liquid so that the operation, inspection and monitoring is facilitated.

2) The piping system must be simple, with the smallest possible number of curves and the right alignment, so as to prevent the creation of air cavities that impede the liquid flow.

3) The motor of the pump must be protected from exposure to humidity, even if defined by the regulations as being constructed as a protected type.

4) The base of the pumps must be reinforced so as to withstand the ship's vibrations and the tendencies which are developed during the start of its operation.

5) The coupler of the pump with the electronic motor must be properly aligned, because a misalignment can create wear and overheating of the bearings which lead to damage of the motor and the pump.

9.5.1 Pump starting procedure.

Prior to starting a pump, the execution of the following preparation checks is indispensable:

1) ***Check the cock***[7] ***valves*** of the monitoring devices, such as pressure gauges, thermometers, etc., to ensure that they are open.

2) ***Check the lubricant supply in the bearings*** to ensure adequate lubrication and cooling or open the cooling supply in the water-cooled bearings, if any, and in the glands. This prevents wear and overheating of the bearings, which may lead to other damage.

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[5] ***Fluid with abrasive properties*** is a liquid which has the potential, due to its content of suspended solid particles, to smooth the surfaces with which it comes into contact.

[6] ***Gap*** or ***clearance*** is the small distance measured between parts or components.

[7] ***Cock valves*** are the small valves that are used in order to stop or supply a fluid to the monitoring gauges.

3) ***Check for sufficient quantity of liquid inside the pump*** for initial starting (depending on pump type). If the liquid level that is sucked is higher than the pump's highest point or the suction system is under pressure, then, the filling of the pump with liquid can be achieved by opening the suction valve and the air vent valve that is installed at the top of the shell (then close it again). Otherwise, if the liquid level is below the pump shell center, the filling with liquid as well as the air extraction is achieved as follows: in positive displacement pumps, by opening the air vent (on discharge side, then close it again), and in dynamic pumps, by a priming pump or an external source of liquid supply.

4) ***Check that the suction valves*** of the pump are open.

5) ***Check the pump shaft*** and ensure by hand that the pump shaft is rotating freely (power is OFF at this point), otherwise it must be checked and corrected. For new pumps, at first starting or after maintenance, it must also be checked that the motor rotation is the same as that required for pump operation.

6) ***Check the discharge valve***, which:

a) In positive displacement pumps, it must be open.

b) In centrifugal pumps, at starting the discharge valve must be closed or almost closed. In this manner, the minimum load of the electric motor which operates the pump is required. Hence, less power is required, which, at starting (if the discharge valve is open), due to overload, reaches the cut-off point of the power supply to the motor. This practice is intended to protect the pump motor (which is designed to operate at the maximum discharge head), and due to resistance to the flow (back flow) of the piping system by the liquid present in it, it results in an increase in power consumption. After starting, it is necessary for the discharge valve of the pump to the piping system to be opened. On ship piping systems where a check valve is installed to the discharge, the disc opening is realized by the liquid, since the operation of the pump causes the liquid pressure to increase.

c) In axial and mixed flow pumps, at start-up the discharge valve should be open, because the torque reaches its nominal value before the pump reaches the normal operating speed.

d) In ejectors, at the outlet (if any) the discharge valve should be open at starting. Also, the suction valve should not be open if the flow of the operating fluid does not reach the normal operating pressure.

9.5.2 Pump operation.

After starting and during operation of the pump, the following must be periodically inspected:

1) The operation of the pump and the motor must be smooth and quiet (e.g., check for noises of metallic percussions, whistles, or other noises beyond the normal ones).

2) The temperature of the shell bearings and the motor (if used as a drive machine) must not exceed the permissible limits.

3) The motor power consumption must be kept constant, according to the operating characteristics of the pump that are designated by the manufacturer. It should be noted that the variation that occurs in the power supply parameters (Amperes, Volts) during starting of the pump motor are considered normal if they do not exceed the limits set by the manufacturer.

4) The developed pressure and flow rate, should be kept constant and consistent with the structural features of the pump, depending on the operating conditions.

5) The leakages to the glands should be maintained at a minimum, since, in this way, pump shaft overheating is prevented. When the sealing is achieved by mechanical seals, there should not be any leakages.

Throughout pump operation, it is essential to perform checks at regular intervals, in spite of the fact that control mechanisms and automations are installed in modern ships. By these checks, it is determined that the pump operates normally, and serious damages that may be caused to the pump by sudden failures are avoided.

To stop a pump, after the desired duration of operation, switch off the driving machine and then close the suction and discharge valves, and the valve of the cooling line for the glands, if any.

Attention: In centrifugal pumps, to avoid the occurrence of hydraulic hammer, which may appear if the pump stops suddenly, gradually close the discharge valve, then switch off the motor, and finally close the suction valve.

9.6 Pump maintenance.

The term ***pump maintenance*** refers to several inspections and works that must be performed at regular intervals during the pump's operational "life". Considering that the wear of a machine is roughly proportional to the square of operating time, prudential controls as well as repairs on defects and wear that are detected manage to:

1) Protect the pump from serious damage,
2) extend the duration of its operational life,
3) ensure the reliability and
4) limit the costs, since the failure is restored at an early stage.

As regards pumps maintenance, apart from external controls during operation, the lubrication of bearings, the control of glands and the transmission system (coupling) of the drive and the motor, regular inspections-repairs are performed which are determind by the manufacturers. These inspections are made by disassembling the pump, and include:

1) Inspection of bearings.
2) Control of packing rings on stuffing box or the mechanical seal of the shaft.
3) Alignment of the coupling and its elastic elements, for possible damage.
4) Inspection of the internal surfaces of the housing for damage from erosion and cavitation.
5) Measurement of the rotor bearings for wear.
6) Control of the radial clearances between impeller and wear rings.
7) Inspection of the rotor surfaces for possible cavitation.
8) Checking the stability of the rotor support on the shaft.
9) Inspection of the sealing to the suction and discharge valves, if any.

Components that exceed the operating limits of wear (according to the ones specified by the manufacturer) are replaced with new, while spare parts numbers of those which have been used are recorded in a list for ordering and storage, in order to be readily available for use when necessary.

These days (almost) all repairs that are required, as well as all the necessary inspections, are contained in a table (check list) completed by the crew which performs the maintenance, and kept in a log file. This table is an aid not only during maintenance, but it is also used as a reference point for the pump status and provides evidence during the audits carried out by the authorities.

The malfunctioning of pumps may be due to various causes. Therefore, the symptoms of abnormal operation and the possible causes could be designated in tabular form, which includes proposals on how to address them.

This table, covering each pump type, is usually contained in the manufacturer's manual and it is a quick reference point which helps to solve the problems presented.

In order to become familiar with the possible malfunction causes of pumps and their respective damage restoration modes, such a table is given below (tab. 9.1).

Table 9.1: *Symptoms and possible causes of pump abnormal operation and proposals for restoring them.*

Symptom - Failure	***Possible causes***	***Restoration/Treatment/Solution.***
Weak or low pump suction	Impurities in the pump filter	Open (dismantling) and filter cleaning
	The tightening of screws (bolts) in the flange connection to the suction, the casing/housing of the pump, or the filter is incorrect.	Check and tighten the screws (bolts).
	The pump impeller rotates in the wrong direction.	Control and reversing of rotation direction.
	There is air or there is not enough liquid in the pump suction.	Filling of the pipe and the suction of the pump with liquid.

Symptom - Failure	*Possible causes*	*Restoration/Treatment/Solution.*
Weak or low pump suction	Leak in the sealing system (packing or mechanical seal).	Check sealing for damage.
	Wear of the pump.	Replacement of worn parts.
	Air in the suction pipe due to damage to the pipe (hole or rupture) or loose bolts in piping system.	Replace the pipe, tightening of the bolts (screws).
	Reduction of the cross section of the suction tube from fouling.	Replace or clean the pipe.
	Incorrect motor speed.	Check and adjust speed according to the construction characteristics of the pump.
No discharge of liquid of the pump, no flow.	Closed valve.	Check the valves.
	Low rotational speed of the pump.	Check the motor. In case of electric motor, the voltage should be checked. If the motor is driven by steam or compressed air, the correct pressure and flow rate should be checked.
	The pump rotates in wrong direction.	Check the direction of rotation.
	Incorrect motor speed.	Check and adjust speed according to the construction characteristics of the pump.
	The impeller or the suction filter in the piping system is completely clogged.	Disassembly and cleaning of impeller or filter.
	Wear of the pump.	Replacement of worn parts.
Low discharge of the pump.	Air inlet from suction or by the gland.	Check the suction system; replace the packings or the mechanical seal.
	Low rotational speed of the pump.	Check the motor. In case of electric motor, the voltage should be checked. If the motor is driven by steam or compressed air, the correct pressure and flow rate should be checked.
	High temperature of the pumped liquid (vapors within the pump - cavitation in suction).	Reduce the liquid temperature, reduce the speed of the pump.
	The viscosity of the liquid is greater than expected.	Heat up the liquid.
	Mechanical damage or wear, e.g., wear to sealing rings of centrifugal pumps, damaged impeller, damaged gland seal or mechanical seal.	Inspection, repair and replacement of worn parts.
The pump does not develop sufficient pressure.	Low rotational speed of the pump.	Check the motor. In case of electric motor, the voltage should be checked. If the motor is driven by steam or compressed air, the correct pressure and flow rate should be checked.
	Air in the liquid.	Venting the shell, check the suction system.

(cont. on the next page)

Symptom - Failure	***Possible causes***	***Restoration/Treatment/Solution.***
The pump does not develop sufficient pressure.	Wrong direction of rotation.	Check of the drive machinery.
	Mechanical damage or wear, e.g. wear of sealing rings of centrifugal pumps, damaged impeller, damage of seal.	Inspection, repair and replacement of worn parts.
Pump operates and excessive noise is produced.	Noises from motor.	Check motor bearings.
	Noises from pump.	Mechanical wear, stop the pump and check.
	Noises from pump - may operate in cavitation conditions.	Reduce the pump speed - regulate flow through the valve, reducing the temperature of the liquid handled.
	Noises from the pump bearings.	Mechanical wear, stop the pump and check the bearings, lubricate or replace.
	Incorrect alignment of the transmission link (coupling).	Alignment of coupling.
Leakage of liquid from the pump.	Wear on the sealing system.	Replacement/repair of sealing system.
	Wear of the sealing of the casing.	Replacement/repair of sealing area on the casing.
The pump suddenly stopped.	Foreign object has entered the pump.	Removal of foreign object, check the pump for damage.
	Damage of motor or of the drive machine.	Check the power supply to the motor and the panel fuses, check the drive machine.
	Possible damage in motor.	Disconnect the pump from the motor and rotate manually.
Lateral (side) wear of rotor.	Incorrect positioning of rotor.	Replace the rotor.
	Incorrect alignment.	Replacing rotor, check of alignment.
Wear of sealing rings within the pump.	Incorrect alignment.	Check alignment, replacement of rings.
	Wear in bearings.	Check the bearings, replacement.
	Axial displacement of the shaft.	Adjustment and alignment of axis.
Strong vibrations of the pump.	The motor speed is too high.	Reduce the operating speed of the motor.
	Worn bearings.	Replace the bearings.
	Poor alignment- misalignment.	Align the pump.
Pump smells and smoke appears.	The pump operates without liquid (dry) for some time.	Immediate stop of the pump, disassemble and check for possible internal damages.
Pump absorbs excessive power (or absorbs more amperes than normal).	Shaft bending or misalignment.	Replace the shaft or align the pump.
	Rotation of the rotor obstructed.	Check the impeller, rotate the pump manually.
	Worn of wear rings.	Check rings and replace.
	Very tight gland packings.	Adjust the gland packings.
	High density or viscosity of the liquid.	Heat the liquid that is handled, reduce the rotation speed.
	High pump rotation speed.	Adjust the rotational speed.

CHAPTER TEN
Study of a pumping system

10.1 Introduction.

Reliability in ship operations, as in every production facility, is based on the circulation or transfer of liquids, which is supported by proportionally sized pumping systems. Also, the energy consumption for the operation of the pumps is one of the major parameters in the development of a plant. Therefore, correct design and correct operation are both important for the pumping systems. In the following paragraphs, some basic principles are summarized, that relate to the proper design of a pumping system, in which the pumps are a structural element.

The design as well as the functional approach of a pumping system requires enough preparation in relation to the objectives which need to be maintained by the system. Therefore, the following steps should be followed:

1) Determination of the conditions and main operational parameters.

2) Determination of direct and future needs that must be met by the pumping system.

3) Collection and analysis of all operating elements.

4) Evaluation of alternative projects and improvements.

5) Specification of technical and financial best practices, taking into consideration all other subsystems.

Then, the best alternative is applied, the technical specifications are modulated, the relevant equipment is procured, and, finally, the installation of the pumping system is carried out.

The control continues after installation, in order to optimize the system and to operate as efficiently as possible. This phase – although, to a significant degree, determined by the initial design options – is even more important, because a pumping system which is operated and maintained correctly has an estimated life span of 15 to 20 years.

A pumping system, as previously mentioned, consists of three main parts: The part comprising ***the suction***, the part comprising ***the discharge*** and ***the pump***. The discharge tube conveys the liquid to the end-use equipment (e.g. heat exchangers, pressure tank and generally to hydraulic equipment). Also, in several cases, the pumping system may be closed or open loop. ***Closed loop*** is when the liquid, after the discharge of the pump and the passage through machinery and equipment, is recirculated, since it returns to the suction pipe [e.g. heat exchangers in cooling water system ICE fig. 8.36 (p. 170) and fig. 10.1]. ***Open loop*** is when the liquid, after the discharge and the passage through the equipment of use, is discarded to the environment [e.g. heat exchangers with fresh water in the central cooling system of ICE fig. 8.35 (p. 169) and fig. 10.2].

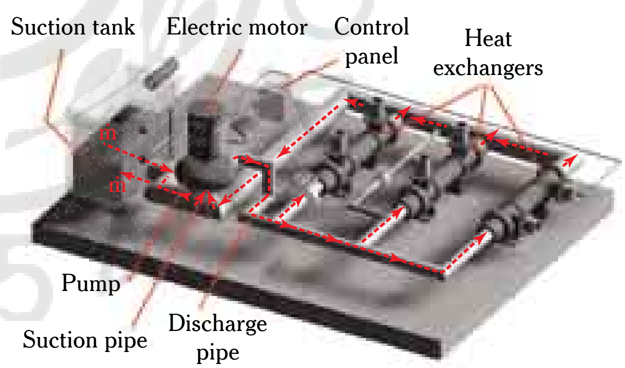

Fig. 10.1
Closed loop pumping.

According to what we saw in previous chapters, a dynamic pump is designed to operate at the best efficiency point (BEP). So, a pump which operates at BEP will have the greatest efficiency, will provide flow rate equal to the optimal V_n and will yield head H_n:

$$\text{BEP:} \quad \eta = \eta_{max} \quad \dot{V} = \dot{V}_n \quad H = H_n$$

Of course, the pump can operate with lower efficiency, and in other operating points, which are

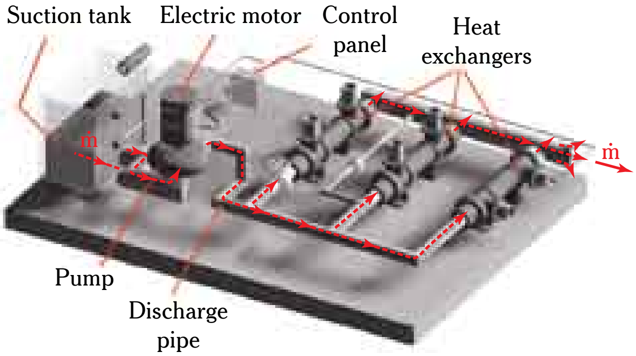

Fig. 10.2
Open loop pumping.

specified by different heads and flow rates, but always on the characteristic curve of the H-$\dot{V}$.

The operating point of the piping system where the pump is installed (or will be installed) is also the operating point of the pump and it depends not only on the pump but also on the pipeline. The intersection between the system and pump characteristic curves is the operating point. Therefore, given the pump and its curves, in order to find the optimal location for the operating point in the piping system, the pipeline and especially the most energy consumptive part of the pipeline – which is the discharge point – must be properly configured.

In order for the pump characteristic curves to be valid, and to achieve a functional operation of the piping system, the suction portion must be properly configurated to avoid cavitation phenomena or even pumping interruption.

According to the above, there are three major issues taken under consideration during the design of a pumping system: ***the proper pump selection, the design of suction and the design of discharge***. The designs of the three parts are highly interdependent. The proper pipe selection depends on a draft design of a suction and discharge. After pump selection (or use of the available one), follows the design of the other two parts and eventually the design of the entire pumping system.

10.2 Pumps selection criteria.

The selection of a proper pump is the most critical issue during the design of a pumping system. It should be clarified that the selection parameters of a pump are more than those that will be mentioned in the following paragraphs, and there are only few cases where selection is restricted to only one particular pump.

A basic prerequisite for the proper pump selection is the comprehensive knowledge of the pumping system where the pump will be installed. More precisely, the following criteria must be taken under consideration:

1) The ***properties of pumped liquid***. Particular attention must be given to the liquid ***viscosity***. If the pumped liquid is viscous, a positive displacement rotary pump will be used, instead of dynamic pumps. However, if it is possible to reduce the liquid viscosity by raising the temperature of migrant liquid (e.g. as in heavy fuel oils transfer), then a dynamic type pump could be used alternatively. Rotodynamic pumps perform poorly while pumping liquids of great viscosity and therefore significant power is needed. If kinematic viscosity is $>20 \cdot 10^{-6}$ m^2/s, the characteristic pumping curves are modified with correction factors.

The ***concentration of solid particles*** as well as their size and hardness is also an important issue. If the liquid has a great concentration of solid particles, a pump that will not clog or fail prematurely will be selected.

If the pumped liquid transfers ***large amount of gas***, pumps that can successfully respond will be selected, such as positive displacement rotary dynamic pumps or nozzle type pumps.

If the liquid has ***corrosive properties***, a special construction pump will be selected. The impeller of the pump as well as the parts that get in contact with water must be able to withstand the corrosion.

High attention must be paid when pumping ***high temperature*** liquids. Particularly, pump materials and specific parts such as the sealing system must be thoroughly examined, especially for pumping temperatures over 90°C.

Knowledge of ***operating pressure*** relates to proper materials selection, while vapor pressure in combination with the operating temperature relates to the Required Net Positive Suction Head of the pump. Finally, the consistency of the pumped liquid is directly related to the pump power.

2) ***System requirements in flow rate and total head***. This is the major criterion for proper pump selection. The optimal flow rate and the total head of the selected pump must meet the system requirements.

The relationship of optimum flow rate and the pump yielding head at a given speed rotation is expressed by the specific velocity of the pump. As highlighted in paragraph 5.5.2, the expression of specific speed may vary, depending on the manufacturer and the pumping system. Here, we exclusively refer to the ***Universal Specific Speed*** (Ω_S), which is dimensionless, and shown in eq. (5.28):

$$\Omega_S = 2\pi \cdot n \cdot \frac{V_n^{0,5}}{(g \cdot H_n)^{0,75}}$$

For conversions from or to expressions of the specific speed (N_S) that are not dimensionless (US, Metric, SI), table 5.2 (p. 96) is used.

It is reminded that axial flow pumps, which have high specific speeds, provide high flow rate and low head. As specific speed reduces, flow rate reduces too, but the provision of higher heads increases. Therefore, high flow rate efficiency decreases in mixed flow pumps and even more so in radial flow pumps, with a respective improvement of head yielding. If the total head is higher and the flow rate lower, steam turbine pumps as well as multi-stage centrifugal pumps are selected, without excluding the case of using static type rotary pumps (especially if the liquid pumped is viscous) or reciprocating pumps (for high heads and low flow rate). The diagram of figure 10.3 shows an overview of the use of positive displacement pumps or of dynamic type pumps in respect to heads and flow rates. The options are limited in area A (high head and low flow, positive displacement pumps) and in area D (low head and/or high flow rate, dynamic pumps). In (most common) area C (medium head, low or medium flow rate) a static or kinetic type pump can be selected. Finally, in area B (high head, low flow rate) the selection can be made between static type pumps and high rotation speed dynamic pumps.

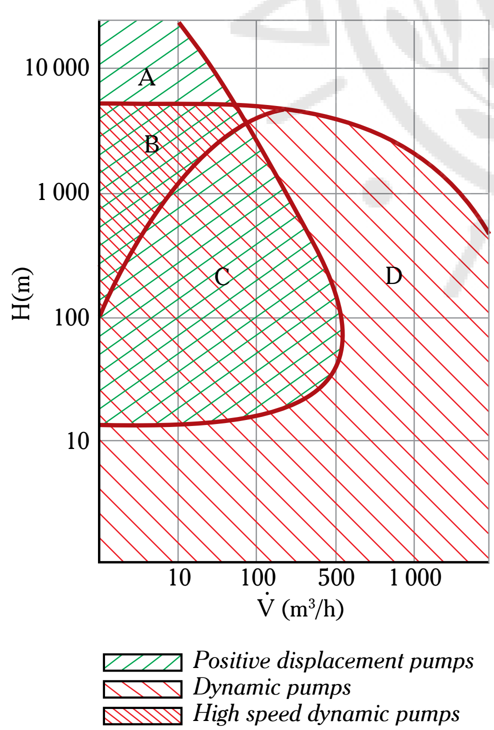

Fig. 10.3
Pump selection based on the total head and flow rate.

For the selection between different types of rotodynamic pumps, the special speed is utilized. To understand the relationship of height and flow rate with specific speed, the heads of five pumps with $\dot{V}_n$ = 100 m^3/h, n = 1 750 rpm and different specific speeds N_S are compared, as they result from the use of equation (5.28) (tab. 10.1).

Table 10.1: *Comparison of pumps with different specific speed.*

Ω_s	***Pump type***	**n** rpm	$\dot{V}_n$ m^3/h	H_n m
0.2	Radial flow	1 750	100	83.21
1.0	Radial flow	1 750	100	9.73
2.0	Mixed flow	1 750	100	3.86
4.0	Axial flow	1 750	100	1.53
5.0	Axial flow	1 750	100	1.14

A big difference in head yield is observed for pumps of different specific speeds.

To avoid the computation required by equation (5.28), the relevant nomogram of figure 10.4 is often used. In it, starting from the flow rate (ordinate axis), along the horizontal path, we go to the rotational speed (abscissa axis). Then, following the oblique blue line we reach the head (which is marked with oblique red lines in fig. 10.4). At the intersection point shall be the specific speed of the pump (on the right scale). For example, a pump having normal flow rate 100 l/s, operating at 1 500 rpm and delivering head at 20 m, has, according to the diagram, specific speed $N_S = 0.95$. [Applying the equation (5.28), it yields $\Omega_S = 0.947$].

Of course, given head and flow rate depend on the pump size, the impeller diameter and the counterclockwise rotation. But if the specific speed is given, the above-mentioned important attributes equally modify the given head and the flow rate. If, for example, the pump in table 10.1 with

$N_S = 2$ operates at rotation speed 1 600 rpm, the flow rate and the head will be lower:

$$\frac{\dot{V}_n}{\dot{V}'_n} = \frac{n}{n'} \Rightarrow \dot{V}_\eta{}' = \ 91{,}4 m^3 / h$$

$$\frac{H_n}{H'_n} = \left(\frac{n}{n'}\right)^2 \Rightarrow H_n{}' \ = 3{,}23 m$$

It can be noted that the knowledge of total head required by the pumping system, implies the knowledge of all the parameters of the pumping system that define it: static head, pressure and speed head, pipe characteristics. The discharge pump diameter is also required, since the total head includes the friction head of the system.

Pump manufacturers provide comprehensive charts (which refer to a group of similar pumps) for the selection of the proper pump in relation to the flow rate and the total discharge head of the system (par. 6.2). Usually, some technical notes are added on such diagrams. According to what was mentioned above, it will be useful if the special speed of each pump category was included. Such a diagram is presented in figure 10.5 (which includes not only the counterclockwise rotation, but also the nominal diameter of the discharge and the diameter of the impeller).

3) ***Net Positive Suction Head required*** (NPSHr). NPSH is also a significant parameter in the most common case where the relation of head – flow rate requires a dynamic pump. This parameter becomes of primary importance, especially when the design of the pumping system does not allow great values of available NPSH. For example, according to its normal flow rate and the respective given head, any pump could possibly be suitable for the system examined. However, if, according to the available NPSHa of the system and the required

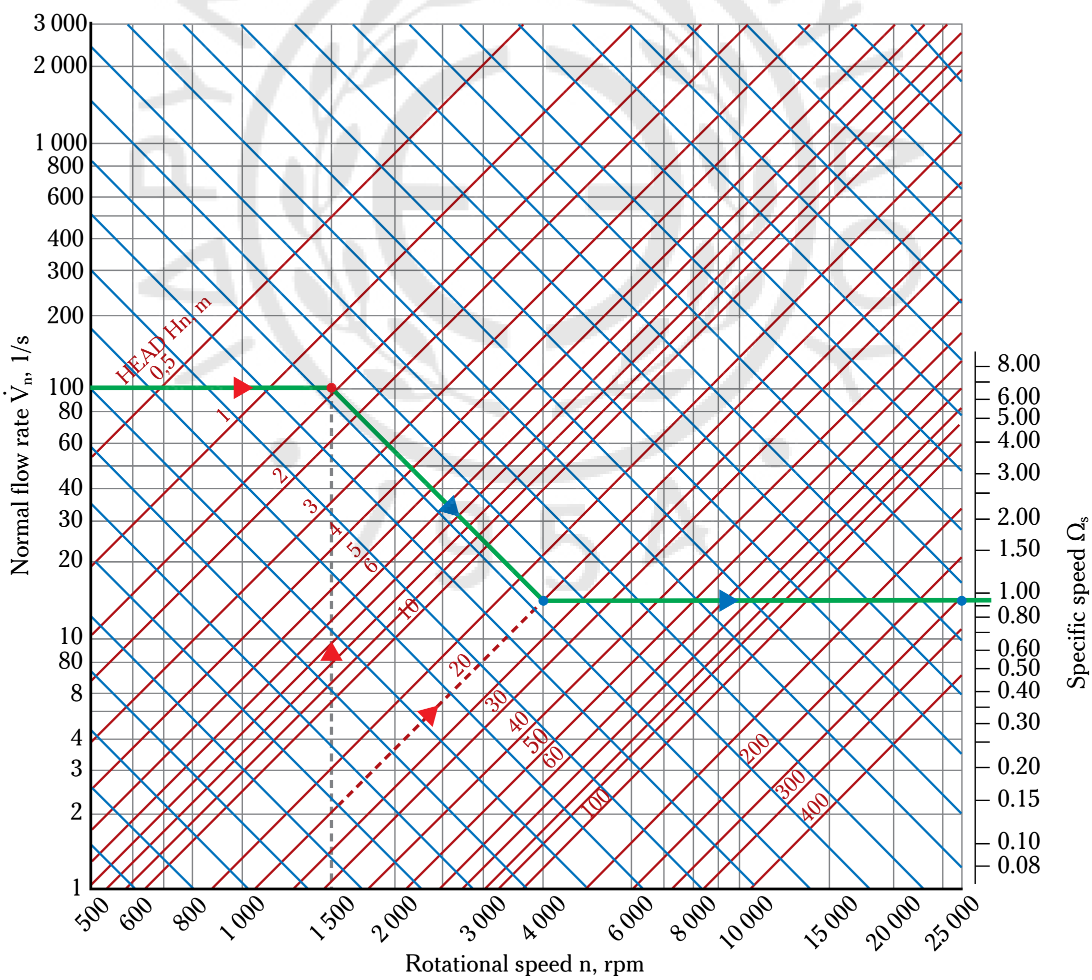

Fig. 10.4

Nomogram of calculating the Specific Speed.

NPSHr of the pump, the optimum operation point is also the cavitation point, the pump must be rejected (unless the available net positive suction head can be increased by modifying the suction design).

4) ***Pump efficiency***. Taking under consideration the aforementioned criteria (that relate to the operational requirements set by the pumping system) the range of the possible selected pumps has decreased significantly. Usually, however, there is a wide range of pumps available. At this point, as the technical requirements have already been covered, the question about the efficient operation of the system arises. The most suitable pump, among the pumps that meet the system requirements, is probably the pump with the highest efficiency.

In order to further clarify the criterion of cost efficient operation, another indicator is usually developed: ***the ratio of normal flow rate to the consumed power of the pump.*** The indicator presents the power consumption supply unit and allows the immediate comparison of the economic aspect of the efficiency.

Example 1.

A dynamic pump is to be mounted in a pumping system with requirements a total head of 50 m and water flow rate of 2 000 m³/h. It is estimated that the pump will operate 8 000 hours per year. If we have to select between two appropriate pumps, where the first one has efficiency of 82% and the other 78%, how much will the difference in the two pumps annual operating costs be? The energy costs is assumed equal to 0.05 €/kWh. The pumps are powered by a sufficient motor, with efficiency 96%.

Solution:

Data: $\dot{V} = 2\,000\ \text{m}^3/\text{h}$, $H = 50\ \text{m}$, $t = 8\,000\ \text{h}$, $\eta_1 = 0.82$, $\eta_2 = 0.78$, $\eta_{pm} = 0.96$, $c = 0.05$ €/kWh

Define the cost difference ΔC of two pumps.

The pump power yield is:

$$P_I = \gamma \cdot \dot{V} \cdot H = \rho \cdot g \cdot \dot{V} \cdot H$$

or $$P_I = 1\,000\ \text{kg/m}^3 \cdot 10\ \text{m/s}^2\ \frac{2\,000}{3\,600}\ \text{m}^3/\text{s} \cdot 50\ \text{m}$$

or $$P_I = 272.5\ \text{kW}$$

The shaft power of the two pumps would be differentiated so that:

$$P_{s1} = \frac{P_I}{\eta_1} = \frac{272.5}{0.82}\ \text{kW} = 332.32\ \text{kW and}$$

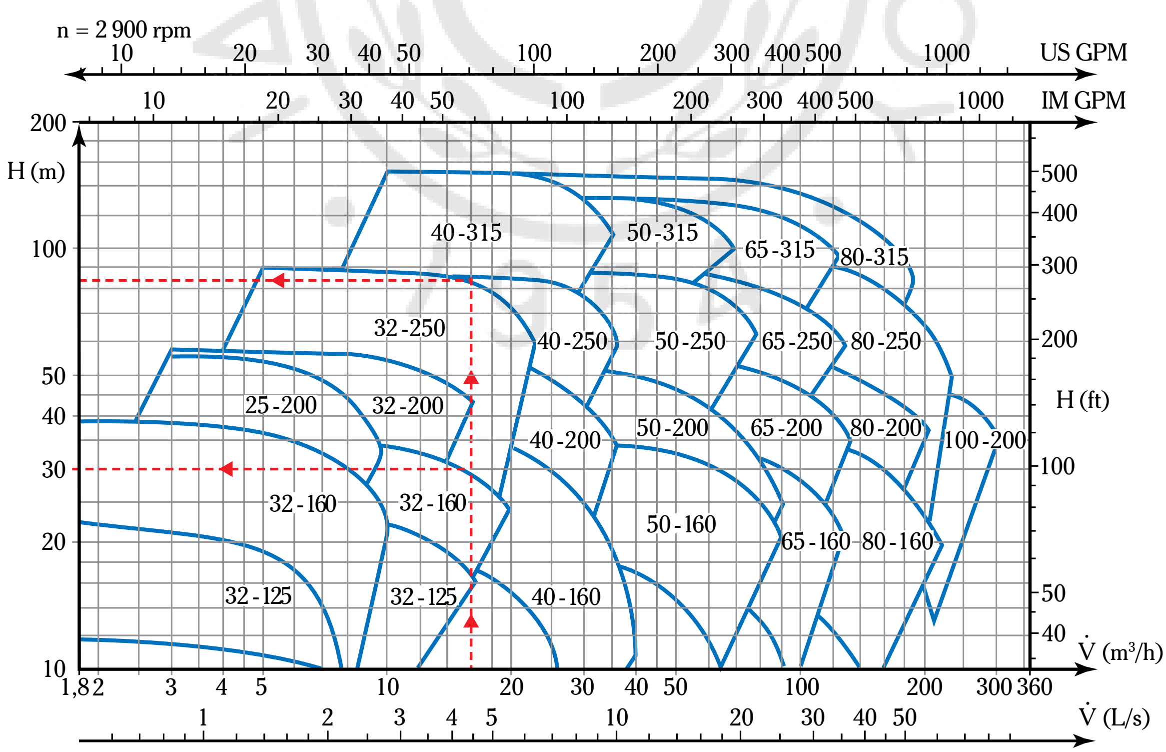

Fig. 10.5

Pump Selection Chart as a function of the provision and the total head of the system.

$$P_{s2} = \frac{P_I}{\eta_2} \; \frac{272.5}{0.78}\,\text{kW} = 349.36\,\text{kW}$$

The power consumption of the motor would be:

$$P_{m1} = \frac{P_{s1}}{\eta_{pm}} = \frac{332.32}{0.96}\,\text{kW} = 346.164\,\text{kW}$$

$$P_{m2} = \frac{P_{s2}}{\eta_{pm}} = \frac{349.36}{0.96}\,\text{kW} = 363.915\,\text{kW}$$

The yearly energy consumption of each pump would be:

$$E_1 = P_{m1} \cdot t = 346.164\,\text{kW} \cdot 8\,000\ \text{h/year} = \\ = 2\,769\,312\,\text{kWh/year}$$

$$E_2 = P_{m2} \cdot t = 365\,515\,\text{kW} \cdot 8\,000\ \text{h/year} = \\ = 2\,924\,120\,\text{kWh/year}$$

The energy difference for each year would be:

$$\Delta E = E_2 - E_1 = 2\,924\,120\,\text{kWh/year} - \\ - 2\,769\,312\,\text{kWh/year} = 154\,808\,\text{kWh/year}$$

The cost ΔC of difference on energy consumption ΔE would be:

$$\Delta C = \Delta E \cdot 0.05\,€/\text{kWh} = \\ = 154\,808\,\text{kWh/year} \cdot 0.05\,€/\text{kWh} = 7\,740{,}4\,€/\text{year}$$

The first pump is more economical.

The economic comparison of the energy consumption of the two pumps could be done with the configuration of the quotients:

$$V/P_1 = 5.77\,\text{m}^3/\text{kWh} = 1.60\,\text{m}^3/\text{J}$$

$$V/P_2 = 5.50\,\text{m}^3/\text{kWh} = 1.53\,\text{m}^3/\text{J}$$

Besides the above-mentioned criteria, there are more criteria that should be taken into account when choosing a pump and the pumping system design, such as the pump geometry (horizontal or vertical), its size, the compatibility with motors, the requirements for stability on the provision (or not) irrespective of load, the operability in a wide range of rotational speeds, the possibility of changing the impeller diameter, maintainability, etc.

Among these additional criteria, the specifications upon which the pump construction was based are considered as the most important. In this field, the International Organization for Standardization (ISO) has adopted some standards while some others are still being developed. In addition, in the US the ANSI / HI standards (American National Standards Institute – Hydraulic Institute) are valid, which are not yet sufficiently harmonized with ISO standards. This should make us very attentive, and we should bear in mind that the reliability of the standardization body constitutes a prerequisite for the provision of equipment.

– Pump Motor.

The pump motor does not affect the hydraulic calculations, but it provides power to the pump shaft, the major part of which is transferred to the pumped liquid (fig. 4.17, p. 79). The power delivered to the pump shaft is less than the power consumed by the engine (due to mechanical losses). The ratio of engine power to the axial force is the ***degree of engine efficiency*** η_{pm}:

$$\eta_{pm} = \frac{P_s}{P_m} \Rightarrow P_s = \eta_{pm} \cdot P_m \qquad (10.1)$$

The pumps that receive energy from a prime mover, which serves other energy purposes, are called attached. In these, the transmission of force to the pump shaft is made in different ways (belts, gears, etc.).

The independent pumps, in which an engine that works exclusively for them is adjusted, are more significant and offer more leeway in the selection process. The engine can be an internal combustion engine (e.g., a diesel engine). In most cases, however, the engine is an electric motor, typically an alternating current motor.

The speed rotation of a pump depends on the power transmission system. The coupling of the motor with the pump shaft entails restrictions on the available operating speeds. Particularly, alternating current motors operates at 50 or 60 Hz and, depending on the connection, they deliver speeds presented on table 10.2.

Table 10.2: *Reference speeds of three-phase motor.*

Number of poles	2	4	6	8	10	12	14
Frequency 50 Hz (rpm)	2 900	1 450	960	725	580	480	415
Frequency 60 Hz (rpm)	3 500	1 750	1 160	875	700	580	500

Source: International Electrotechnical Commission – IEC.

In practice, the engines rotate at more speed (depending on the power output and construction). In case of using ***motor speed controllers***, operation is also possible on other speed levels. The motion transmission belts, gearboxes or the use of turbines or ICE as drivers, allow the desired adjustment of the rotation speed.

It is obvious that choosing an independent pump requires the selection of the respective engine. In some cases, the use of a pump also defines the type of engine. Thus, a fire pump should not depend on electric power itself (or at least it should be able to operate without it). Therefore, the choice of an ICE with suitable power as the driving machine of the pump is obvious.

The selected engine to run a pump must have an additional safety power reserve and meet the power requirements of the pumping system. Major factors that affect the engine selection are the initial cost and the reliable operation. Electric motors have lower initial cost. The higher cost of variable speed motors can be recouped through the significant energy savings, especially when the pump operates in long periods and treats variable loads. There is often a large amount of steam available which justifies the additional cost of a steam turbine (although, in this case, the maintenance cost increases).

An important issue related to the efficient operation of centrifugal pumps, but also to the choice of a suitable engine, is the ***flow rate control***, especially when the pump is experiencing variable loads. In order for the pump to meet the variable needs, we must, ***firstly***, employ a bypass valve system, or, ***secondly***, modify the characteristic curve of the system or of the pump, in order to move the system back to the desired operating point.

In the ***first case***, that of using a bypass valve system, part of the pump flow rate from the discharge to the suction is recycled. Flow rate control is good but rather expensive, due to the energy consumed for pumping the liquid which is recirculated.

In the ***second case***, ***throttle valves*** are used to displace the pump curve or the system curve. Even if the flow rate control is satisfying, the energy loss is significant (since the system operating point is away from its Best Efficiency Point).

The displacement of the characteristic pump curve and, consequently, the system operating point, is the indicated solution (the operating point is still in the optimal position). However, it has high initial cost since ***multiple speed motors*** or ***adjustable-speed drivers (ASD)*** must be used.

The aforementioned are further clarified in figure 10.6. Suppose that the system, equipped with a centrifugal pump, operates at point A with a flow rate $\dot{V}_A$ and consumes power P_A. There are three options to reduce the flow rate from $\dot{V}_A$ to $\dot{V}_B$:

1) A bypass valve system is installed and recycles the excess flow rate $\dot{V}_A - \dot{V}_B$ from the pump discharge to the suction tank. The pump and system curves, and therefore the operating point, do not change. The power consumption remains equal to P_A.

2) The characteristic curve of the system is displaced from position 1 to position 2 (increasing local losses by throttling the discharge valve). The operating point slides to point C (lower efficiency level). It can be noted that the power P_C is lower than the initial P_A. For this reason, the use of this method has an advantage over the recycling solution. Yet, the operation of the pump in long time periods away from the Best Efficiency Point (BEP), usually entails significant risks.

3) The third option includes shifting the characteristic curve of the pump $H-\dot{V}$ downwards, reducing the rotational speed from n_1 to n_2. The operating point moves from A to B. The pump continues to

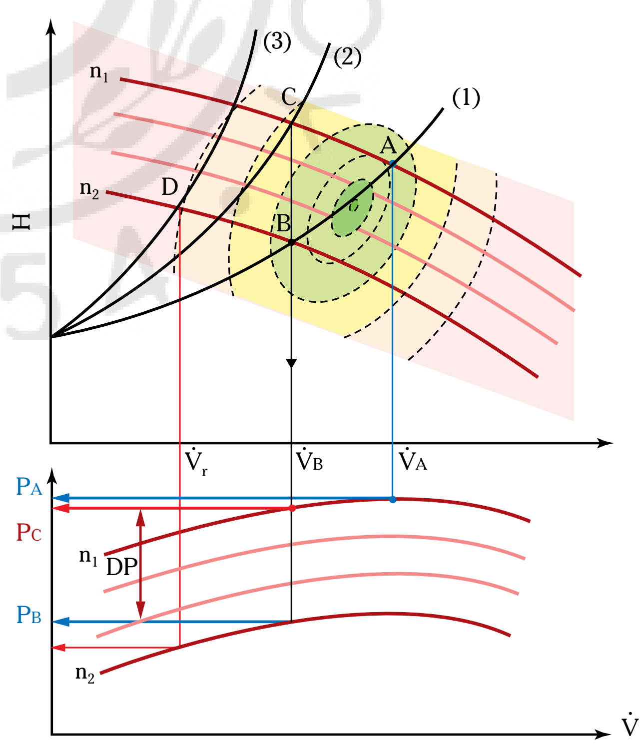

Fig. 10.6
Control of pumping system flow.

operate in the optimum efficiency area. The power consumption is now equal to P_B, significantly less than the original P_A and the P_C. The energy benefit is particularly important and, from a techno-economic point of view, it is the appropriate solution, despite the higher initial cost. To further reduce the flow, one of the previous solutions can be used, without a substantial energy cost, since now the consumed power is not very high.

10.3 Design of suction.

As mentioned previously, in order for pump operation to be unhindered (and thus for the pumping system to run smoothly as well), and in order for the characteristic curves to be valid, the creation of cavitation conditions should be avoided. This means that during the design of suction the greatest possible net positive suction head available (NPSHa) must be ensured, or, more accurately, the difference of M must be ensured between the NPSH available from the system and the NPSH required by the pump (where M is called the margin net positive suction height):

$$M = NPSHa - NPSHr$$

Generally, it is recommended that the difference be more than 25% of the required NPSHr of the pump.

In accordance with equations (6.14), (6.14a) and (6.14b) the NPSHa is increased when: ***firstly***, the static suction head H_{sts} is reduced, and, ***secondly***, the amount of losses in the suction pipe is reduced.

The reduction of static head H_{sts} at the suction is achieved by placing the pump in a place where there is the smaller possible difference of height from the free surface of the suction tank, and, if it is possible, lower than that (so that the suction static head is negative). The positioning of pumps lower than the suction tank, not only helps prevent cavitation, but also solves the problem of filling the dynamic pumps which are not self-priming.

Reduction of losses in the suction pipe and obtaining the system functionality is achieved by the following measures:

1) ***Reduction of the suction tube length to the minimum necessary***. Although the length of the suction pipe is generally small, it should be considered that a doubling of the pipe length means a doubling of the linear losses. However, attention must be paid in order not to create other problems. One of them regards the location of the pipe suction point. Thus, if the input of the suction pipe is located near the free surface of the liquid, during the pump suction intense turbulence and hollow vortex is generated (fig. 10.7). This can possibly result in the pump sucking in air, with all the unpleasant consequences that are examined in paragraph 5.7 (p. 104). This phenomenon is avoided if the difference between the pipe suction point level and the surface of the tank is satisfactory. According to the Hydraulic Institute, the minimum height of immersion (S_{min}) in an open tank is calculated by:

$$S_{min} = d_{in} + 2.3 \cdot v_{in} \cdot \sqrt{\frac{d_{in}}{g}} \qquad (10.2)$$

where: d_E the inlet diameter of the liquid in the suction tube (in m) and v_E the input speed (in m/s). In combination with the minimum height of immersion, correct distances must be kept between the suction tube opening of the walls (usually> 2.5 d_{in}) and the bottom of the tank.

Example 2.

In an open tank, estimate the minimum suction immersion height of a tube, if the flow rate is 180 m^3/h and the diameter of the inlet to the suction tube is 22 cm.

Solution:

Data: $\dot{V} = 180\, m^3/h = 0.05\, m^3/s$, $d_{in} = 0.22\, m$

Required: S_{min}

We calculate the input speed:

$$v_{in} = \frac{4 \cdot \dot{V}}{\pi \cdot d_{in}^2} \Rightarrow v_{in} = 1.32\, m/s$$

Fig. 10.7
Vortex formation at the pump suction tank.

then by equation (10.2) is given: $S_{min} = 0.67$ m.

2) ***Increasing the diameter of the suction pipe.*** The short length of the suction pipe does not make it particularly costly to place a pipe with increased diameter (which is more expensive), and leads to a significant drop in losses. The basic criterion for the selection of the appropriate diameter is the desired speed within the suction pipe. This should be low in order to minimize losses, but sufficient so that it can entrain the suspended solid particles from the liquid stream and prevent segregation and sedimentation. Although each system has potential variations, the recommended standard suction speeds are as follows:

a) for water: 1.0 – 1.5 m/s,
b) for light oils: 1.0 m/s,
c) for viscous liquid: 0.5 m/s
d) for hot liquids: 0.6 m/s.

In case the pumped fluid contains a large amount of suspended solids, the recommended minimum speed is 1.0 m/s.

It should be noted that, if the pipe diameter is greater than the diameter of the pump suction flange, a contraction pipe fitting[1] is required. The gradual contraction should be as small as possible in angle and eccentric, in order to reduce the flow turbulence at the liquid import and avoid the creation of air cavities in the pump inlet.

3) ***Minimizing local losses in the suction pipe.*** This is achieved by avoiding the unnecessary components in the installation (e.g., more valves). If the installation of contraction or diversion parts is needed (e.g., bends), the best is to have large curvature radius.

However, there are inevitable local losses by ***fittings*** which are needed for proper function of the suction, and these cause increase of losses.

First there is the entrance of the liquid from the tank to the suction pipe. There, the local losses coefficient K, depending on the configuration of inlet, ranges from 0.02 to 0.8. Seeking the least possible losses, often the entrance construction is ***bell-shaped***.

If it is required to prevent the introduction of solid particles in the tube inlet, special ***filters*** are placed there.

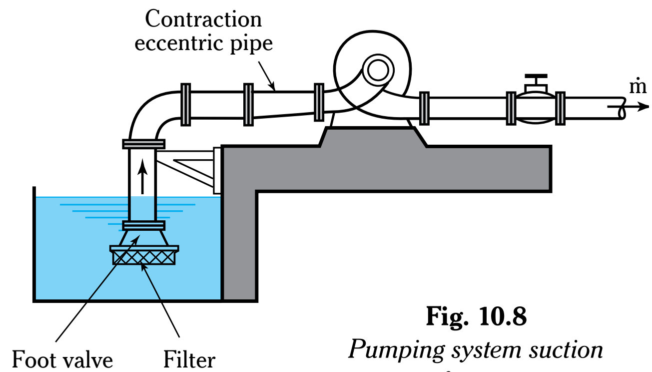

Fig. 10.8
Pumping system suction configuration.

If a dynamic pump is located higher than the suction tank and does not have a priming system, a ***foot valve*** (or a check valve) is mounted to prevent the emptying of the suction tube. The local losses coefficient of the filter and the foot valve is sufficiently large, ranging from K = 2 to K = 6.

In figure 10.8 pumping system is shown with suction filter, foot valve, and contraction eccentric pipe.

4) ***Ensuring tightness.*** Apart from the actions that increase the NPSHa to prevent cavitation by the liquid vaporization, it is very important to avoid cavitation from gasses. It is reminded that on the suction pipe the pressure drop is significant. Particularly if the suction tank is under atmospheric pressure and located lower than the pump, then the pressure in the suction pipe is significantly less than atmospheric pressure. Therefore, the lack of complete sealing will result in air suction with negative impact on pump performance, even the loss of pumping capability. For this reason, before the first time a pumping system operates, it is subjected to hydraulic pressure test of the suction line to ascertain the sealing.

10.4 Design – Calculation of discharge.

Another important aspect of the design relates to studying the energy efficiency of the pumping system. Some basic parameters are already given by the needs the system is called to serve. Therefore, the static head and the pressure head that has to be covered by the pump are usually given. The distance at which the system transfers the liquid is also approximately known. Therefore, the length

[1] A ***contraction pipe fitting*** or ***eccentric reducer*** is a piece of tube which is constructed so as to have a gradual decrease of its diameter.

of the pipeline has a minimum value, which will likely be exceeded when planning the installation. Finally, the migrant liquid and the handling conditions are known (so, the properties of the liquid are also known, such as temperature, pressure levels in which it will be found, the viscosity, corrosivity, etc.). Knowing the transport conditions allows ***the selection of appropriate pipes construction materials and components***.

The sum of the static head, the pressure head, the velocity head and the friction head is the ***total head*** a pump has to cover. The first three heads define the energy needs of the system; it is beneficial energy which is assigned to the liquid through the pump. Knowing the needs a pumping system has to cover, those three heads can be configured. For example, if water transfer in a space with differential pressure 5 bar is needed, the pressure head is:

$$H_p = \frac{\Delta_p}{\gamma} = \frac{\Delta_\rho}{\rho \cdot \gamma} = \frac{5 \cdot 10^5 \text{Pa}}{1000 \,\text{kg/m}^3 \cdot 10 \,\text{m/s}^2} = 50 \text{ m}$$

where $g = 10\,\text{m/s}^2$.

If the altitude difference is 15 m, the static head is the same as well. If the water should exit from a nozzle at a speed of 20 m / s, the velocity head is:

$$H_V = \frac{v^2}{2g} = \frac{20^2}{2 \cdot 10} = \frac{\text{m}^2/\text{s}^2}{\text{m/s}^2} = 20 \text{ m}$$

In contrast, the head losses (linear and local energy losses due to friction), depend on the flow rate and configuration of the pipeline. Considering that the suction pipe is short in length and has relatively large diameter, there are according to the loss equation, low losses, and hence ***the head losses are primarily due to the discharge pipe***. Therefore, the inclination of the characteristic curve of the pipeline depends on the choices made in respect to this.

The objective is for the operating point of the system (that is, the intersection of the characteristic curve of the pipeline system with the characteristic curve of the pump), firstly, to be at ***the optimum operating range of the pump***, and, secondly, for ***the cavitation area to be avoided*** under any circumstances. When the suction is designed properly, the cavitation region will be outside the optimum operating range. ***Therefore, for the design and selection of the discharge pipe, the basic data are the optimum supply V_n and the corresponding head H_n.*** As is expected, attention must

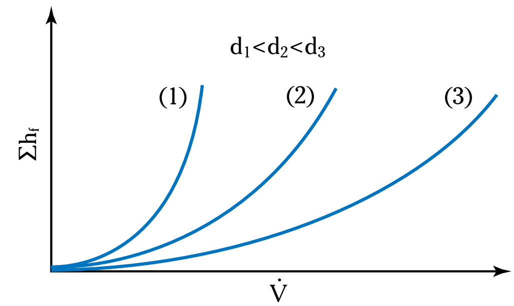

Fig. 10.9
Curves $\Sigma h_f - \dot{V}$ of various discharge diameters.

be given to minimize the local losses, by avoiding the use of fittings that are not essential. Also, the length of the discharge tube should be as small as possible, avoiding unnecessary paths.

However, the most important parameter is the diameter of the discharge tube. As shown in figure 10.9, a small discharge pipe diameter means an abrupt slope in the head loss flow rate curve (and hence of the pipeline), since, as the diameter grows, so the slope decreases.

One of the most important tasks when designing a pumping system, is the selection of the appropriate diameter. The calculation is shown in the following example.

Example 3.

A water pumping system has static head 15 m, and suction and discharge tanks under atmospheric pressure. A pump with characteristic curves shown in figure 10.10 will be installed to the system. The value of suction losses at normal pump flow is 3 m and the total coefficient of local losses in discharge equals to 12. The discharge pipe is made of cast iron and 150 m in length. Calculate the diameter of the discharge pipe (in mm), so that the pumping system can operate with maximum efficiency.

Solution:

Data: $H_{st} = 15$ m, $h_f = 3$ m, the $H-\dot{V}$ curve of figure 10.10, $L_d = 150$ m, $\varepsilon_d = 0.00026$ m, $K_d = 12$ water kinematic viscosity $\nu = 10^{-6}\,\text{m}^2/\text{s}$.

Requested is the optimum diameter dκ.

Step 1: From the curve H-V of figure 10.10, we take the optimal flow rate and the corresponding Head (for impeller diameter of 335 mm):

$$\dot{V}_n \approx 180\,\text{m}^3/\text{h} = 0.05\,\text{m}^3/\text{s},\ H_n \approx 30\,\text{m} = H$$

For these values the diameter of the discharge pipe will be calculated.

Step 2: The energy equation of the pumping system gives:

$$H = H_{st} + \frac{p_2 - p_1}{\gamma} + \frac{v_2^2 - v_1^2}{2g} + \Sigma h_f \Rightarrow$$

for $p_1 = p_2 \Rightarrow p_2 - p_1 = 0$ and $v_2 = v_1 \Rightarrow v_2 - v_1 = 0$ resulting that:

$$H = H_{st} + \Sigma h_f \Rightarrow \Sigma h_f = H - H_{st} = (30 - 15)\ \text{m} = 15\ \text{m}$$

$$\text{and} \quad h_{s1f} + h_{d1f} = \Sigma h_f \Rightarrow h_{d1f} = \Sigma h_f - h_{s1f} \Rightarrow$$

$$\Rightarrow h_{d1f} = (15 - 3)\ \text{m} = 12\ \text{m}$$

Step 3: A typical problem of calculating the diameter of a pipe must be overcome. According to the losses equation:

$$h_d = \frac{8}{\pi^2 g}\left(f \cdot \frac{L}{d_d} + \Sigma K_d\right) \cdot \frac{1}{d_d^2} \cdot \dot{V}_n^2$$

In this equation unknown are the friction coefficient and the diameter. This is solved by trials (for each value of diameter d is obtained a corresponding value of f′ from Moody's diagram). Thus a table is formed, where v′ is the speed, ε is the roughness of the pipe, d′ is the diameter of tests, Re the Reynolds number, h′ the calculated losses, h losses that result from Step 2. By consecutive tests, the desired diameter is determined (the total Head losses h_{df} is criterion for corrections) (tab. 10.3).

Therefore, the discharge pipe that will be select-

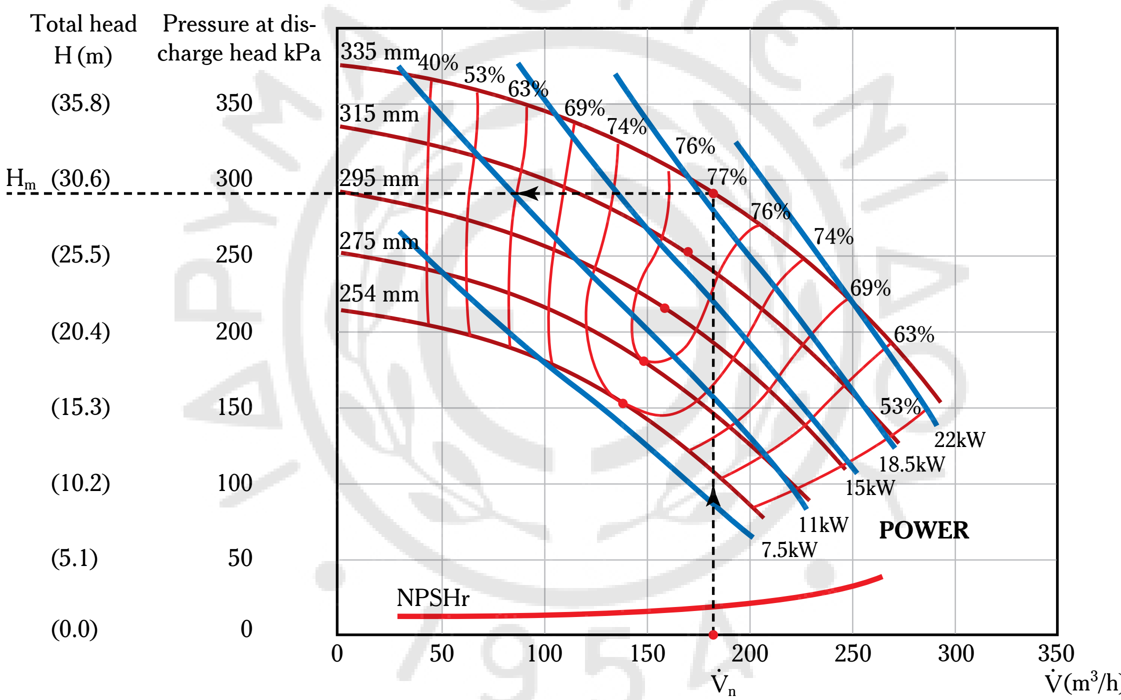

Fig. 10.10
Centrifugal pump characteristic curves.

Table 10.3: *Discharge pipe diameter calculation by the test method.*
(h_{df} = 12 m from step 2 calculation)

d′ (mm)	**v′** (m/s)	**ε/d′**	**Re′**	**f′**	**h'_{df}** (m)	**$h'_{df} - h_{df}$**	**d** (mm)
100	6.37	0.00260	$6.37 \cdot 10^5$	0.0254	103.5	91.5	d > 100
200	1.59	0.00130	$3.18 \cdot 10^5$	0.0217	3.65	−8.35	d < 200
150	2.83	0.00173	$4.24 \cdot 10^5$	0.0230	14.29	2.29	d > 150
160	2.49	0.00163	$3.98 \cdot 10^5$	0.0227	10.49	−1.51	d < 160
155	2.65	0.00168	$4.11 \cdot 10^5$	0.0228	12.20	0.20	**d ≈ 155**

ed will have an inside diameter of $d_d \cong 155\,mm$.

In the process of calculating the optimal diameter, thorough calculation is not needed, for three reasons:

1) The curves data are approximate.

2) It is sufficient for the system operating point to be near the BEP (Best Efficiency Point) and

3) the selected diameter depends on the existing standardized inner diameters (from those commercially available).

However, any carelessness in the calculations is not acceptable, because the cost of discharge pipes and their fittings is very important, and may reach up to 40% of the investment. Also this is the usual reason why, in pumping systems, the best energy-efficient diameters of discharge pipes are not used, but a little smaller (and thus cheaper). Namely, ***the optimal techno-economic diameter is selected***.

Specifying the calculation process for choosing the economically and technically optimal diameter, several important ***factors*** must be taken into account, including:

1) The ***purchase cost*** (pipes, connections, components, electrical equipment and automations).

2) The ***installation cost***.

3) The ***operating costs*** (energy consumption).

4) The ***maintenance costs***.

5) The ***lifetime estimation*** by the manufacturers.

The purchase cost is increased roughly in line with the diameter, small changes are presented in the installation costs, while operating costs are increased significantly by reducing the diameter (presenting low values to the field of optimum pump operation). If the above cost parameters are plotted on a graph (the two first aggregated), with ordinate the diameter, and summed, the total cost curve that is obtained is shown in figure 10.11, where it is taken into account that factors 4 and 5 are the same. At this graph, the techno-economic field of optimal diameter is identified. (Knowing the market prices in a given period, the corresponding curve is specified and modified).

The problem of optimal techno-economic diameter selection becomes more complex if we add the parameter of the tube construction material and the maintenance costs. Generally, the diameter is chosen so that the velocity of the liquid in the discharge pipe in normal operation ranges between 1.4 and 3.w5 m/s.

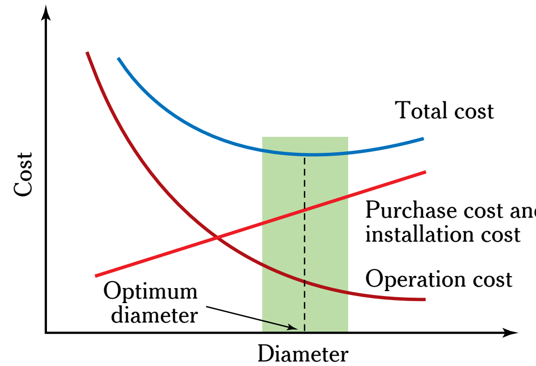

Fig. 10.11

Optimum discharge pipe diameter calculation.

It is emphasized that the above computation does not relate to the diameter of the suction pipe, which is defined solely by operational factors (see par. 10.3).

Although the calculation of the pipe diameter is very important, the design of the discharge pipeline does not depend only on this. The pipeline should be designed depending on the space and the process that it serves within the pumping system, and various fittings and piping supports must be placed in it.

10.5 Life cycle cost analysis.

Apart from the running cost, the financial dimension entails the ***initial pump purchase cost*** and the ***maintenance cost*** as well as any support for the product (and, thus, its ***reliability***). Thus, if the more efficient pump of Example 1 is more expensive by €35,000 than the other, and both feature the same degree of reliability, the amortisation of the initial cost difference will be about 5 years. With an operating life expectancy of 15 years, the first pump should be selected.

The above observations constitute the factors that are imported into a feasibility study on design and equipment provision. Such study is now based on an analysis of costs for the life cycle of the equipment or a part of it. Internationally it is known as a ***Life Cycle Cost (LCC) Analysis***. The European Pump Manufacturers Association (Europump) and the Hydraulic Institute have a qualified edition specializing in pumps (“Pump Life Cycle Costs”), which can be obtained by interested parties. The entire process is based on the operational life time of the equipment, the costing of the initial purchase, installation, operation and maintenance for the life expectancy. The complete equation of LCC is defined as:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_s + (C_{env} + C_d)$$

where: C stands for the cost and the indicators identify the individual costs for the equipment life cycle. In particular:

1) ***ic*** (initial cost): ***Purchase price*** (pump, system, pipe, etc.).

2) ***in*** (installation): ***Cost of installation and initial operation.***

3) ***e*** (energy): ***Energy costs*** (for the entire life cycle of the pump).

4) ***o*** (operating): ***Operating costs*** (normal surveillance labor costs).

5) ***m*** (maintenance): ***Cost of pump maintenance.***

6) ***s*** (loss of production): ***Loss of production - down time***.

7) ***env*** (environmental): ***Environmental costs***.

8) ***d*** (dismantling): ***Cost of Decommissioning***.

For pumps and pumping systems in general, the major costs are the energy consumption costs, the operating and maintenance costs, and the initial cost. The ratio is expressed (in general) in figure 10.12. Therefore, from a techno-economic point of view, the potential choice of a pump that is based on direct costs only, i.e., the purchase price, is wrong.

Such analysis, when used as a tool for the comparison of (technically adequate) alternatives, leads to the selection of the most effective choice (at least with the available data). Also, this analysis can and should be used as ***a continuous performance optimization tool*** for installed pumping systems, in relation to the energy consumption limitations, the maintenance activities, etc. In order to have the system under constant control, and as efficient as possible, the following are proposed:

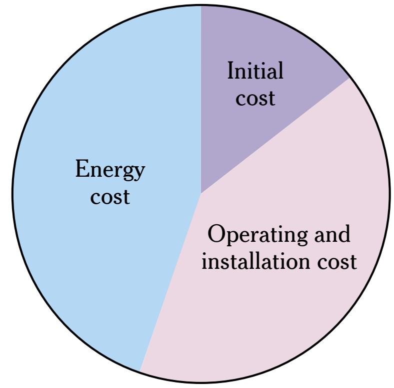

Fig. 10.12
The most significant life cycle costs in a pumping system.

1) A complete record of the pumping system components must exist (e.g., tanks, pipes, pumps, valves, elements of speed control system or of flow direction, etc.). The file should include standard information such as the component name or part number, the manufacturer, model, size, etc.

2) The performance curves of pumps, as well as the design characteristics curves of pipelines shall be maintained, and the operating points and flow velocities for various loads encountered by the system must be recorded.

3) Identification of components that cause increase in losses of performance for possible future (or direct) replacement (e.g. a ball valve which has very high loss coefficient and its installation in the pumping system might not be necessary).

4) Identification of pumps operated with low yield. A conservative or extemporaneous design of the system often leads to the use of oversized pumps, which waste power. The reduction of the impeller diameter or the future replacement of the pump with a smaller one, is perhaps the right action, and

5) identification of pumps with high maintenance costs. High maintenance costs may indicate that perhaps the pump is not suitable for the pumping system. A remedy study may solve the problem; otherwise the pump needs to be replaced.

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Working model of a Ctesibius piston pump.
Ctesibius (285-222 BC) was a Greek mathematician, inventor and founder of the School of Engineering and Mathematics in Alexandria.

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