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HYDRAULICS
IN INDUSTRIAL AND MOBILE APPLICATIONS

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ASSOFLUID

HYDRAULICS
IN INDUSTRIAL AND MOBILE APPLICATIONS

Italian Association of Manufacturing and Trading Companies in Fluid Power Equipment and Components

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We also would like to thank all these Companies, cited at the end of this manual, that contributed significantly to the making of this book by offering their data, technical drawings, technological innovation, pictures of various devices and operating principles.

Lastly, we would like to thank Mr. Carlo Maria Rozzi De Hieronymis and Mr. Mike Rowley: they provided us an invaluable help in revising the final draft of the text.

FOREWORD

The continuous evolution of technology, the integration of electronic system and the constant performance improvements that oil hydraulic components have been undergoing in the last year, from the point of view of safety and reliability, made it necessary to carry out a thorough revision of the previous ASSOFLUID Oil Hydraulics Manual. The text and drawings in the manual have been significantly enhanced and modified, as well as its layout.

The main purpose of this new manual is to help readers to approach oil hydraulics, by offering a wide overview of this sector's components and systems whilst offering also useful in-depth treatments of some of the countless industrial and automotive applications of this subject.

Because of its purpose, this Manual's target includes, in addition to training and educational sectors and oil hydraulic components and system manufacturing companies, all people dealing with this sector, which are provided with a useful window on the many need and purposes Oil Hydraulics can fulfil and meet today.

ASSOFLUID wants to carry on in its goal of supporting and developing professional and technical training for this sector, as stated in the Association's Charter, providing everybody with a fully up-to-date tool, both from the technical and legislation points of view.

Vincenzo Caprari
ASSOFLUID President

Milan, September 2007

INTRODUCTION

Being closely linked to a mechanical or electric generation, **FLUID POWER** is one of the most important technologies employed in the manufacturing and service industries.

Its applications in the different industries are so numerous that a number of pages would be required to list all of them and this list would have to be continuously updated to keep up with world innovations.

Our reference point to understand fluid power concepts is the ISO 5598 standard that defines it as *'means whereby signals and energy can be transmitted, controlled and distributed using a pressurized fluid as the medium'*.

The hydraulic generator – commonly referred to as 'pump' – operated by an electric or endothermic motor transfers energy to the fluid, which is channelled into pipes by means of special valves, controlled and conveyed to oil hydraulic motors, linear and rotary actuators.

The key element of this process is pressure, a very simple physical principle, which by the way is often improperly used in everyday language. Pressure is defined as the force per unit of area applied on a surface, in other words a weight, a spring, a fluid, a mass pressing on a surface that is able to sustain this stress.

Pressure develops when you press a finger on a wall, put a book on a desk, sit in an armchair and so on. However, these actions are purely static and the resulting pressure affects only the dimensioning of the supporting material. A typical construction example in this sense is the floor of a room: structural engineers take into consideration the overall furniture weight, the maximum number of people that can enter the room and the concrete slab mass itself in order to calculate how much concrete is required to sustain the resulting pressure.

This concept must be applied in a dynamic manner for fluid power, which means the pressure transmitted to a load must allow it to translate or deform. The problem is no longer the resistance of the supporting surface to the force applied on it, but a force that can outweigh the force the load opposes.

This principle can easily be understood with the example of the manual transfer of a heavy object: the object moves only when the manual force outweighs the force the object opposes. Translation in fluid power occurs by replacing manual force with an actuator operated by a pressurised fluid.

A relief valve regulates the ensuing pressure, which should be higher than required due to physical reasons depending mainly on pressure drops; a flow control valve and another valve regulate respectively fluid speed (which affects load translation speed) and the actuator direction.

Any fluid can be used for these purposes, while circumstances determine the type of fluid. By fluid we mean also a gas whose distribution in a container is uniform, like a liquid. The main difference is that liquids are almost incompressible whereas the volume of all pressurised gases reduces and they are consequently compressible.

Fluid power is subdivided into two independent technologies: **pneumatics**, that is the use of compressed air for light applications or complex automations, and **oil hydraulics,** where the fluid is an almost incompressible liquid suitable for heavy fixed or moving applications. Oil hydraulics is also known as **hydrodynamics, hydrostatic transmissions** (this can be mistaken for the specific 'hydrostatic drive') and **fluid engineering** (used also in pneumatics); 'oil' refers to the fluid used in most applications. 'Oil hydraulic' is often replaced simply by **hydraulic** both because oil is not contained in all fluids and to simplify definitions like hydraulic systems, hydraulic pumps, etc.

These two techniques often complement one another. In pneumatics there is a widespread use of pneumohydraulic actuators like pressure multipliers and speed hydraulic controls, while sometimes in oil hydraulics it is worth controlling directional valves by means of compressed air and accumulators demand nitrogen at an adequate pressure. Still, both technologies are found in some configurations where they are independent from each other but they are operated by a single central unit, like in some industrial presses or farm machinery.

Each of these two technologies, albeit structurally similar, concerns so many issues that it is impossible to cover all of them in a single volume. As a result, references to pneumatic transmissions will not be commented on in this work.

Brief history

Ancient scholars already sought the link between the force resulting from the movement of a fluid and the ensuing movements, the relations between volume and speed and pressure drops in pipe bottlenecks.

In his work entitled ***Pneumatica***, the Greek Hero of Alexandria (about 2nd – 1st century BC), a pupil of Ctesibius who invented the first compressed air musical instrument, described how some 'machines' work, for instance fountains, robots, siphons, water or air clocks that were pressurised with primitive manual pumps.

The Roman engineer Vitruvius (1st century BC) elaborates on Hero's ideas and presents the fundamental principles on which the main modern age physicists focused in the seventh book (on hydraulics) of his treatise ***De architectura***.

There are practically no or few significant scientific breakthroughs in the 1500 years following Julius Caesar's age because men in the post-Roman Empire, Charlemagne's Holy Roman Empire, the 'fatal' year 1000 and during Humanism were too busy discussing philosophical and religious problems, so little attention was paid to technological issues.

This situation totally changed during the Renaissance. Men opened their eyes, noticed the marvels surrounding them and wanted to understand their meaning. Historical events went hand in hand with new scientific ideas. It was the age of Lorenzo the Magnificent, Michelangelo, Giordano Bruno, Erasmus, Machiavelli, Palestrina, Gian Galeazzo Sforza and Ludovico il Moro, Leo X, Martin Luther, the Council of Trent, Charles V, the 'Rey Prudente', Richelieu, Olivares, and Shakespeare. It was the splendid epoch of Gutenberg, Leonardo, Galileo, Kepler; the scientific community flourished, humanistic and philosophical ideas went in parallel with science since the latter did not make sense without the former and vice versa.

That was also the age of Colombus, Vespucci, those 'Conquistadores' and settlers who created the Americas technology owes so much.

The first hydrodynamic applications were created, albeit inspired by the Roman civilisation, especially monumental fountains. For instance, the monumental fountains of the marvellous villa d'Este in Tivoli were based on the principle of communicating vessels. A 'hydraulic organ' (17^{th} century) designed by Claudio Vernard could be found in the same villa: water entering a closed cave pressurised the air that was then driven through organ pipes; a pressurised water jet made a giant toothed cylinder rotate, which activated the organ keys.

The invention of the steam engine in the late 18^{th} century marked the beginning of the phenomenon known as 'The Industrial revolution'. While before its advent energy could be obtained only from human or animal force, windmills or watermills, steam engine energy was produced non-stop by a reliable mechanical apparatus that, unlike natural resources, was not affected by atmospheric phenomena.

The improvements to the steam engine made mainly by James Watt made it exploitable in the manufacturing and transport industries. The relation is evident: the cylinder-steam distribution valve group can be considered the real predecessor of fluid power.

The studies and experiments by Torricelli, Pascal, Bernoulli, Boyle and Mariotte, Gay Lussac, along with the invention of the electric motor by Galileo Ferraris and internal combustion by Rudolf Diesel, led to the 20th century technology.

FIELDS OF APPLICATION OF OIL HYDRAULICS

As said above, it is difficult to list all the numerous industries where fluid power is employed. Still, oil hydraulics alone is used in very different fields and, unlike pneumatic transmissions that are hampered by unwieldy compressors, it is perfectly suitable for the applications on moving vehicles.

In general, oil hydraulic applications are divided into two large areas; for convenience of classification, these include many systems that actually do not belong to them.

Non-movable oil hydraulic systems of fixed machines placed in a specific environment are classified as **industrial** or **stationary** applications, e.g. plastic extrusion or drawing presses, lathes, milling cutters, etc.; this field also includes the numerous systems that are found in fixed machines, like in the food and pharmaceutical industries, even though they do not fall into the heavy industry category.

The **mobile** field includes oil hydraulic systems of the so-called **self-propelled machines**, i.e. machines that need to move on the spot, like for instance shovels, farm tractors and snowploughs. The naval and aerospace industries too fall into this category.

Oil hydraulics is widely used also in race and ordinary cars, but this industry is deemed independent and this work will deal with it very briefly.

It is important to highlight that many parts of hydraulic circuits can be used in the stationary and mobile fields, yet these often demand different components; as a result, the planning phase should also involve a suitability evaluation of each unit by considering both its adjustment/working characteristics and its structure compatibility with fixed or moving systems.

STANDARDS

The following chapters deal with the regulations issued by national and international standardisation organisations. These regulations cannot be quoted in full for copyright reasons, but subject development is based on them.

The organisations regulating the numerous world industries are the UNI in Italy and the ISO internationally speaking.

These organisations often amend their regulations; the standards quoted in this work are obviously updated to the latest version prior to publishing.

ASSOFLUID Technical Secretariat keeps abreast of the latest amendments and provides industry professionals with information.

This work also explores some of the standards on machinery safety issued by foreign organisations and CEN in order to protect workers' health and safety.

Note that the recommendations issued by the CETOP are no longer in force because they were included in different ISO standards.

- ISO (International Organization for Standardization)
- CETOP (European Oil Hydraulic and Pneumatic Committee)
- CEN (European Committee for Standardization)
- UNI (Ente Nazionale Italiano di Unificazione – Italian Organization for Standardization) - Italy
- DIN (Deutsches Institut für Normung e.V. - German Institute for Standardization) - Germany
- VDMA (Verband Deutscher Maschinen- und Anlagenbau – German Engineering Federation) - Germany
- ANSI (American National Standards Institute) - USA
- NFPA (National Fluid Power Association) - USA
- CNOMO (Comité de Normalisation des Moyens de Production - Standardization Committee for Manufacturing and Equipment) - France
- IEC (International Electrotechnical Commission)
- CEI (Comitato Elettrotecnico Italiano - Italian Electric, Electronic and Telecommunication Standardisation Institution) - Italy
- NEC (National Electrical Code) – USA
- SAE (Society of Automotive Engineers - for the classification of mineral oils) - USA

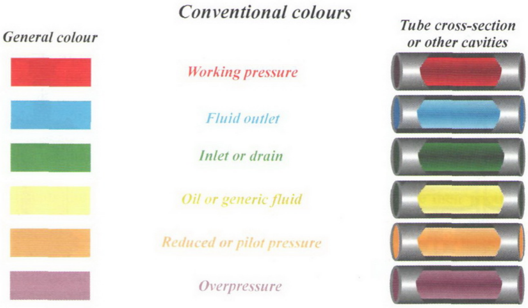


Figure I.1

Under ANSI standards, Figure I.1 shows the colours used in the figures in this work for pressurised, outlet, drain, suction, reduced pressure and control fluids.

BASIC HYDRAULIC CIRCUIT

A hydraulic circuit for the on/off control of an actuator is made up of different elements whose dimensioning depends on the load to operate.
This system can be divided into three different generic groups (see Figure I.2):

1. *Pumping system*, made up of a tank (11) filled with the hydraulic fluid (13), an operation prime mover (2) with its feed/control panel (1), a volumetric pump (4) connected to the prime mover through a flexible joint (3) and its suction pipe.
2. *Adjustment and distribution system*, made up of a relief valve (7), a pressure gauge (5) connected to a cut-off valve (6), a directional valve (8) for cylinder control.
3. *Actuator*, represented by a double-acting cylinder (9) in the figure.

The elements are connected through rigid pipes or hoses.

The prime mover, which is generally electrical in industrial applications and endothermic in mobile ones, is mechanically connected to the hydraulic pump; this can be a gear, vane, screw or piston pump depending on the circumstances. The fitting can be axial and direct if the number of revolutions (rpm) of the former equals the latter or it can consist in a mechanical speed reduction unit if the pump needs a different speed.

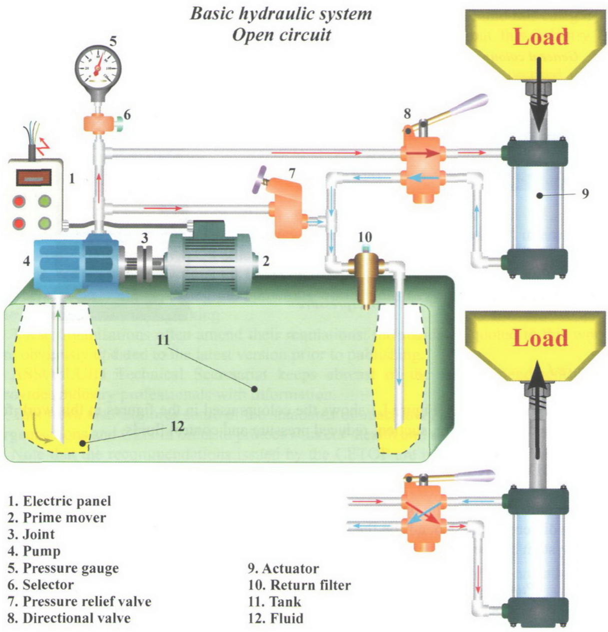


Figure I.2

It is now worth saying that, as far as flow changes are concerned, it is not advisable to change the number of revolutions of the prime mover, except for some special cases where the use of this technology is related to energy conservation (for instance plastic injection machines): revolution changes of three-phase electric motors are difficult and ('inverter'-like) electronic systems are needed, whereas the changes of operation parameters in moving vehicles (endothermic motor) affect the parameters of other circuits such as a shovel where the hydrostatic drive is totally independent from bucket operations.

The fluid the pump sucks (usually mineral oil with additives, but sometimes it is water, water/glycol and other chemical compounds) is pumped into the pipes and it is sent first to the maximum pressure valve and then to the directional valve.

The relief valve controls the pressure, if necessary, at the maximum level the pump or the weakest circuit component can sustain (as a result, the pressure needed to move the load must be lower than the pump rated value).

A pressure gauge operated by its cut-off valve displays the circuit pressure level.

By acting on the directional valve lever, the fluid operates the linear actuator piston; as a result, the rod rigidly connected to this entails the upward or downward movement of the load.

While the system is working, some minute solid particles like rust, welding and working residues, etc. can detach from the internal part of components and enter the fluid. These particles are driven between the moving elements of the different components during fluid recycle and they irremediably damage their friction surfaces. This is why any oil hydraulic circuit demands appropriate filters that should be dimensioned and put in specific spots depending on the needs of each plant.

The above example takes this into consideration, which is common for linear applications, the fluid is sucked from the tank, sent to the actuator and sent back to the tank passing through the filter. This system is called an **open circuit** because of its direct unloading and the resulting pump suction from the tank (Figure I.2).

A **closed circuit** (Figure I.3) is required in many applications like winches and hydrostatic drives where the actuator is a motor operating in both rotation directions. The unloading pipe is not connected to the tank but directly to the pump and, as a result, the fluid flows through a circle from the pump to adjustment valves and the motor; a second auxiliary pump sucking from a small tank replenishes oil.

Basic hydraulic system
Closed circuit

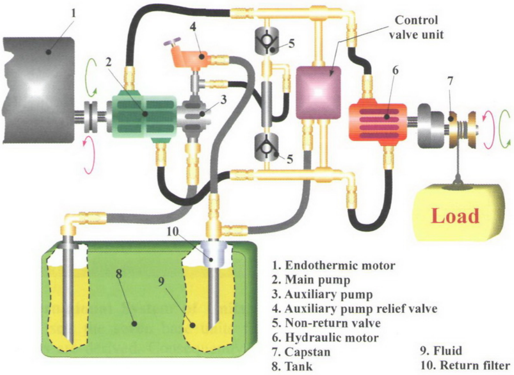


Figure I.3

Chapter 1

PRINCIPLES OF HYDRAULICS

Hydraulics is the science that studies the physical laws of fluids and it is generally subdivided into two parts: **hydrostatics** deals with fluids *at rest* whereas **hydrodynamics** considers fluids *in motion*.

By **fluid** we mean the phase of matter in which molecules are not packed closely together like in solids but they are free to flow. Besides all the liquids, gases too are considered fluids even though their molecules have repulsive forces leading them to repel from each other. A fluid can be held only in a container that determines the fluid shape: while a liquid fills the lower part, a gas fills the whole container regardless of the gas quantity and the shape of the container.

Liquid fluids only are employed in oil hydraulics, the only exceptions being the pneumatic controls of some applications and compressed gases used in accumulators; as a result, the generic term 'fluid' denotes oil and any other liquid used in the system.

UNITS OF MEASUREMENT

SI International System		
Quantity	***Unit***	***Symbol***
Length	metre	m
Mass	kilogramme	kg
Time	second	s
Electric current intensity	ampere	A
Temperature	Kelvin (Celsius)	K (°C)
Luminous intensity	candela	cd
Amount of substance	mole	mol

The **International System of Units**, also known as **SI**, which is acknowledged worldwide, sets the seven base units from which all the units used in the various technologies are derived. Consequently, surfaces are measured in square metres (m^2) and volume in cubic metres (m^3).

These two units are not functional in mechanics and oil hydraulics: cm^2 and mm^2 are used for surfaces, cubic centimetres (cm^3) for motor and pump displacement and cubic decimetres (dm^3) or rather litres (1 litre = 1 dm^3) for piping and component flow.

The International Systems also allows bar (1 bar = 10^5 pascal) to measure pressure.

Mass per volume - ρ

The mass per volume ρ (rho) of a body is the ratio between the body mass and its volume:

$$\rho = \frac{\text{mass}}{\text{volume}} = \text{kg/m}^3$$

Note that the ratio between weight and volume ($\gamma = \rho \bullet g = N/m^3$) was defined as 'specific weight' γ in the obsolete technical system of measurement units.

Mass per volume is often referred to as '**density**'.

Speed - v

The speed v of a body is defined as the ratio of the distance in metres S to the time in seconds taken to travel it:

$$v = \frac{\text{space}}{\text{time}} = \text{m/s}$$

In many cases the SI measure of length (metre) is not functional and it is thus converted into its multiples (kilometres) or submultiples (centimetres, millimetres, microns).

The SI unit of measurement for time (second, sometimes abbreviated to 'sec') is often converted into minutes (min).

Acceleration – a

It is the ratio between speed v and time t. Since v = m/s, acceleration is:

$$a = \frac{m/s}{s} = \text{m/s}^2$$

Gravity Acceleration - g

Its value is equivalent to the gravitational pull offset by the centrifugal force resulting from the Earth's rotation, which means it is the acceleration that keeps bodies on the Earth motionless. Its average value is:

$$g = 9.81 \text{ m/s}^2$$

Force – F

Force F is the product obtained by multiplying the mass m by the acceleration a.

$$F = m \cdot a = kg \cdot m/s^2 = N \text{ (newton)}$$

Forces were measured in kilogramme-force (kgf) or kp (kilopond) in the obsolete technical system of units of measurement:

$$F = m \cdot g = kg \cdot 9.81 \text{ m/s}^2 = kgf \qquad \text{hence } 1 \text{ kgf} = 9.81 \text{ N}$$

Torque/Twisting Moment – M

Torque or twisting moment is the system of two parallel, equivalent and opposite forces that, when applied to a body, tend to make it rotate around an axis which is perpendicular to its plane.

Previously measured (technical system) in kilogrammes per metre (kilogramme-metre = kgm), in the SI it is the product of 1 newton and 1 metre.

$$M = N \cdot m = Nm$$

Since this unit is rather small for fluid power, one of its multiples (daNm, decanewton per metre) is used instead; daNm is slightly bigger than the obsolete technical unit kgm.

Work and Energy – A and E

The unit of measurement for work or energy, the joule (J), is the product of the force F in newtons and the space S in metres:

$$A = \text{newton} \cdot \text{metre} = J$$

Power – P, N

The ratio between work in joules and time in seconds is equivalent to the power P or N.

$$N = \frac{J}{s} = W$$

The units of measurement of power are the watt or its multiples kW (kilowatt = 1000 watts) or MW (megawatt = 1000000 watts).

Power is usually shortened to P, yet N is instead used in this document so as not to confuse it with pressure, whose symbol is a lower case p.

Frequency - f

Frequency is the number of cycles a variable quantity makes per unit of time.

$$f = \frac{\text{number of cycles}}{\text{second}} = \text{n/s} = \text{Hz}$$

Frequency is measured in Hertz (Hz).

Rotation Frequency (number of revolutions or speed) – n – rpm

It is the number of revolutions of a pump or motor shaft. However, in mechanics it is better to consider the number of revolutions made in one minute.

Besides n, another very common abbreviation is rpm (revolutions per minute).

$$1\text{n},\ 1\text{rpm} = \frac{1}{60\text{s}} = \text{revolutions/min}$$

Heat – q

By heat we mean the kinetic energy resulting from the quick motion of atoms and molecules that make up bodies. Its units of measurement in the technical system were the calorie (cal) or the kilocalorie (kcal), 1 calorie and 1 kilocalorie being the amount needed to increase the temperature of respectively 1 gramme and 1 kilogramme of water from 14.5 °C to 15.5 °C at atmospheric pressure. It is measured in J (Joule) under the SI.

$$1\ \text{cal} = 4.187\ \text{J};\ 1\text{kcal} = 4187\ \text{J};\ 1\text{kW} = 1\ \text{kJ/s} = 3600\ \text{kJ/h}$$

Pressure - p

Pressure p is defined as the ratio between a force F and the surface S to which it is applied:

$$p = \frac{\text{force}}{\text{surface}} = \text{N/m}^2\ \text{(pascal)}$$

Pressure and its units of measurement are further discussed in the following paragraphs.

British and US Measurement Units

Despite the worldwide dimension of the SI, British measurement units are still employed in the United States of America, which means length is measured in inches or feet, mass in pounds, volume in cubic inches or gallons and pressure in PSI.

	SI International System	***British and US Units***	***Conversion SI → USA***	***Conversion USA → SI***
Length	Metre -m-	Inch -in- Foot -ft-	1m=39.37in 1mm=0.039in 1m=3.28084ft 1mm=0.003ft	1in=0.0254m 1in=25.4mm 1ft=0.3048m 1ft=304.8mm
Surface	m^2	in^2 ft^2	$1m^2=1550.4in^2$ $1m^2=10.76ft^2$	$1in^2=0.000645m^2$ $1ft^2=0.0929m^2$
Volume	cm^3 m^3 dm^3 or Litre	in^3 ft^3 Gallon -gal-	$1cm^3=0.061in^3$ $1m^3=35.31ft^3$ $1dm^3$=1litre= 0.265gal	$1in^3=16.39cm^3$ $1ft^3=0.02832m^3$ $1gal=3.78dm^3$
Mass	kg	Pound -lb-	1kg=2.2046lb	1lb=0.4536kg
Force	newton -N-	lbf (lb force)	1N=0.225lbf	1lbf=0.4483N
Density	kg/m^3	lbf/ft^3	$1kg/m^3$= $0.0624lbf/ft^3$	$1lbf/ft^3$= $16.02kg/m^3$
Speed	m/s	ft/s	1m/s=3.3ft/s	1ft/s=0.3048m/s
Torque	N•m	lbf•ft	1N•m= 0.737lbf•ft	1lbf•ft= 16.02N•m
Pressure	pascal -Pa- bar=10^5Pa	psi (Pound/ Square Inch- lbf/in^2-)	1bar=14.5psi	1psi=0.069bar
Work	Joule – J-	lbf•ft	1J=0.737lbf•ft	1lbf•ft=1.356J
Heat	Joule –J- (1cal=4.187J) (1J=0.239cal)	BTU	1J=0.0009BTU	1BTU=1055.1J

Power	Watt –W- Horse power (Cavalli vapore) -CV-	ft•lbf/s Horse power -HP-	1W= 0.73ft•lbf/s 1CV=735W	1ft•lbf/s= 1.3558W 1HP=745.7W
Kinematic Viscosity	m^2/s cSt $-mm^2/s$	ft^2/s	$1m^2/s=0.15ft^2/s$	$1ft^2/s=6.896m^2/s$
Temperature	Degree Celsius -°C-	Degree Fahrenheit -°F-	°C= (°F-32)/1.8	°F= °C•1.8 + 32
Flow	m^3/s	in^3/min	$1m^3/s=$ $2^{-7}in^3/min$	$1in^3/min=$ $0.273 \cdot 10^{-6}m^3/s$
Oil hydraulic flow	1litre/min= $1/60000m^3/s$	gallons/min -GPM-	1l/min= 0.265GPM	1GPM= 3.78l/min
	SI International System	***British and US Units***	***Conversion SI → USA***	***Conversion USA → SI***

ELEMENTS OF HYDROSTATICS

A fluid in a container that is not subjected to heat or mechanical stress is considered an ideal, balanced liquid at rest. The pressure at all points of the liquid mass is known as "**hydrostatic pressure**".

Pressure

Expanding on what was written in the introduction, we stress that pressure p is the result of a force Facting on a surface S (Figure 1.1).

$$p = \frac{\text{force}}{\text{surface}} \qquad S = \frac{F}{p} \qquad F = p \cdot S$$

Pressure

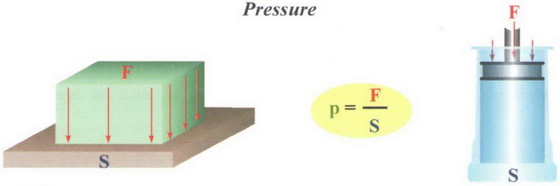


Figure 1.1

This notion in hydraulics can be represented as a force F acting on a piston without leakages placed on the free surface of a liquid held in a cylindrical container.

Pressure was measured in kgf/cm^2 in the obsolete technical system. This unit can easily be understood because everybody knows the value of one or more kilogrammes. Since force is obligatorily measured in newtons (N) under the International System, the decanewton (daN = N • 10) is adopted because it is more similar to the old unit; by the way, the decanewton is almost equivalent to the kg (1 kgf = 0.981 daN) and they can even be confused. As we are seeing later on, SI pressure is measured in N/m^2 and also daN/cm^2.

The following two examples make understand the notion of pressure better.

1) Given a force of 100 daN applied on the piston of a cylindrical container with a bore of 3 cm, how much pressure is exerted on the liquid in the container?

The piston surface is:

$$S = \frac{d^2 \bullet \pi}{4} = \frac{3^2 \bullet 3.14}{4} = \frac{28.26}{4} = 7.065 \text{ cm}^2$$

The pressure is:

$$p = \frac{F}{S} = \frac{100}{7.065} = 14.15 \text{ daN/cm}^2$$

2) The liquid in a cylinder with a bore of 10 cm has to be pressurised at 20 daN/cm^2. How much force should be applied on the piston?

The piston surface is:

$$S = \frac{d^2 \bullet \pi}{4} = \frac{10^2 \bullet 3.14}{4} = \frac{314}{4} = 78.5 \text{ cm}^2$$

The force is:

$$F = p \bullet S = 20 \bullet 78.5 = 1570 \text{ daN}$$

This phenomenon can also be taken into account without the piston, for instance, in the case of a liquid in a container with a hole in the lower part.

If the container is full, the liquid jet reaches a specific distance; as the liquid level goes down, the jet distance reduces constantly and the liquid falls perpendicularly when it is almost empty (Figure 1.2).

The force is equivalent to atmospheric pressure plus the mass (weight) of the liquid found in the container. As the fluid volume reduces, the force exerted by atmospheric pressure does not vary, but the mass decreases, thus reducing the overall pressure. This is why the jet is weaker and weaker.

A fundamental prerequisite for the success of the experiment is that the hole diameter does not vary during the experiment. As a matter of fact, we are going to see later on that a change in the opening entails different conditions linked to hydrodynamic laws.

Simple experiment on pressure

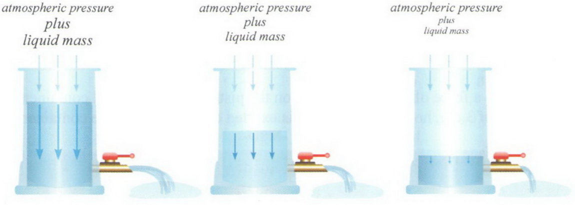


Figure 1.2

A famous example is the case of submarines that must be equipped with metal sheets that can endure pressure hundreds of metres below the sea level; a bathyscaphe for deep-sea exploration must be equipped with several-millimetre-thick steel armour.

The principle is always the same. The air exerts a pressure on the water free surface; we all sustain air pressure when we are on dry land, albeit at different levels depending on whether we are in the mountains or in a low-lying area; deep-sea pressure is determined by the weight of the water above.

According to **Stevin's law** (Simon Stevin, Belgium, 1548-1620), in order to determine the value of real hydrostatic pressure (i.e. if atmospheric pressure equals 0) of a fluid column on a surface S, the overall liquid weight on the vertical on the surface S has to be calculated. If the density ρ (kg/m^3) is known, we can determine it by multiplying density by the volume of the fluid column and gravity acceleration.

$p = \frac{\rho \bullet V}{S} \bullet g$ and since $\frac{V}{S} = h$ (height), hydrostatic pressure p can be determined as follows:

$$p = \rho \bullet g \bullet h \qquad \text{hence } h = \frac{p}{\rho} \bullet \frac{1}{g}$$

As for the cylindrical container without the piston, we can determine the pressure p exerted by a water column whose h is 100 cm on the bottom of a container whose surface S is 10 cm^2. Water density = 1000 kg/m^3, which is 0.001 kg/cm^3.

Given that the volume V of the water column is:

$$V = S \bullet h = 10 \bullet 100 = 1000 \text{ cm}^3 \text{ (1 litre)}$$

The pressure exerted by the liquid is:

$$p = \frac{\rho \bullet V}{S} \bullet g = \frac{0.001 \bullet 1000}{10} \bullet 9.81 = 0.981 \text{ N/cm}^2$$

Or more easily:

$$p = \rho \bullet h \bullet g = 0.001 \bullet 100 \bullet 9.81 = 0.981 \text{ N/cm}^2$$

The force F exerted by the water column of 1000 cm^3 is:

$$F = p \bullet S = 0.981 \bullet 10 = 9.81 \text{ N} = 1 \text{ kgf}$$

If the water column is 300 cm high, the pressure is:

$$p = \rho \bullet h \bullet g = 0.001 \bullet 300 \bullet 9.81 = 2.943 \text{ N/cm}^2 = 0.2943 \text{ daN/cm}^2 \text{ (bar)} = 0.3 \text{ kgf/cm}^2$$

Density is constant and pressure is therefore proportional to height, which is physically defined as '*static head*'.

From this principle defined by Stevin's law we can deduce also that the sum of pressure head p/ρ and elevation z at all the points of a liquid at rest is constant.

$$\frac{p}{\rho \bullet g} + z = \text{const}$$

The sum of pressure head and elevation is called *piezometric head* H (piezo from the Greek *piézein* → to press) (Figure 1.3).

Piezometric and pressure heads

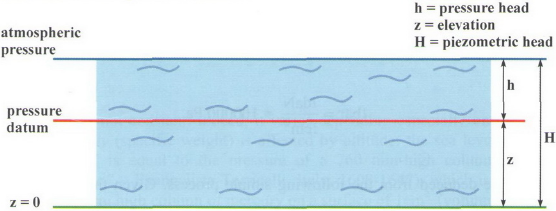


Figure 1.3

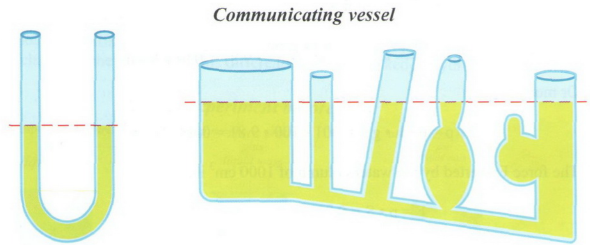


Figure 1.4

We can demonstrate that pressure (in this case atmospheric pressure) acts uniformly on a horizontal plane by filling a *U-shaped hose* or a system of *communicating vessels* with a liquid (Figure 1.4): the fluid reaches the same level in all the vessels.

Note that sometimes this experiment can prove to be unsuccessful. As a matter of fact, if one or more vessels have a cross-section smaller than 1 mm^2, the liquid in the vessels does not reach the normal level: a liquid having high adhesion properties (water, oil, etc.) exceeds it, whereas a liquid with poor adhesion properties like mercury does not even reach the normal level.

Under the International System (S.I.), the unit of pressure is the pascal (Pa), which is defined as the pressure exerted by 1 newton (N) per 1 m^2. Derived from the Giorgi System (the Italian electrical engineer Giovanni Giorgi proposed his system in 1901), the pascal is far smaller than the other units usually employed in oil hydraulics that vary from about 1000000 to 400000000 Pa. For practical reasons, one of its submultiples is therefore preferred, that is the MPa (Megapascal – 1MPa = 1000000 Pa = 10 bar).

Actually, the pressure unit used is the **bar**; it is defined as the pressure exerted by the force of 1 daN (decanewton) per 1 cm^2, which is quite similar to the atmospheric pressure unit (see below).

$$1\text{bar} = \frac{1\text{daN}}{1\text{cm}^2} = 100000 \text{ Pa}$$

This can be deduced from the following simple process. Given that $1\text{Pa} = \frac{1\text{N}}{1\text{m}^2}$, it can be multiplied by 100000 ($1\text{bar} = \frac{1\text{N}}{1\text{m}^2} \bullet 100000$) and the surface can then be converted into cm^2:

$$1\text{bar} = \frac{1\text{N}}{10000\text{cm}^2} \bullet 100000 \qquad 1\text{bar} = \frac{1\text{N}}{1\text{cm}^2} \bullet 10 \qquad 1\text{bar} = \frac{1\text{daN}}{1\text{cm}^2}$$

The bar is equivalent to a million baryes (10^6 baryes) in the obsolete CGS system (centimetre, gramme, second). The unit of measurement called **barye** (from the Greek *barys* → heavy) is equivalent to 1 dyne/cm^2, that is 0.1 N/m^2. The dyne is thus equivalent to about a milligramme.

The UK unit of pressure is the **psi**, which stands for:

$$\frac{\text{pounds}}{\text{square inch}} \left(\frac{\text{lbf}}{\text{in}^2} \right)$$

1 pound = 0.4536 kgf, 1 square inch (2.54^2) = 6.4516 cm^2
1 psi = 0.4536/6.4516 = 0.0703 kgf/cm^2 • 0.981 = 0.06895 bar
1 bar = 1 psi / 0.06895 = 14.5 psi

Under the UK system, the pressure measured with an instrument is often referred to as **psig** (psi gauge).

Except for fluid suction that is measured in absolute bars, fluid power always take into consideration **relative pressure**, i.e. air or liquid pressure as compared to the pressure of the surrounding environment. However, we must consider that the air surrounding the Earth exerts a pressure called *atmospheric pressure* (Figure 1.5); the sum of relative and atmospheric pressure is known as *absolute pressure***.**

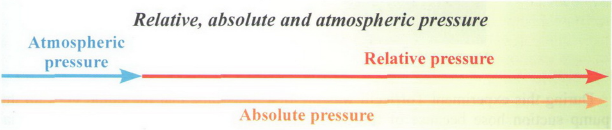


Figure 1.5

Air pressure is not constant: indeed, it varies depending especially on the elevation because air *density* (specific weight) is affected by altitude: the sea level atmospheric pressure at 0 °C is equal to the pressure of a 760 mm-high column of mercury (*"Torricelli's tube"* – Evangelista Torricelli, Italy, 1608-1647), which is equivalent to the mass of a 76 cm high column of mercury on a surface of 1cm^2 (volume 76 cm^3).

Given that mercury density ρ is 13590 kg/m^3, i.e. 0.01359 kg/cm^3:

$$p_{atm} = \frac{V \bullet \rho}{S} = \frac{76 \bullet 0.01359}{1} = 1.033 \,{}^{\text{kgf}}\!/_{\text{cm}^2}$$

or

$$p_{atm} = \rho \cdot h = 0.01359 \cdot 76 = 1.033 \ {}^{kgf}\!/_{cm^2}$$

Torricelli's experiment can be explained more easily: the atmospheric pressure applied to mercury is uniform at all points of the liquid (Pascal's principle – see below). Consequently, atmospheric pressure and the pressure of the column of mercury are equivalent, i.e. atmospheric pressure is equal to the pressure of a column of mercury with a pressure head h of 76 cm (Figure 1.6).

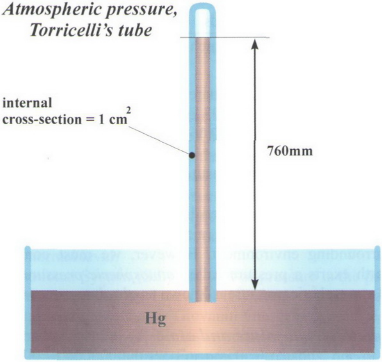


Figure 1.6

During this experiment Torricelli was the first to assume that the liquid goes up the pump suction hose because of the force atmospheric pressure exerts on its external surface, whereas his contemporaries claimed the liquid was sucked by the vacuum.

The atmospheric pressure measured with a column of water instead of a column of mercury is equal to 10.333 metres, that is the volume of 1.033 litres per 1 cm^2 equivalent to the weight of 1.033 kgf. Given that:

1kgf = 9.81N, 1N = 0.102 kgf (kilogramme-force) $\quad 1\text{bar} = \dfrac{1\text{daN}}{1\text{cm}^2} = \dfrac{1.102\text{kgf}}{1\text{cm}^2}$

The atmospheric pressure is therefore about as much as the 1 bar; since it is never constant, it is considered as equivalent to 1 bar for practical applications.

Pascal's Principle

Pascal's law states that *every fluid transfers the same pressure exerted on it at all points in all its mass.*

If a force F is applied into a spherical container (Figure 1.7) filled with a liquid or gaseous fluid through a fluid-tight piston with a cross-section S, a constant $p = \frac{F}{S}$ pressure is applied at all points of the container.

As force decreases and the piston cross-section does not vary, pressure too goes down; if the container was much bigger while the force applied and the piston surface did not vary, pressure would stay the same.

A simple test to examine the validity of Pascal's principle is shown in Figure 1.8.

Pascal's principle

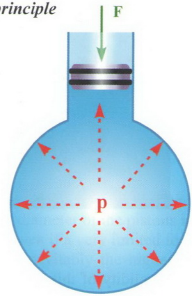


Figure 1.7

Demonstration of Pascal's principle

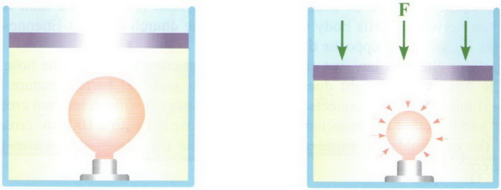


Figure 1.8

A rubber balloon filled with gas is attached to a weight and put to the bottom of a container filled with any liquid. By applying a force on the piston the balloon shrinks but it keeps a spherical shape: this means pressure is applied to the whole balloon surface in a uniform manner.

The same phenomenon occurs inside the balloon: the compressed air acts uniformly on the internal surface. Obviously, what is being compressed, thus allowing the piston to move downward, is not the incompressible liquid, but the air inside the balloon. When the force stops, the compressed air inside the balloon pushes the liquid and the piston back into their initial position due to Pascal's principle.

Figure 1.9

In order to demonstrate the validity of this principle, Pascal himself devised a very bizarre experiment (Figure 1.9). He connected a 15-metre vertical hose to a wooden barrel filled with water. He poured water into the hose until the barrel collapsed when water reached a specific level. As a matter of fact, the pressure of the column of water (acting on each cm^2 of the inner surface of the barrel) was so high that its overall force was substantial.

Blaise Pascal was born in 1623 in Clermont-Ferrand (France) and died in Paris in 1662 when he was 39. His body was buried in the church of Saint-Étienne-du Mont, next to the right column opposite the presbytery.

By applying Stevin's law, we can understand the cross-section of the hose does not affect the result if its height does not vary. Two identical barrels endure the same pressure if one of them is connected to a 1500 cm hose (a) with an internal cross-section of 1 cm^2 and the other is connected to a 1500 cm hose (b) with a 100 cm^2 cross-section, even if (b) can hold 100 times as much water as (a). Given that water density is $0.001 kg/cm^3$:

$$p = \frac{\rho \bullet V}{S}$$

(a) $$p = \frac{0.001 \bullet 1 \bullet 1500}{1} = 1.5 \text{ kgf/cm}^2$$ (b) $$p = \frac{0.001 \bullet 100 \bullet 1500}{100} = 1.5 \text{ kgf/cm}^2$$

or $$p = \rho \bullet h = 0.001 \bullet 1500 = 1.5 \text{ kgf/cm}^2$$

Hydraulic press

A very important application of Pascal's principle is the hydraulic press. In two tubes with different volumes connected by a pipe and filled with an incompressible fluid, the pressure exerted on the smaller piston and transmitted by the fluid acts uniformly on the whole surface of the larger piston.

Forces (see Figure 1.10) are directly proportional to the cross-sections of the cylinders:

$F_1 : S_1 = F_2 : S_2$ or $F_1 : F_2 = S_1 : S_2$ if we calculate F_2 from the proportion we obtain:

$$F_2 = \frac{F_1}{S_1} \bullet S_2 = F_1 \bullet \frac{S_2}{S_1} \quad \text{or} \quad \frac{F_2}{F_1} = \frac{S_2}{S_1} = \frac{D^2}{d^2} \quad \text{hence} \quad \frac{F_2}{F_1} = \frac{D^2}{d^2}$$

The force on the smaller piston increases the force on the larger piston in proportion to the areas of the two pistons. If the diameter of the pistons is changed proportionally, a very high pressure can be generated even if a very limited force is transmitted.

However, Figure 1.10 shows that inevitably the higher pressure is, the shorter the piston stroke is.

Hydraulic press

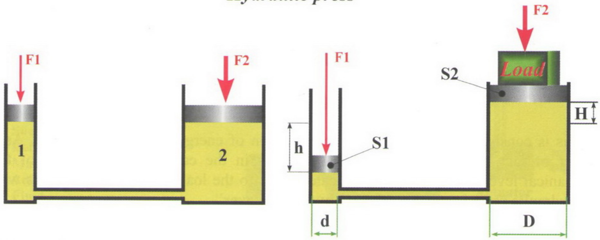


Figure 1.10

From the formula above $\frac{F_2}{F_1} = \frac{D^2}{d^2}$ we obtain $F_1 = \frac{d^2}{D^2} \cdot F_2$

For instance, with pistons d = 10 mm, D = 100 mm and the load to move F_2 = 1000 N, the force F_1 to apply is:

$$F_1 = \frac{10^2}{100^2} \cdot 1000 = 10 \text{ N}$$

(a force equal to about 1 decanewton can lift 100 of them!).

Yet, there is a problem with piston strokes. Indeed, if the small piston stroke is 500 mm, the load translation amounts to only 5 mm:

$$H = \frac{d^2}{D^2} \cdot h \qquad H = \frac{10^2}{100^2} \cdot 500 = 5 \text{ mm}$$

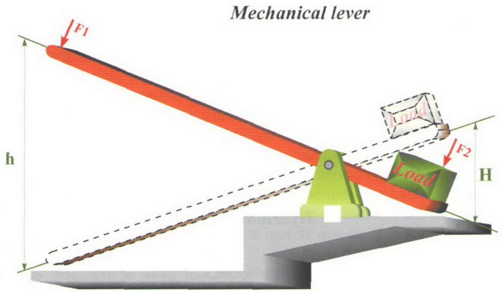


Figure 1.11

This is consistent with the law of conservation of energy or equivalent work. As a matter of fact, the conditions are the same as in the case of the principle of the mechanical lever whose fulcrum is located close to the load (Figure 1.11); this is why the hydraulic press is considered as a "**hydraulic lever**".

The hydraulic press described so far is rather unsuitable for practical applications because once the translation occurs, the press cannot retain its position and the resulting strokes are not satisfactory.

The system proves to be really operational if the smaller piston is equipped with a lever, two valves and a fluid tank (Figure 1.12).

The smaller cylinder (1) is connected to a tank (2) holding the fluid, usually oil, via a check valve (3) and a hose. Its piston, operated by the manual operation lever (5), sucks oil from the tank [suction] during the upward movement, whereas during the downward movement the piston pushes it [delivery] through the valve (4) and the hose into the larger cylinder.

The non-return valve (3) prevents the fluid from returning into the tank during delivery, while the non-return valve (4) keeps the pressurised oil inside the cylinder (2) during suction.

The number of lever (5) operations determines the overall stroke of the cylinder piston (6). Exceptional forces can be applied on the larger cylinder by lengthening the lever where the input force is exerted. In terms of physics this means that overall positive work is equivalent to the overall negative work.

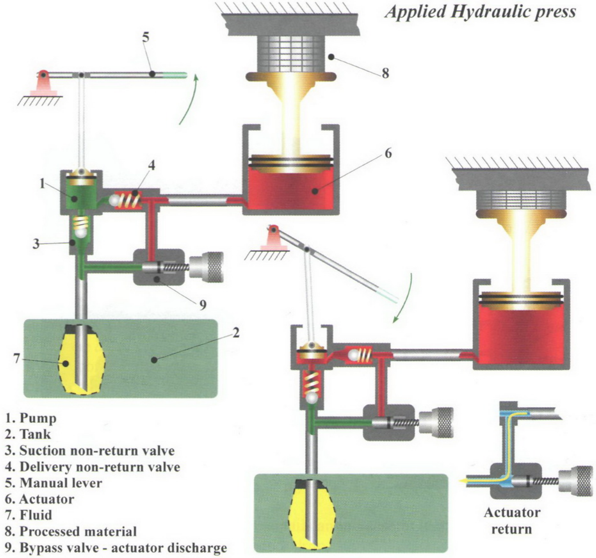


Figure 1.12

Fluid power applications are based on the principle of the hydraulic press: the smaller cylinder is the pump, the larger cylinder is the actuator.

In an automatic system an electric or endothermic motor pump replaces the manual pump; in addition, an automatic system is equipped not only with non-return valves, but also actuator directional control valves and pressure/flow control valves.

Figure 1.13 (schematic drawing of an industrial deep-drawing press) shows an application example based on the principle of the hydraulic press.

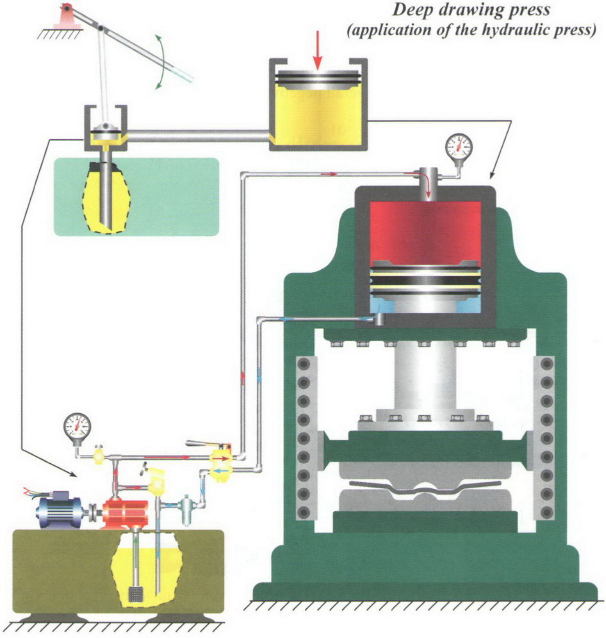


Figure 1.13

Compression of a perfect gas

Oil or any other liquid in gas-loaded accumulators (see chapter 14) is pressurised by a pre-compressed gas, usually nitrogen. We are now going to explain some principles about gas compression that are going to be useful later on for accumulators.

The fundamental quantities that characterise a gas are volume, temperature and pressure.

Boyle-Mariotte law (Robert Boyle, Ireland, 1627-1691 / Edme Mariotte, France, 1620-1684): *the volume and pressure of an ideal gas at a constant temperature are inversely proportional.*

This means that the pressure of a fixed amount of gas doubles by halving the volume the gas can occupy; *the product of volume and absolute pressure is constant.*

$$p_{a1} \bullet V_1 = p_{a2} \bullet V_2 = p_{a3} \bullet V_3 = ... = \text{constant}$$

If 8 m^3 of ideal gas are compressed at atmospheric pressure and constant temperature so that the gas occupies a volume of 1 m^3, what is its final pressure?

$$p_{a1} \bullet V_1 = p_{a2} \bullet V_2$$

p_{a1} = atmospheric pressure = 1bar
$V_1 = 8\ m^3$
$V_2 = 1\ m^3$
$p_{a2} = ?$

$$p_{a2} = \frac{p_{a1} \bullet V_1}{V_2} = \frac{1 \bullet 8}{1} = 8 \text{ absolute bar, hence a relative pressure of 7 bar.}$$

First law of Gay-Lussac (Joseph Gay-Lussac, France, 1778-1850): the *volume and the absolute temperature of an ideal gas at constant pressure are directly proportional.*

Second law of Gay-Lussac: *the absolute pressure and temperature of an ideal gas at fixed volume are directly proportional.*

Given a constant pressure, the volume V of an ideal gas is directly proportional to its absolute temperature T (°K):

$$V_1 : V_2 = T_1 : T_2$$

and, given a constant volume, its pressure is directly proportional to its temperature:

$$p_{a1} : p_{a2} = T_1 : T_2$$

This means an increase in pressure entails an increase in temperature and, vice versa, a decrease in pressure makes temperature drop.

The laws of Boyle-Mariotte and Gay-Lussac are fully valid only for ideal gases, whereas pure *real* gases (such as hydrogen oxygen, nitrogen) or gas mixtures like air do not exhibit these exact properties.

ELEMENTS OF HYDRODYNAMICS

Hydrodynamics deals with fluids in motion inside piping and *penstocks;* we are going to take into consideration only the latter as far as fluid power is concerned. By penstock, piping or only cross-section we mean everything concerning the translation of fluids in oil hydraulic circuits like hoses, valve ports, pumps, actuators and so on.

Flow

The volumetric flow rate Q (Figure 1.14) is defined as the volume of liquid V that passes through a cross-section S per unit time t.

Flow determination

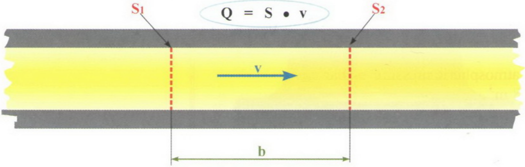


Figure 1.14

Assuming every particle has the same average speed v, the liquid therefore flows from the cross-sections S_1 and S_2 within a time t; if b is the distance between S_1 and S_2, the volume of the liquid is $V = S \cdot b$, hence:

$$Q = \frac{V}{t} = \frac{S \cdot b}{t}$$

The volume V of the liquid that passed from S_1 to S_2 is also equivalent to the product of

$S \cdot v \cdot t$ (cross-section • speed• time), hence:

$$Q = \frac{V}{t} = \frac{S \cdot v \cdot t}{t} = \mathbf{S \cdot v} \text{ (cross-section • speed)}$$

Under the International System, flow is measured in m^3/s, but it is better to use dm^3/min or litre per minute (l/min or lpm) in fluid power. The UK units of flow employed in oil hydraulics are cubic inches per minute (in^3/min) or gallons per minute (GPM) (see conversion table above).

The **continuity equation** for flow states that *if the cross-section varies, flow is constant at all points of the hose* (this is fully valid for mass flow rate and partially valid for volumetric flow rate because fluid compressibility should be taken into account). As a result, *every cross-section change results in a speed change* (Figure 1.15).

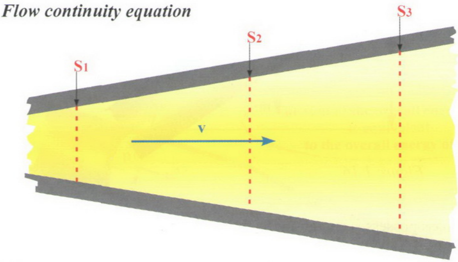


Figure 1.15

In practice, the volume of the fluid passing through the cross-section S_1 is equal to the volume of cross-section S_2 and cross-section S_3:

$$Q = S_1 \bullet v_1 = S_2 \bullet v_2 = S_3 \bullet v_3 = \text{const}$$

Speed is inversely proportional to the cross-section:

$$\frac{v_1}{v_2} = \frac{S_2}{S_1}$$

The speed of the different fluid particles varies depending on the friction they have to face and even for radial (or traverse) speed components; for practical reasons, **average speed** must be therefore considered, that is the average of all the speeds of particles, in other words the theoretical speed at which the liquid passes through a cross-section.

Forms of energy in a fluid

Energy is the capacity of a body or a system to do work due to specific characteristics; work is defined as the transfer of the force application point along its action line. Universally speaking, energy can neither be created nor destroyed but it can be changed in form depending on the physical conditions of the system (for instance, the mechanical energy due to friction is transformed into heat energy, the electric energy of an electric motor transforms into mechanical energy through a drive shaft).

The energy a fluid possesses can have four different forms. Consider a liquid mass m, whose centre is at a height z from the reference plane x, runs through an inclined hose at a specific speed v and pressure p (Figure 1.16).

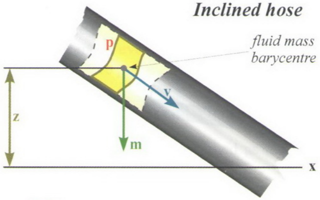


Figure 1.16

The liquid mass possesses energy in the following forms:

1) *Potential energy* E_{pot}: it depends on the position of the fluid mass vis-à-vis the plane below; consequently, it is proportional to the mass m and height z of its centre from the plane.

$$E_{pot} = m \bullet g \bullet z$$

2) *Pressure energy* E_p: it is the product of mass and pressure head ($p/\rho g$).

$$E_p = m \bullet g \bullet \frac{p}{\rho g} = m \bullet p/\rho$$

3) *Kinetic energy* E_c: a fluid in motion has a kinetic energy that is proportional to the mass and the square of its speed.

$$E_c = \frac{1}{2} mv^2$$

4) *Internal or thermal energy*: it results from the internal friction of the fluid (pressure drops). Unlike the other three forms, it is a negative form of energy that needs limiting with proper solutions in oil hydraulics.

Potential and pressure forms of energy are static, while kinetic and internal forms of energy are dynamic.

Bernoulli's Principle

Inspired by the law of conservation of energy, which states energy is neither created nor destroyed, Bernoulli's principle is one of the fundamental law of hydraulics. (Daniel Bernoulli or Bernouilli, Switzerland, 1700-1782 – author of the fundamental book "*Hydrodynamique*", 1738).

In short, the principle (Figure 1.17) states that *the overall energy of a liquid mass in any cross-section of a flow at steady state is equivalent to the sum of all forms of energy* (potential energy E_{pot}, pressure energy E_p, kinetic energy E_c, thermal energy e) *and it is the same throughout the hose…* obviously as long as no machines that absorb or produce energy are connected to the pipe.

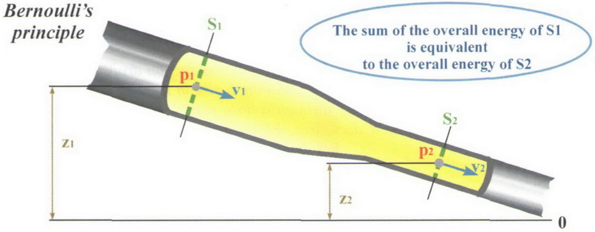


Figure 1.17

If the mass m equals 1 (ex. $E_{pot} = m \cdot z = 1 \cdot z = z$), the different forms of energy can be considered as heights and in terms of mathematics the principle is:

$$z_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} + e_1 = z_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + e_2$$

Since $\rho \cdot g$ is equivalent to the obsolete specific weight γ, the theorem can be written as:

$$z_1+p_1/\gamma+V_1^2/2g+e_1 = z_2+p_2/\gamma+V_1^2/2g+e_2$$

At present, physicists recommend that the following formula should be preferred to obsolete notions dating back to the technical system, like the specific weight:

$$\rho g z_1 + P_1 + \tfrac{1}{2} \cdot \rho \cdot V_1^2 + e_1 = \text{const}$$

Bernoulli's Principle in oil hydraulics

In a horizontal *delivery* hose $z_1 = z_2$ whose cross-section is constant $S_1 = S_2$, speed is constant, $v_1 = v_2$ and kinetic energy is negligible compared to the substantial pressure of an oil hydraulic system. The principle can thus be simplified as follows:

$$\text{Delivery hose} \rightarrow \frac{p_1}{\rho g} + e_1 = \text{const}$$

The complete equation of Bernoulli's principle must be used for *suction* hoses whose pressure is less than atmospheric pressure:

$$\text{Suction hose} \rightarrow z_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} + e_1 = z_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + e_2$$

The pressure of the fluid in a pipe with a constant cross-section drops along the flow due to friction and cross-section reduction, entailing a rise in thermal energy, while its speed increases. When the fluid flows through a hose with a larger cross-section, its pressure increases and its speed decreases. Flow rate does not vary in both cases (Figure 1.18).

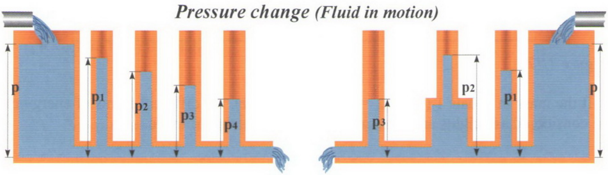


Figure 1.18

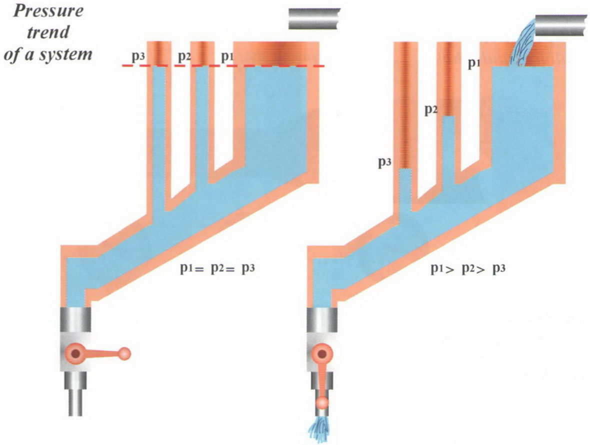


Figure 1.19

Consequently, pressure 'trend' under static and dynamic conditions is an important parameter affecting all oil hydraulic and pneumatic systems. Figure 1.19 shows that pressure at all points is the same if the valve is closed, whereas by opening it pressure drops near the opening due to Bernoulli's principle as stated above. *The pressure of a fluid at rest (hydrostatics) is the same at all points of the system at the same depth; the pressure of a fluid in motion (hydrodynamics) changes depending on the system conditions.*

Figure 1.20 shows the application of Bernoulli's principle to oil hydraulic hoses.

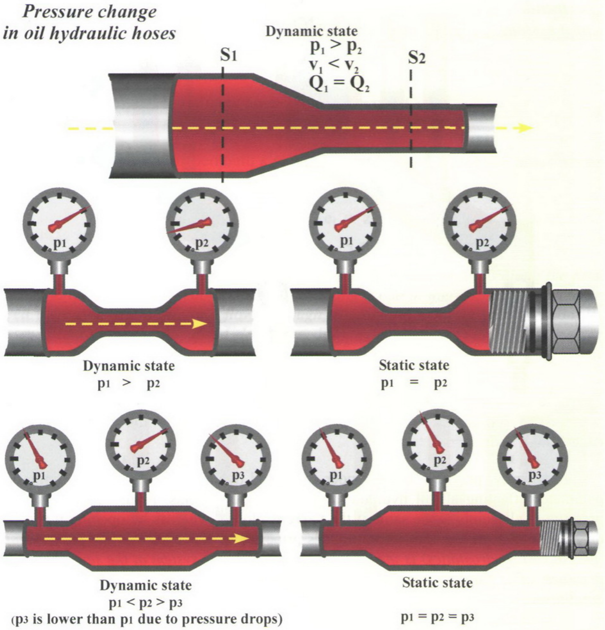


Figure 1.20

Torricelli's Principle

When a valve in a tank is opened, h being the head between the valve and the free surface, the speed of the fluid flowing out of the opening is equivalent to the speed of a solid falling from the same height due to gravitation (fall of bodies).

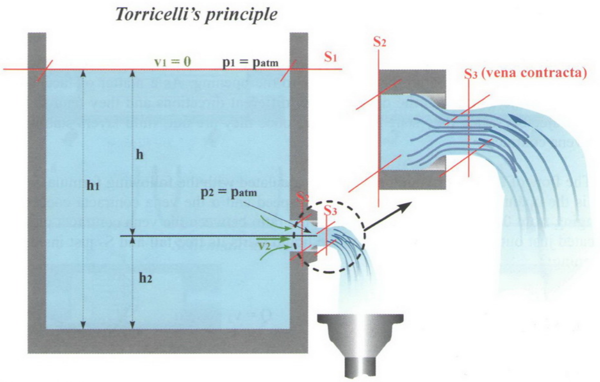


Figure 1.21

If Bernoulli's principle is applied to cross-section S_1 (free surface – atmospheric pressure) and cross-section S_2 (located just outside the tank at atmospheric pressure), the resulting formula is (see Figure 1.21):

$$h_1 + \frac{p_1}{\rho} + \frac{{v_1}^2}{2g} = h_2 + \frac{p_2}{\rho} + \frac{{v_2}^2}{2g}$$

Since speed is $v_1 = 0$ and pressure are $p_1 = p_2 = p_{atm}$, we obtain:

$$h_1 = h_2 + \frac{{v_2}^2}{2g}$$

Since $h_1 - h_2 = h$ (pressure on the discharge opening), the discharge speed is:

$$v_2 = \sqrt{2\,g\,h}$$

Torricelli explained his formula as follows: *"the speed of a fluid flowing out of an opening at atmospheric pressure is equivalent to the speed of a body falling in a vacuum from a height that is equal to the head"*.

The phenomenon known as 'vena contracta' should be taken into account in order to determine the discharge flow of the fluid.

Newton (Isaac Newton, England, 1642-1727) pointed out that Torricelli's speed ($v_2 = \sqrt{2\,g\,h}$) is not reached in the opening cross-section but in a smaller section slightly downhill, the so-called 'vena contracta point', where all the layers move in parallel, have the same speed and are perpendicular to the opening. As a matter of fact, liquid particles in the tank reach the opening from different directions and they tend to keep their trajectory when they start flowing out; once they are out, fluid layers stabilise in the vena contracta.

The flow of the fluid running out can be calculated with the following formula, where S_2 is the opening cross-section, v_2 the flow speed and α the vena contracta coefficient ranging from 0,5 to 0,8; this coefficient is the ratio between the vena contracta point S_3 located just outside the opening when the liquid starts its free fall and S_2 just inside the opening:

$$\alpha = \frac{S_3}{S_2} \qquad\qquad Q = v_2 \bullet S_2 \bullet \alpha$$

Laminar and turbulent flow

A *'flowing fluid'*, i.e. a liquid in motion, can be affected by two different conditions: the *laminar flow* or Poiseuille's flow in which fluid threads flow in parallel and the *turbulent flow* in which liquid particles have irregular trajectories they keep on changing, which entails continuous stirring.

The traditional empirical method to determine the type of motion consists in putting a drop of coloured liquid on the central layer surface of the flowing fluid: it is a laminar flow if only the central layer is coloured, whereas it is a turbulent flow if the whole liquid volume of that point is coloured (Figure 1.22).

A fluid flowing in a hose at a very low speed has a laminar motion; by increasing speed gradually, the flow experiences first the so-called 'critical transition', in which the motion can be both laminar and turbulent, and secondly it becomes steadily turbulent.

If speed v in m/sec, the bore of the hose in m and kinematic viscosity ν in m^2/s (stokes – see chapter 2) are known, the so-called *'Reynolds number'* (R_e) can be employed to determine whether the flow is laminar or turbulent:

$$R_e = \frac{v \bullet d}{\nu}$$

Roughly, the flow in cylindrical hoses is laminar or turbulent if: $R_e < 2200$ = laminar flow - R_e ranging between 2200 and 4000 = critical transition - $R_e > 4000$ = turbulent flow.

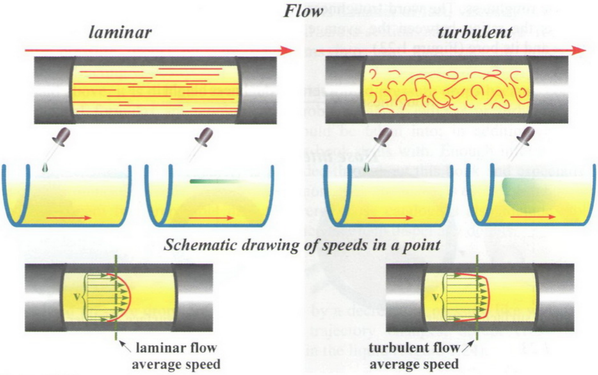


Figure 1.22

It is difficult to determine the exact critical Reynolds number at which a flow starts being turbulent and indeed there is still a 'region of transition' characterised by instability at $R_e > 4000$. These data are nonetheless significant because the flow can be turbulent even if R_e is much smaller since it is affected by viscosity, speed, the cross-section and internal roughness of the hose.

Having said that, a number of remarks about the internal roughness and the circular, elliptical, square, rectangular cross-section of the hose should account for the fact that transitions from a type of motion to another are different due to geometrical reasons (for instance, the transition is circular cross-sections occurs when R_e is about 800, whereas it occurs between 100 and 400 in bottlenecks with a sharp bend). Some claim that if the fluid is likely to be in a region of transition it is advisable to bring it to a verified level of turbulent flow by increasing speed and considering hose dimensioning carefully, but others believe this entails substantial pressure drops. Other results can be achieved by using the complex diagram known as 'Nikuradse diagram'; however, since complete definitions are rather complex, we are going to outline flow condition briefly as follows:

- ✓ Laminar flow in suction and discharge hoses and meatus;
- ✓ Turbulent flow in high-pressure hoses and, with a few exceptions, opening of directional and control valves.

Internal resistances

In oil hydraulic circuits, as well as in other types of circuits (pneumatic, electric, thermal, etc.), the flowing fluid encounters some resistance due to bottlenecks, bends and hose roughness. The word 'roughness' means the type of hose internal unevenness and it is the ration between the average heights of the bumps on the internal hose surface and its bore (Figure 1.23).

$$\text{Roughness} = \frac{\text{average roughness height}}{\Phi \text{ bore}}$$

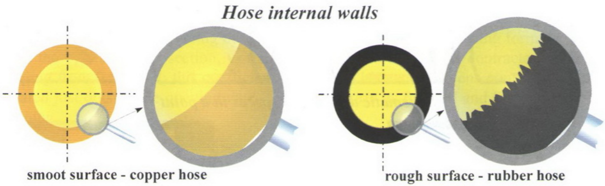


Figure 1.23

Glass, copper, brass hoses and hoses made of some types of steel are known as **smooth** hoses because of their low roughness, while hoses made of other types of steel and especially rubber hoses have a high level of roughness that causes pressure drops.

The resistance the fluid faces is knows as 'distributed pressure drops' in straight hoses and gentle bends and as 'localised pressure drops' in hose constrictions, sharp bends, etc.

Widespread pressure drops

Widespread pressure drops in straight hoses are mainly affected by two factors: internal friction of fluid molecules, that is viscosity (see chapter 2), to a greater extent and hose roughness to a lesser extent, as said above.

Pressure drop is a decrease in pressure due to friction: pressure at the end of the hose is less than the pressure the pump applies; their difference is pressure energy that transformed into heat.

Widespread pressure drops can be limited by using low viscosity fluids, adequate flow speeds and bores. Yet, the choice of fluid viscosity in a system can be influenced by internal hose resistance only to a limited extent: since many oil hydraulic components are designed and built so as to be operated only with a very limited viscosity range, an increase or decrease in viscosity would entail malfunctions the system cannot endure.

For instance, take into consideration the case of an axial piston pump that requires a fluid with a viscosity range of 30-80 centistokes; a lower viscosity would bring about excessive leakages and pressure drops, while a higher viscosity would entail excessive stress on the prime mover motor, to name but one drawback.

Although viscosity is the main cause for widespread pressure drops, hose dimensioning and speed too play a role in it. As a matter of fact, viscosity is inversely proportional to temperature, hence a small cross-section and excessive roughness increase pressure drops; pressure transforms into heat that heats up hoses, thus inevitably affecting viscosity.

As said above, it is difficult to calculate widespread pressure drops because complex mathematical formulas are required and problems about laminar or turbulent flow, viscosity, roughness and cross-sections should be taken into; in addition, all these remarks go beyond the practical aspects this book deals with. Enough information on how to dimension hoses adequately is provided throughout this book and especially in the chapter on hoses; we are going to stress more than one that it is essential to buy each type of element from reputable manufacturers whose catalogues also include tables, monograms and useful experience-based advice that help dispel any doubts.

Localised pressure drops

Localised pressure drops are characterised by a decrease in pressure like widespread pressure drops, but they are caused by trajectory changes, abrupt cross-section narrowing or widening that entail turbulence in the liquid (Figure 1.24).

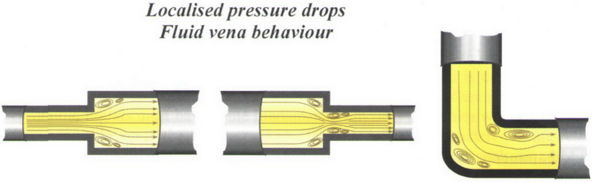


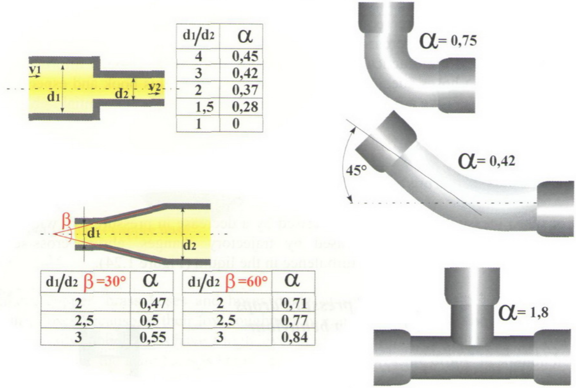
Figure 1.24

These drops in oil hydraulic systems result from the increase/decrease in hose bores, bends, connections or any other component like valves, actuators and filters. Like in widespread pressure drops, viscosity plays a major role and the remarks on widespread pressure drops also apply to localised pressure drops.
If fluid speed v (m/s), density ρ in kg/dm^3 (dm^3 is used for bar compatibility) and the coefficient α are known, pressure drop Δp can be calculated with the following formula:

$$\Delta p = \frac{\alpha \bullet v^2 \bullet \rho}{200} \text{ (bar)}$$

The coefficient α varies depending on the type of component that entails the pressure drop. Figure 1.25 shows the coefficients of the main units employed. As far as narrowing is concerned, α is influenced by the ratio of the two bores; as for widening, the inclination angle is taken into account along with the ratio of the two bores.

Localised pressure drops - Coefficient α



d_1/d_2	α
4	0,45
3	0,42
2	0,37
1,5	0,28
1	0

d_1/d_2 β=30°	α
2	0,47
2,5	0,5
3	0,55

d_1/d_2 β=60°	α
2	0,71
2,5	0,77
3	0,84

Figure 1.25

Fluid hammer

If a valve in a pressurised circuit is closed suddenly, the so-called 'fluid hammer' occurs, which means an oscillating wave propagates throughout the fluid from the valve itself to the other closed end of the hose. Oscillation usually causes a vibration that can easily be recognised because of its sharp and fading. This phenomenon can easily be triggered by suddenly closing a ball valve of any home plumbing system. Oscillation is dealt with in the final part of chapter 16.

The vibrations fluid hammer causes can damage systems in the long run because of high oil hydraulic pressures; oscillation in self-propelled machines can even affect the human body, especially the backbone. As a result, this oscillation must be taken into account in the design of a system: the phenomenon magnitude must be estimated and the system must be equipped with inlet/outlet slow damping valves. Fluid hammer in proportional oil hydraulics can always be avoided by adequately slowing down flow rate transients thanks to valve design.

Chapter 2 focuses on other phenomena and characteristics all fluids share but that are specific to oil hydraulics, like viscosity, compressibility, etc.

Chapter 2

HYDRAULIC FLUIDS

In the previous chapter we dealt with problems about fluid statics and dynamics without elaborating on the characteristics they need in order to be employed in fluid power.

Even if it is true that most systems need mineral oil due to their peculiarities, we cannot neglect the fact that in some particular systems this fluid is unsuitable for the working process because of both its environmental impact and its flammability.

Hydraulic liquids must have some fundamental physical properties without which the system cannot be up to the working standards needed. These characteristics are related on the one side to some factors embedded in the liquid itself such as viscosity, compressibility, working temperature etc., and on the other to the positive response to physical phenomena that can occur in a hydraulic system like, for instance, cavitation.

In brief, under normal working conditions hydraulic fluids should possess good thermal conductivity, compatible viscosity, efficient transmission response, lubricant power, chemical stability and, as much as possible, safety characteristics such as a tendency to catch fire at a temperature as high as possible and non-toxicity for people and the environment.

On this point we have to specify that, besides water and biodegradable synthetic or vegetable-based fluids (the former is mainly employed for energy transmission in food and pharmaceutical industries, while the latter are still under-exploited), unfortunately the most used fluid power liquids are very polluting and harmful to health. Physiological and environmental risks can be reduced only through a design involving adequately measured pipes, appropriate seals, safe components, the positioning of drip trays under the machinery and all the measures needed to avoid occasional leakages, along with a targeted and correct maintenance.

The first part of this chapter is devoted to liquid properties related to the system working characteristics, while the second focuses on hydraulic fluids classification.

PROPERTIES OF HYDRAULIC FLUIDS

Under working conditions, the hydraulic fluid is subjected to physical phenomena related to its own characteristics. The considerable powers needed in the oil hydraulic filed affect its choice and its properties have to be therefore known in detail.

Viscosity

In the previous chapter we have already mentioned viscosity as a key factor in the pressure loss for laminar or transitory flow, while it does not affect turbulent flow. In terms of physics viscosity can be defined as the **inner friction** due to the motion of the particles that make up the liquid mass or, more plainly, the fluid resistance to flow.

A very easy test to empirically measure viscosity consists in pouring a liquid into a cylindrical container and putting any stick, like a match or a toothpick (a spoon would stir the liquid too turbulently) perpendicularly in the middle of the container. Hold the stick while rotating your hand to describe a circle with as small a radius as you can; the rotation must occur in the centre of the cylinder. The liquid will start revolving and then the rotation will gradually expand on the whole surface forming a whirlpool. In other words, the molecules stirred in the middle progressively involve peripheral molecules in the turbulent motion.

This phenomenon takes place very quickly in a very fluid liquid like water, while it takes longer in a very viscous liquid such as chocolate, paint, oil, etc. Viscosity is thus inversely proportional to liquid fluidity: the more the former is, the less the latter is.

Friction between the molecules of a liquid in motion

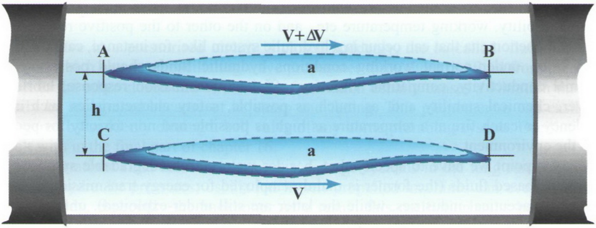


Figure 2.1

In order to simplify it and to avoid phenomena related to centrifugal force and peripheral speed involved in the previous case, this phenomenon can be described taking into consideration a liquid mass flowing in a hose and two parallel plane elements (AB and CD) of a surface (a) at a (h) distance (Figure 2.1). The two elements will flow at different speeds (V and V+ΔV) and the quicker one (V+ΔV) will drag the slower one (V) while the latter will tend to slow the former.

The friction of molecules, that is the resistance between the threads, results in an F force Isaac Newton defined with the following formula:

$$F = \mu \bullet a \bullet \frac{\Delta V}{h}$$

The **μ** (read mu) coefficient depends on the liquid nature and it is referred to as **dynamic absolute viscosity**; with regard to SI units of measurement it will result in:

$$\mu = \frac{F \bullet h}{a \bullet \Delta V} = \frac{N \bullet m}{m^2 \bullet m/s} = \frac{N}{m^2} \bullet s \quad \text{or Pa} \bullet \text{s (pascal} \bullet \text{second)}$$

In general, dynamic viscosity is still measured in *centipoise* (**cP**) according to the CGS system (centimetre, gramme, second):

$$1\ cP = 10^{-3}\ N \bullet s/m^2$$

In practice, viscosity is related to the mass per volume on the liquid at issue. The ratio between μ dynamic viscosity and ρ mass per volume is **ν kinematic viscosity** (ν: Greek consonant, read ni):

$$\nu = \frac{\mu}{\rho}\ (m^2/s)$$

Since the relation between the force in newtons and the mass in kg is:

$$1N = 1kg \bullet m \bullet s^{-2}$$

kinematic viscosity is:

$$\nu = \frac{\mu}{\rho} = \frac{N/m^2 \bullet s}{kg/m^3} = \frac{kg \bullet m \bullet s^{-2}}{m^2} \bullet s \bullet \frac{m^3}{kg} = m \bullet s^{-2} \bullet s \bullet m = m^2/s$$

Like dynamic viscosity, kinematic viscosity too is still measured according to the CGS system in *centiStokes* (**cSt**), submultiple of Stokes (St):

$$1\ St = 10^{-4}\ m^2/s = 1\ cm^2/s \qquad 1\ cSt = 10^{-6}\ m^2/s = 1\ mm^2/s$$

At present, viscosity is determined by measuring the flow time of a fluid through a **calibrated capillary tube**. This apparatus is calibrated by means of a standard liquid whose viscosity is known; a constant is thus obtained. Flow times measured in seconds are multiplied by the constants to establish kinematic viscosity in mm^2/s.

A method once used to measure viscosity and replaced by the above system consists in using a **viscometer**, an instrument made up of a container with a calibrated orifice on the bottom (Figure 2.2). The fluid (f), poured into the container (a), is kept at a constant temperature by the liquid (g) in the tank (b), which is heated by an electric resistance (c) controlled by a thermostat (t). By opening the valve (d) of the calibrated orifice (e), the quantity of fluid that runs out in a certain time is gathered in a tube (p). In this process the viscosity grade is measured in Engler, Saybolt or Reedwood units depending on the type of viscometer employed.

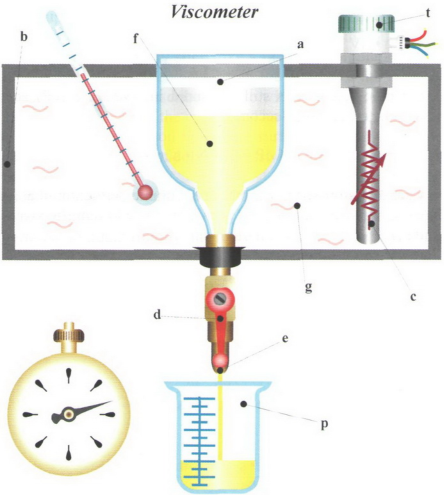


Figure 2.2

The term of comparison is the viscosity of distilled water, which is nearly constant even if the temperature varies. If water viscosity is set to 1° (E – SUS – RJ), viscosity grade is the ratio between the time in seconds water takes to run out and the same value for the fluid at issue.

The three types of viscometer differ only in the quantity of fluid running out. European *Engler* (**°E**) viscometer measures 200 cm^3 of fluid, U.S. *Saybolt Universal*

Seconds (**°SUS** or **°SSU**) viscometer 60 cm^3 whilst English *Reedwood* (**°RJ**) viscometer 50 cm^3.

For instance, if 200 cm^3 of a particular oil run out ten times as slowly as the same volume of water, Engler viscosity equals 10 °E.

As temperature is a key factor for viscosity grade, measurements are normally carried out first at a constant 20° C temperature and then at higher temperatures; the result is thus expressed adding the corresponding temperature, for example: 10 °E/50 °C, or 2.8 °E/90 °C.

Fluid viscosity at 20 °C is higher than viscosity at higher temperatures. In oil hydraulics a normal oil can have the following viscosities according to its temperature: 25 °E at 20 °C → 6 °E at 50 °C → 1.8 °E at 100 °C.

The following experimental formula can be employed to convert Engler units into centiStokes (mm^2/s):

$$\nu\,(cSt) = 7.6 \cdot °E \cdot (1 - 1/°E^3)$$

For example 2 °E equal $7.6 \cdot 2 \cdot (1 - 1/2^3) = 15.2 \cdot (0.875) = 13.3$ cSt, or $9\ °E = 7.6 \cdot 9 \cdot (1 - 1/9^3) = 68.4 \cdot (0.9986) = 68.3$ cSt.

In order to quickly determine a normal oil hydraulic viscosity, the formula can be simplified as:

$$\nu\,(cSt) = 7.6 \cdot °E\ (mm^2/s)$$

Given ν kinematic viscosity in mm^2/s or cSt, we can convert it into μ dynamic viscosity through the formula:

$$\mu = \nu \cdot \rho$$

Let's sum up the different conversions between °E, cSt and cP assuming, for instance, an oil with a mass per volume of $\rho = 910\ kg/m^3$ and a kinematic viscosity of 12 °E which becomes in cSt:

$$\nu = 7.6 \cdot 12 \cdot (1 - 1/12^3) = 91.2 \cdot (0.99) = 90.3\ mm^2/s$$

and converting into dynamic viscosity μ ($N \cdot s/m^2$):

since $1\ cSt = 10^{-6}\ m^2/s$ e $1\ cP = 10^{-3}\ N \cdot s/m^2$:

$$\mu = \nu \cdot \rho = 90.3 \cdot 10^{-6} \cdot 910 = 0.082\ N \cdot s/m^2 = 82\ cP$$

The following table draws an approximate comparison between the three measurements with the viscometer (°E, °SUS, °RJ) and their respective cSt.

VISCOMETER – COMPARISON BETWEEN MEASUREMENTS			
Engler (°E)	***Saybolt (°SUS)***	***Reedwood (°RJ)***	***CentiStokes (cSt)***
2	65	57	13 mm^2/s
3	102	88	22 mm^2/s
4	135	120	30 mm^2/s
5	175	150	38 mm^2/s
7	245	230	53 mm^2/s
10	350	310	76 mm^2/s
15	520	460	115 mm^2/s
20	700	610	150 mm^2/s
30	1050	920	230 mm^2/s
50	1750	1550	380 mm^2/s
70	2450	2150	530 mm^2/s
100	3500	3100	760 mm^2/s

The variation of kinematic viscosity according to temperature can be calculated with good accuracy by means of the formula tested by Carlo Rozzi De Hieronymis:

$$\log \nu = a\,T^2 + b\,T + c$$

where T = temperature in °K; a, b, c are three reading points (for instance 273, 313, 473 °K).

Viscosity Index

Viscosity Index (VI) is a parameter that simply relates the change in viscosity in a fluid with temperature: the higher VI is, the less the change is (Figure 2.3). In the case of mineral oil, nowadays refining has reached such high standards that there are no mineral bases available with VI lower than 95 – 100. Consequently it is necessary to have at one's disposal stable enough hydraulic fluids, that is fluids that are able to limit viscosity changes also when heat conditions are critical. These conditions can be preserved by adding some particular **additives** (polymers) to the lubricant bases obtained from petroleum distillation; these additives improve VI by restricting viscosity change in an acceptable range.

In practice, a low-VI oil can be improved (reaching 100 – 105) by adding an adequate quantity of additive. However, it is crucial to make clear that the resulting mixture deteriorates under high pressure and heat stress, thus foiling the additive effectiveness. As a result, systems must be provided with an oil whose VI is 100, that is an oil resulting from the most advanced refining processes. Oils with additives are used for controls or other systems subjected to low-medium pressures, with a limited temperature range, little leakage and intermittent duty, that is to say non-subjected to long-term pressure.

In addition, *as pressure increases, so does viscosity* but this phenomena occurs especially above 300 bar and in general it can be deemed negligible.

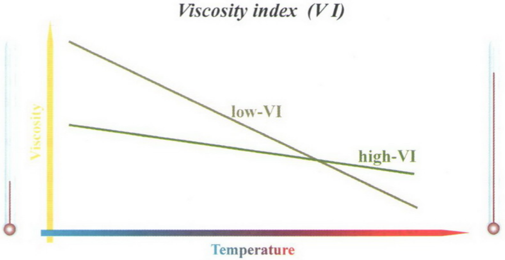


Figure 2.3

Viscosity Grade standards

Under ISO standards, the kinematic viscosity grade of fluids is expressed in cSt (mm^2/s), measurements shall be performed at a temperature of 0 °C, 40 °C, 100 °C and in any case viscosity at 40 °C must be reported with the number contained in the code. The types of oil that are most employed in fluid power are those ranging from ISO VG 22 to ISO VG 68.

ISO – KINEMATIC VISCOSITY CLASS ν (mm^2/s or cSt)			
Class	***ν max(0 °C)***	***ν medium(40 °C) and min-max limit***	***ν min(100 °C)***
ISO VG 10	90	10 (9÷11)	2.4
ISO VG 22	300	22 (19.8÷24.2)	4.1
ISO VG 32	420	32 (28.8÷35.2)	5
ISO VG 46	780	46 (41.1÷50.6)	6.1
ISO VG 68	1400	68 (61.2÷74.8)	7.8
ISO VG 100	2560	100 (90÷110)	9.9

The Figure 2.4 chart expands the previous ISO VG table because it shows the whole trend of oil kinematic viscosity (note that viscosity lines look straight because the scale employed is logarithmic. Actually, changes are extremely fast up to 20 °C while they tend to stabilise on low values as temperature increases).

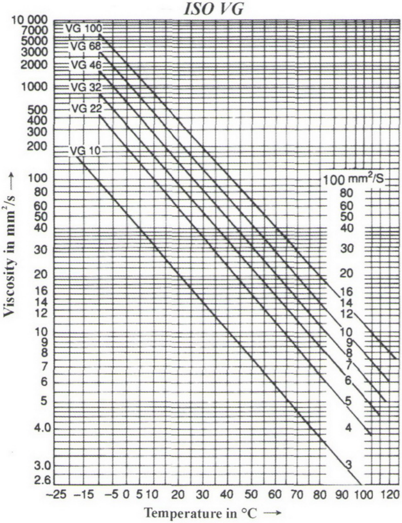


Figure 2.4

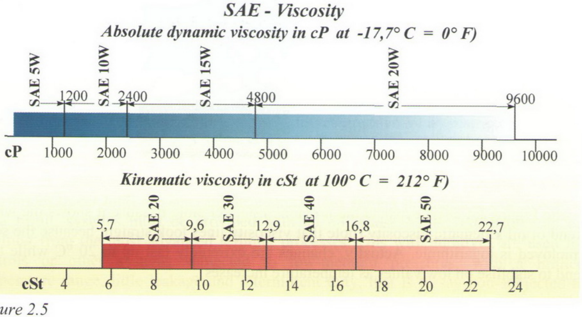


Figure 2.5

The Society of Automotive Engineers (SAE) established viscosity ranges at specific temperatures, identifying them with proper codes. The number followed by the letter W stands for μ absolute dynamic viscosity at 0 °F (-17.7 °C) while ν kinematic viscosity is calculated at 212 °F (100 °C).

Note that this classification concerns multigrade oils employed in the automotive industry (gear, motor oils, etc.).

According to the table (Figure 2.5), we can establish for instance that a SAE 15W-20 oil has a dynamic viscosity ranging from 2400 to 4800 cP and a kinematic viscosity at 100 °C ranging from 5.7 to 9.6 cSt.

Influence of viscosity in oil hydraulic systems

In oil hydraulic systems the fluid is pumped throughout the whole system and it also lubricates mechanical parts internally. Unsuitable viscosity can seriously undermine system performances. Let's detail the problems that could occur.

High viscosity (very dense fluid)

- Fluid resistance to flow resulting into pressure drop.
- Slowing of controls and mechanical movements.
- Pump cavitations.
- No or inefficient air/oil separation in the tank.
- Energy loss due to increased friction and higher torque of prime mover for the higher effort the pump requires.
- Increased temperature because of the power loss due to higher friction.

Low viscosity (not very dense fluid)

- Drop in η_v volumetric efficiency of the pump due to increased internal leakage (in gear or vane pumps between them and the stator, in radial or axial piston pumps between them and cylinders and in the distribution unit).
- Increased internal leakages in the whole system resulting in a rise in temperature.
- Excessive wear and seizing especially of the pump due to the fluid that is not dense enough for an adequate lubrication.
- Leakage of actuator cylinders promotes air inlet causing power loss and movement lags.

Compressibility

While presenting the main hydraulics notions, we considered liquids as totally incompressible elements; actually, any fluid subjected to pressure is compressible.

In respect to fluids employed in oil hydraulics, we can empirically establish that this compressibility is inversely proportional to mass per volume: it is higher for oil, less considerable for water and far less considerable for synthetic fluids.

Compressibility reduces the initial liquid volume to a negligible extent in simple and low-pressure systems but it is quite relevant in the design of large cylinders and in any system with large high-pressure capacities.

For oil we can observe a 0.7% volume reduction at a pressure of 100 bar neglecting temperature, kinematic viscosity and the type of oil, all of which affect this phenomenon, though to a lesser extent. If we neglect what is stated above, volume change (ΔV) is established with the following formula:

$$\Delta V = \beta \cdot V_0 \cdot (p_1 - p_0)$$

$$\beta = 10^{-4}\ cm^2/daN = 10^{-4} \cdot bar^{-1}$$

where β is the compressibility coefficient, V_0 the volume at rest, p_0 and p_1 respectively initial and final pressure.

The following table shows the trend of the β coefficient in respect of pressure change.

MINERAL OIL COMPRESSIBILITY	
Pressure (bar)	***β (10^{-4} cm^2/daN)***
100	0.75
200	0.70
300	0.65
400	0.61
600	0.56
800	0.53

If high pressure is not required, the following volume changes at 100 bar are valid:

COMPRESSIBILITY–VOLUME CHANGE	
Fluid	***Δp = 100 bar***
Mineral oil	$\Delta V = 0.7\%$
Vegetable-based oil	$\Delta V = 0.5\%$
Water and emulsified water-oil	$\Delta V = 0.4\%$
Water-Glycol and synthetic fluids (polymers)	$\Delta V = 0.35\%$

For instance, if oil at atmospheric pressure in a 400 litres (dm^3) capacity is brought to a pressure of 250 bar, volume change is:

$$\Delta V = V_{initial} \cdot 0.7\% \cdot p/100 = 400 \cdot (0.7/100) \cdot (250/100) = 400 \cdot 0.007 \cdot 2.5 = 7 \text{ litres}$$

In practical terms, volume loss affects working time. As a matter of fact, if we assume to use a pump with a flow of Q = 10 l/min and a working pressure of 300 bar with $\beta = 0.65 \cdot 10^{-4}$ to increase the pressure of the oil contained in a large cylinder with a capacity of V = 70 litres, volume reduction is:

$$\Delta V = \beta \cdot V \cdot p = 0.65 \cdot 10^{-4} \cdot 70 \cdot 300 = 13650 \cdot 10^{-4} = 1.4 \text{ litres}$$

Since pump flow is 10 l/min, the time needed to bring the oil in the cylinder to 300 bar is the ratio between ΔV and Q:

$$t = \frac{\Delta V}{Q} = \frac{1.4 \quad l}{10 \quad l/min} = 0.14 \text{ minutes} \qquad 0.14 \cdot 60 = 8.4 \text{ seconds}$$

This means those 8.4 seconds are an inevitable dead time that deeply affects production cycles.

If this cylinder was the main actuator of a press and the production required two cycles a minute including the dead time for controls, rod descent and ascent, loading and unloading of the piece under production, structure elasticity, the delay would equal 16.8 seconds including compression time [60 + (8.4 • 2)]; the 120 pieces a hour should instead be processed in (16.8 • 60 = 1008 s) about an hour and 17 minutes.

Even if it is not possible to act on compressibility, time can be reduced by replacing the pump with a higher flow one. In the previous case, a pump with a flow of Q = 50 l/min would cut time to about an hour and four minutes.

Inclusion of air and vapours

Now we will briefly mention the phenomena of cavitation and inclusion of air in suction, which cannot be considered as their fluid properties, although they are closely linked to fluid performance. These points will be further discussed in the chapter devoted to hydraulic pumps.

The two phenomena, which differ from each other only in their initial formation process, cause inefficiency or even the collapse of the pump.

In both of them the phenomena occur in the suction hose for the presence of vapour, gas or air bubbles which mingle with the fluid and enter the pump compression area causing corrosion between moving parts and the stator.

Corrosion progressively produces pitting that reduces the performance of the hydraulic generator, increased noise and vibrations and finally mechanical seizing.

By **inclusion of air** we mean the inclusion of air bubbles in the suction fluid arising from neglected connections and especially from the air/oil foam that could be in the tank and that was produced during unloading. In **cavitation** the bubbles of the suction phase result from the vapour tension of fluid and dissolved gas.

Vapour tension

The boiling point with the subsequent transformation of any liquid into vapour depends on the ratio between a specific temperature and the pressure on its free surface.

It is common knowledge that water boiling temperature is 100 °C but this parameter is valid only and exclusively if atmospheric pressure equals 1.02 bar (sea level). As pressure decreases, water transforms into vapour at a lower temperature so that, for example, the boiling point at a pressure of 0.02 bar is around 15 ÷ 20 °C.

Consequently, vapour tension reflects the transformation point of the liquid into gas with reference to the pressure on the free surface and the temperature, or rather the absolute pressure at which the liquid boils at a set temperature.

For hydraulic fluids such as mineral oil and synthetic ones, vapour tension is far higher than it is for water and this allows to reach considerable suction depressures.

This and the lubrication problem are the main obstacles to the use of water in fluid power: low vapour pressure does not promote high-level suction depressures and limits the use of H_2O to few applications.

Density

Synonym of *mass per volume* (see previous chapter), the density of a fluid is expressed by the mass of 1 m^3 at the temperature of 20 °C and it is compared to the equivalent mass of distilled water, both of them at atmospheric pressure.

The following table shows the *average* mass per volume (or density) of the main hydraulic fluids.

DENSITY OF HYDRAULIC FLUIDS

Fluid	***Mass per volume (kg/m³)***
Mineral oil	870 ÷ 900
Water	1000
Water/Glycol	1060
Water/Emulsifiable oil	920 ÷ 940
Vegetable-based oil	930
Chlorinated hydrocarbons	1400
Phosphoric esters	1150
Silicones	930 ÷ 1030

It is important to note that fluids are bought by weight, but hydraulic parameters like flow etc. are calculated in volume; consequently, density value has to be related to the volume that is suitable for that particular system.

Lubricant power

In a moving component, sliding metal parts should have totally smooth surfaces to avoid wearing due to friction, which is impossible unless at exorbitant costs.

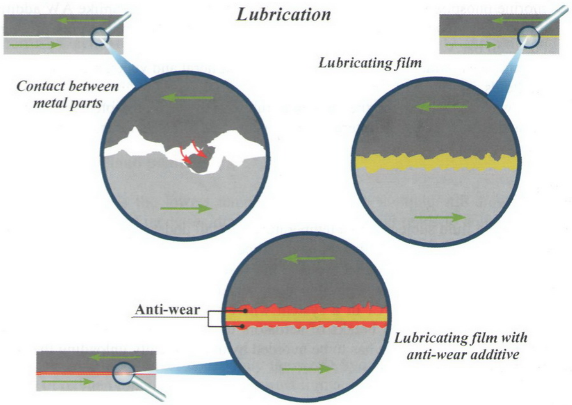


Figure 2.6

The lubricant power of a fluid is its ability to get through and stay between moving parts, that is to create a film that prevents physical contact between two sliding parts.

To explain the phenomenon of wearing clearly, let's imagine to zoom in on an area between two moving parts: microscopic flaws, which are also found in machined surfaces with close tolerance, come into contact during the motion, so they tend to detach and then weld in different points due to local high temperatures; this phenomena continuously expand until parts seize up. As the lubricant film prevents metal contact, it helps maintaining organs perfectly efficient (Figure 2.6).

The ideal condition consists in a very wide slot between the parts so that any lubricant fluid, even a very viscous one, can get through and form the film. However, this condition can occur at a low-medium pressure while a higher-pressure demand would make leakages soar dramatically entailing substantial energy loss.

There is therefore a limit (known as 'boundary lubrication') beyond which the conditions above occur: leakages take place above it (wide slots) whilst below it the film limit is too subtle and metal unevenesses enter into contact and break up. With the strict conditions of boundary lubrication, overpressure and pressure drops occur respectively in the narrowest and largest areas of unevenesses, with the risk to break the lubricant film and subsequently wear mechanic parts due to an increase in temperature.

The natural lubricant property of the fluid under such conditions can be improved by adding AW ("Anti-wear") or EP (" Extreme pressure") additives.

Zinc/phosphorus-based additives are the most common AW additives; since they form a film on sliding parts, they prevent previously detached particles from adhering.

Zinc/chlorine/phosphorus-based EP additives too stop welding but, unlike AW additives that start acting as temperature increases, they perform their action at a pressure higher than 200 – 250 bar.

EP and anti-wear additives can be mix with both mineral and vegetable-based oils and synthetic fluids.

Finally, we cannot neglect the fact that lubrication also promotes the sliding of dynamic seal parts, that is the internal components that slide on seals, which would be short-lived without lubrication.

Dilation

A free surface fluid subjected to a temperature rise/fall undergoes a volume change. If the fluid is inside a sealed container and temperature rises, it cannot expand, so it compresses itself with a subsequent pressure increase.

This phenomena in oil is not negligible at all because pressure goes up by 9 bar when temperature increases by 1 °C. In a system where temperature increases for inherent reasons during the prolonged breaks of actuators under high pressure, compressibility resulting from this phenomena has to be avoided by automatically unloading the surplus volume in an accumulator.

Antifoaming properties

In a hydraulic circuit the release phase is very likely to produce a foam made up of bubbles of air or another gas trapped in the fluid. The foam settles in the tank and it is sucked by the pump and pumped throughout the pressurised system causing increased noise and vibrations; yet, the widespread compressibility of the gas reduces working pressure.

Some special silicone-based additives weaken fluid surface tension, thus promoting bubble suppression promptly.

Demulsibility

This term refers to the fluid property to separate from or repel water.

The main reason why water is found in the system is the condensation of the air humidity (dew point) found inside the upper part of the tank. Demulsibility properties of hydraulic fluids often need to be developed by adding specific additives.

It is however a good idea to plan cyclic interventions to remove water from the tank. In systems where the fluid is oil, this is very easy: since water density is higher than oil density, water settles at the bottom and the whole unwanted volume can be unloaded by opening the unloading valve at the bottom.

Fluid active life

Certainly every fluid, including fluids with an excellent composition, has like any elements its own natural life beyond which its performances decline progressively. Yet, the active life of a hydraulic fluid is undermined by a series of factors like inclusion of air, water, metal particles and working conditions. Contamination entails the formation of corrosive acid substances and deposits, an increase in viscosity and very bad heat transmission.

Adequate additives ensures a prolonged "performance life" of the fluid, but large capacity tanks, which means tanks fit to contain substantial fluid volumes and a substitution of it at regular intervals, further guarantee system regularity.

Pour point and Cloud point

The Pour point is the lowest temperature at which a liquid still flows. At lower temperatures viscosity increases dramatically and the fluid tends to solidify. The fluid becomes opalescent before solidification: this phase is known as Cloud point.

"Pour point depressant" additive lowers the Pour point while it does not affect the Cloud point.

Oxidation and rust formation

As far as oxidation, water is certainly the worst hydraulic fluid because the massive quantity of oxygen found in it develops a rust formation that is often intolerable.

Water and air inevitably enters circuits through faulty seals in mineral oil systems.

Oxidation already starts at low temperatures on the free surface of the tank in vegetable seed oil: good results can be achieved if nitrogen is pumped into the tank.

In order to prevent mineral oil from oxidising, it must be mixed with antioxidant additives.

The oxidation that took place in the fluid entails the progressive degradation of oil, which loses its film properties with subsequent rust formation in the different internal parts of the components with higher leakages, slow responses and finally seizing. "Antirust inhibitors" (antirust additives) settle on the surfaces of internal components and form a film that protects the metal from oxidant agents.

Fire resistance

Fire prevention is an important problem concerning systems both at work and at rest. For example, consider the effects of the spouts from a broken hose of pressurised oil that hits a high-temperature surface or, for static conditions, the flammable potential of the substantial fluid volumes that are found in the tanks.

At least in this case, water and the fluids derived from it prove to be better because there is no fire risk. Mineral oil has instead remarkable flammable properties, with liquid Fire point and vapour Flash point of about 150 – 180 °C (as viscosity increases so does inflammability). The situation is so much better for synthetic fluids and vegetable oil that they are often (wrongly) classified as non-flammable fluids; it is more appropriate to define them as "fire-resistant fluids". The following table shows the fire point, flash point and spontaneous combustion of some hydraulic fluids.

FIRE RESISTANCE			
Liquid	Fire point (°C)	Flash point (°C)	Spontaneous combustion (°C)
Mineral oil	180	150	245
Phosphoric ester	330	310	610
Chlorinated hydrocarbon	400	380	650
Silicone	335	285	480

HYDRAULIC FLUIDS CLASSIFICATION

The previous paragraph makes us understand that fluids found in oil hydraulics (Figure 2.7) can be classified as:

1) Oil obtained from hydrocarbons (mineral oil)
2) Water
3) Mixtures with water
4) Synthetic fluids
5) Biodegradable oils obtained from vegetable seeds

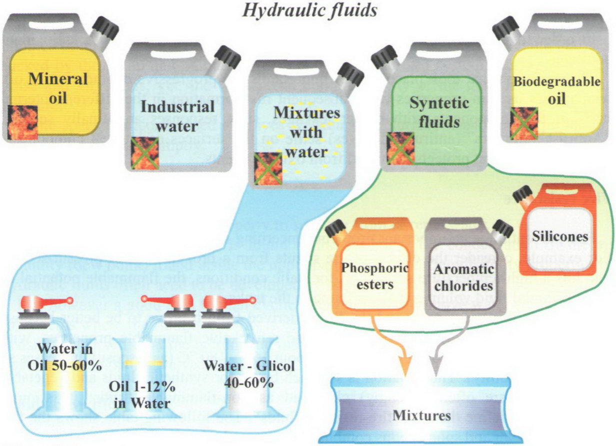


Figure 2.7

Mineral oil

Nowadays it is the most common hydraulic fluid thanks to its good lubricant properties, its remarkable viscosity index (VI), its reasonable price in comparison to synthetic fluids, its easy availability and its limited toxicity. Additives can be added to it to meet the system requirements and they preserve its properties for a long time without causing significant deposits, corrosive effects and evaporation. It is a good heat conductor, compatible with most elastomers and its contact with paints does not entail particular problems. On the contrary, its inflammability makes it the worst hydraulic fluid vis-à-vis fire resistance. The use of additives to improve its viscosity index (the more VI is, the less viscosity is in respect to low temperatures) and others like anti-wear, antirust, antioxidant products should be planned carefully so as not to run the risk of early depletion at high working temperatures.

Although mineral oil can be employ at rather higher temperatures in other technologies, in hydraulic systems it is advisable not to exceed 70 – 75 °C and in any case never to exceed 100 °C for peaks due to heat dissipation.

The German standardisation board (DIN) and later the ISO classified mineral oils according to the formula of the additives they contain (table follows). Since ISO standards are quite recent, sometimes the code found on available drums refers to the similar DIN.

Classification of mineral oils according to the additives they contain

DIN (code)	**Additives**	**ISO** (code)
H	none	HH
HL	Antirust + antioxidant + antifoam	HL
HLP	Antirust + antioxidant + antifoam + anti-wear	HM
HLP–D	HLP + Detergent + dispersant	not envisaged
not envisaged	HM + additives to improve VI viscosity index	HV

As shown in the table above, ISO HH (DIN H) oil contains no additives. For optimal results it is necessary it is a high-quality fluid (solvent-refined paraffinic oils) and it is employed in elementary, non-continuously working equipments at working temperatures below 45 °C; for instance, HH oil is often used for manual pumps.

ISO HL class is suitable for oil hydraulic systems that are not subjected to remarkable wear like, for example, the systems made up of gear or vane pumps subjected to limited pressures. The use of HM class oils substantially reduces the wear between sliding metal parts, especially pumps and motors. VI viscosity index has to be excellent where external ranges of temperature are rather high (open-air stationary systems, self-propelled machines, ship deck equipment). ISO HV oil is therefore the ideal mineral fluid under these circumstances.

In addition, some special machine tools require HG oils (multifunctional lubricants); the hydraulic oil employed in various self-propelled machines is the same as the one used for gear lubrication (under SAE).

WORKING RANGE OF TEMPERATURE	
Starting	***Maximum***
-23 °C (-10 °F)	54 °C (130 °F)
-18 °C (0 °F)	83 °C (180 °F)
-18 °C (0 °F)	99 °C (210 °F)
+10 °C (50 °F)	99 °C (210 °F)

The table above shows minimal starting and maximum working temperatures in stationary or mobile oil hydraulic systems.

Water

The most economical fluid that can be used in oil hydraulics is certainly water purified from lees, rust, metal particles, etc. by means of decantation and filtration.

Components for applications with water and mixtures with water

Figure 2.8

As it is the only totally non-flammable liquid but it has none of the fundamental lubricant and antioxidant properties, water use is confined to particular applications like in the food and pharmaceutical industries where the toxicity of other fluids causes serious physiological damage.

Water is perfectly suitable for high-pressure tests for hoses intended for drinking use because of its antipollution properties and its low compressibility ($\Delta V = 0.4\%$, which is lower than oil compressibility). It is no wonder these tests can be performed at pressures above 800 bar: indeed, pressure multipliers are used because water volumetric pumps cannot sustain high pressures.

At present there are excellent systems that use water as a fluid and that are equipped with bronze, stainless steel and other rust-resistant materials (Figure 2.8) on the market.

Mixtures with water

Mixtures with water and lubricant fluid can be employed in systems where fire risk is rather high if synthetic fluids are to be avoided. These mixtures are classified as:

a) Oil-in-water (HF – A)
b) Water-in-oil (HF – B)
c) Water – Glycol (HF – C)

It is worth making clear that cyclic inspections of the tank are essential as far as mixture with water are concerned. The working at temperatures above the average causes much evaporation with the risk of considerable level decrease and unbalanced mixture ratios. In addition, oil tends to separate and float during long rest times: the pump sucks only water at starting.

By **oil-in-water** mixture we mean the adding of mollifiable oil to the overall water volume in a ratio ranging from 1% to 12%; this enhances the lubricant power and prevents open air systems from freezing. This mixture has therefore an excellent balance of viscosity and temperature, it does not attack elastomers and paints but its lubricant power is rather poor.

Water-in-oil emulsions are made up of a 50% – 60% ratio of mineral oil; additives are added into the resulting mixture in order to improve its properties.

This substance is not incompatible with paints and elastomers, has a good temperature-viscosity balance and is not much flammable; however, its lubricant power is lower than pure oil power. The system temperature must not exceed 65 °C.

Water – glycol mixture is made up of a glycol ratio ranging between 40% and 60% plus anti-wear, antirust, etc. additives.

Glycol [from the words glyc(erine) and (alch)ol] is chemically defined as "divalent aliphatic alcohol" (in organic chemistry 'aliphatic' refers to a compound in which carbon atoms are linked in an open chain) while glycerine, trivalent aliphatic alcohol, is a fundamental component in fat substances.

Water – glycol viscosity is slightly higher than the viscosity of water-in-oil mixture and it is not much flammable; However, it has a poor lubricant power and it is incompatible with zinced parts, Vulkollan seals and most paints except for epoxy and vinyl paints.

Synthetic fluids

Synthetic fluids, that is **phosphoric esters**, **aromatic chlorides**, **silicones** and **mixtures**, are used in systems with high fire risk.

They share some mineral oil properties like antioxidant and lubricating power, but they are expensive and incompatible with some elastomers and paints. They do not change up to a working temperature of 150 °C (350 °C for silicones); they are suitable for Teflon, nylon, butyl and silicone rubber seals and viscosity vis-à-vis pressure is slightly poor due to the density ranging from 1100 kg/m^3 to 1400 kg/m^3.

We already mentioned its high cost, which is four to eight times higher than mineral oil cost.

Biodegradable vegetable-based oil

Despite its properties, which are inferior to mineral oil ones, vegetable-based oil proves to be an excellent alternative to traditional fluids, especially in the mobile industry. It is better than mixtures with water, less expensive than synthetic liquids (but it possesses their main properties), compatible with paints and elastomers and it has an excellent anticorrosive power.

Obtained from soya or rape seeds, this fluid at 50 °C has a kinematic viscosity $\nu = 15 \div 70$ cSt, a fair VI viscosity index, excellent lubricant power, good heat stability, low volatility, a fair flash point, a lower compressibility than the homonymous mineral: $\Delta V = 0.5$ with $\Delta p = 100$ bar and the principal additives can be added to it.

However, in comparison to the oil obtained from hydrocarbons, vegetable-based oil has a poor hydrolytic stability (it is hydrolysed with water forming acid compounds) and lower antioxidant power; its pour point is lower than mineral oil and its cost is more than double.

Biodegradable synthetic-based oil

Synthetic ecological fluids are divided into **synthetic esters** and **polyglycols**. Overall, their properties are better than vegetable ones but they are limited in the applications due to their high cost (three to four times as much as mineral oil for polyglycols; eight times as much for synthetic esters).

Conclusions

We stress again that at present mineral oil is the best hydraulic fluid and perhaps biodegradable oil, if enhanced, will be the hydraulic fluid of tomorrow if we consider today's world just tendency to seek products that are less harmful to health and pollute less. We remind that available oil hydraulic components are generally defined in their code according to mineral oil.

If oil is replaced by a synthetic, water-based or biodegradable fluid in an existing or planned system, it is important to check fluid compatibility by reaching out to the manufacturers of *all* the components.

Except for food and pharmaceutical applications with water and biodegradable fluids, the choice of the synthetic fluid should be influenced only by fire problems; otherwise, the preference obviously goes to mineral oil because of its inexpensiveness and excellent properties.

The issues discussed in this chapter refer to the inherent properties of hydraulic fluids and the opportunities to improve them by adding additives. In order to have a complete figure it is absolutely necessary to take into account the problems of *contamination* (see chapter 13) of fluids themselves and *filtration* systems that remove impurities that entered the liquid in different manners during the working process.

The following table compares and sums up the main characteristics of hydraulic fluids.

Comparison between hydraulic fluids (* biodegradable fluids)

Fluid	Density (kg/m3)	Viscosity index (VI)	Lubricating power	Fire resistance	Antioxidising power	Min-max working temperature (°C)	Max temperature (°C)	Compatibility with standard elastomers	Compatibility with standard paints	Toxicity	Average cost
Mineral oil	from 870 to 900	From 70 to 100	Excellent	Poor	Excellent	Excellent	from -5 to 70	Excellent	Excellent	Low	100
*Vegetable oil	920	210	Excellent	Good	Fairly good	Fairly good	from -10 to 70	Excellent	Good	No toxicity	250
*Polyglycols	1100	from 150 to 200	Excellent	Good	Good	Good	from -30 to 80	Fairly good	Good	No toxicity	350
*Synthetic esters	920	200	Excellent	Good	Good	Good	from -30 to 100	Good	Good	No toxicity	700
Water-in-oil	from 915 to 940	High	Fairly good	Excellent	Good	Good	from 0 to 50	Excellent	Excellent	Low	200
Water-glycol	1060	High	Fairly good	Excellent	Good	Good	from 0 to 50	Poor	Poor	Low	400
Chlorinated hydrocarbons	1430	Low	Good	Good	Fairly good	Fairly good	from -5 to 70	Poor	Poor	High	700
Phosphoric esters	1270	Low	Excellent	Good	Fairly good	Fairly good	from -5 to 70	Poor	Poor	High	500
Mixtures of esters and chlorides	1150	Low	Excellent	Good	Fairly good	Fairly good	from -5 to 70	Poor	Poor	High	600
Silicones	From 930 to 1030	High	Fairly good	Excellent	Fairly good	Fairly good	from -5 to 90	Poor	Poor	Low	<700

Chapter 3

OIL HYDRAULIC PUMPS - GENERAL

The human heart is a pump. Pumps are the heart of the hydraulic system.

A fluid power closed circuit is very similar to blood circulation in human beings. Arteries and veins work like respectively delivery and discharge hoses…and filters too are found in both systems!

Blood moves substances to muscles, which act as human organic actuators, while hydraulic fluids move mechanical actuators. In addition, hydraulic systems with an on/off or proportional control are like the orders given by the brain with a bit of imagination: the lifting of a weight, food ingestion and ordinary physiological actions do not require major mental efforts and this also applies to the controls of a simple on/off system; actions involving an intellectual effort like writing, planning, painting, playing a musical instrument, performing an equation, repairing a motor, managing your family, etc. demand experience or studies; in other words, such an intellectual effort involves what the brain stored over time and a parallel with the data needed to manage an electroproportional system can be drawn.

We are not able to explain the analogy between a prime mover and what fuels and moves the heart because the heart is not a mechanical device resulting from technology but a marvellous gift, the mystery of life.

But now let's go back to fluid power…

INTRODUCTORY REMARKS

Pumps transform the mechanical energy the prime mover generates into hydraulic energy. The transmission from one part to the other is usually performed by their respective shafts connected via a flexible drive coupling that offsets misalignment; a gear over or reduction unit must often be interposed, especially in the case of endothermic motors. With a few exceptions, standard revolution speeds for three-phase electric motors are compatible with the direct connection to the pump; speed is usually 1400 or 2800 rpm, but if the system is equipped with an electronic device known as 'inverter', the number of revolutions can be adjusted from the minimum to the maximum limit.

As we stated in the introduction, it is important to note that, with a few exceptions, the revolution speed of the prime mover/pump, connected directly or via an adapter, has to be constant in oil hydraulics: as a result, flow, which depends on the number of revolutions, does not increase or decrease using speed changes, but it is affected by varying pump internal displacement using mechanical or automatic systems. Inexperienced readers should not be mislead by the fact that a sudden albeit temporary increase in the number of revolutions of a diesel motor occurs when a lifting hoist or a wheel loader are operated: at that very moment the hydraulic actuator needs more power (for instance, the excavator shovel faces a bigger obstacle), the revolution speed tends to drop and the mechanical device on the endothermic motor brings the system speed to a standard level by widening the supply for diesel fuel.

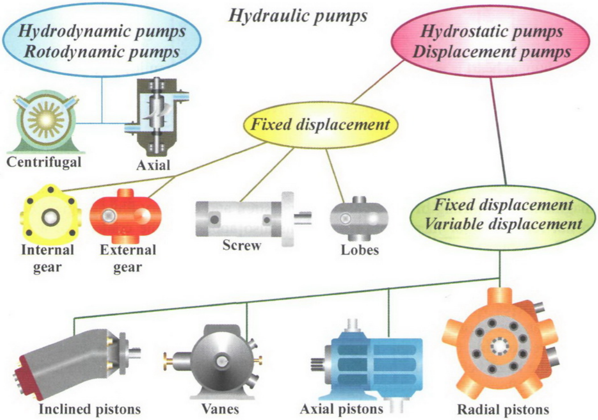


Figure 3.1

The pump per se is essentially a device that transfers the liquid: in other words it generates a flow, which can be measured as a given volume passing through per unit time; instead, pressure results from the fluid encountering an obstacle like a narrowing, the cylinder piston or motor vanes. *The pumps does not generate pressure but it should be designed as to sustain the pressure the system requires*; sentences like 'this pump generates a pressure of 150 bar' are very popular, actually it should be phrased as ' this pump *can sustain the overall resistance the fluid encounters* (pressure) in the whole circuit'.

Hydraulic pumps in general applications are actually either used to move a liquid that encounters almost no resistance, except for hose internal friction, or in pressurised systems.

The physical principle falls into hydrodynamics in the former case and hydrostatics in the latter case.

Hydrodynamic pumps – usually referred to as '**rotodynamic pumps**' – suck a liquid at rest and simply moves it to another point; *hydrostatic pumps* – or '**displacement pumps**' – suck a liquid from the tank too, but moves it to the actuator system. Displacement pumps not only should be designed so as to sustain the ensuing high pressures but its internal parts should also prevent leakages between suction and delivery or, in other words, the return of the fluid into the tank (Figure 3.1).

Prime mover

By 'prime mover' we mean the mechanical power generator connected to the pump.

Three-phase electric motor is the best generator in stationary oil hydraulics: it is light and quite economical, it can easily be operated by remote control, it has few moving parts (rotor with two bearings), it needs no refuelling and little maintenance, it stops in real time, it starts quite rapidly even if its initial electrical energy demand is substantial (this is its main drawback) and control parts (fuses, electromagnetic switches) need over-dimensioning.

Oil hydraulic systems in self-propelled machines are connected to the single diesel motor; both traction and actuators in some vehicles, like excavators, combine harvesters, etc. employ oil hydraulic circuits, whereas in most cases traction in other vehicles, like agricultural tractors, is mechanical and it is performed by a drive shaft and the lifting implements are connected to the hydraulic pump. The so-called 'external power take-off' of these tractors can be connected to another pump on the hydraulic circuit of external towed vehicle (like balers, seeders, mowers) or to the hydraulic system that raises the trailer dump body consisting of a pump and a telescopic cylinder. The prime mover for the oil hydraulic use of a road vehicle can be electric (direct current): sometimes a system of pump/diesel connection is too expensive, especially if hydraulic actuators are limited and do not require high performance; this applies for instance to the dump bodies of small vehicles or truck loading/unloading platform where the compact unit made up of tank, electric motor, pump and valves operates the actuating cylinder. The pump in heavy automotive vehicles is usually connected to the Diesel and the hydraulic circuit is often destined to the handling of the crane for general lifting, moving platforms or large container lifts; in some cases, an additional endothermic motor is employed only for oil hydraulic operation, for instance some revolving drums of concrete mixers (see chapter 20).

There are few low- and medium-powered self-propelled machines that have an electric system: batteries (direct current) powers the electric motor axially connected to the hydraulic pump. Finally, pneumatic motors too are suitable for some applications because they do not have starting problems and they are ideal for explosive environments.

Intermittent operation

Every system experiences operational and standby phases. As each phase alternates with the other in a relatively short time, strategies to limit wear, overheating, noise and energy costs should be devised.

Standby phases of the intermittent operation are performed putting the pump in standby, which means offsetting the delivery pressure by making the pump unload while the mechanical/hydraulic generator unit keeps on revolving (*vent* connection on the relief valve).

Obviously, if standby phases are rather long, it is advisable to stop the motor revolution (no precise data can be provided because each system has its own timing); it is clear that standby phases should be more spaced out in case of connections to internal-combustion motors. Pump standby by venting in self-propelled machines powered by electric batteries is viable only if the system does not deplete batteries in a short time.

Rotodynamic pumps

Unlike displacement pumps, rotodynamic pumps are designed so as to allow an adequate leakage between suction and delivery sections. The liquid already found in the stator (pre-filling must be performed each time if the pump is above the free surface) reaches a high speed by means of a rotor made up of an impeller; the resulting kinetic energy pushes the fluid into the delivery hose and the vacuum generated on the other hole allows fluid suction.

If the delivery hose narrows or is obstructed, the wide space between the stator and the impeller promotes the leakage of the liquid surplus that goes back to the tank without affecting the effort of the prime mover; yet, such an apparatus is the main cause for low volumetric efficiency.

Rotodynamic pumps are subdivided according the type of impeller into fixed vane *centrifugal* pumps (the liquid moves in a radial manner) and *propeller* pumps (the liquid moves in an axial manner) (Figure 3.2).

Unlike displacement pumps, a slight increase in pressure in rotodynamic pumps entails a substantial flow drop.

Rotodynamic pumps are simply used to move liquids like the transport of wine, oil, beverage, etc. in the food industry, water tapping for irrigation or the drainage of flooded rooms…

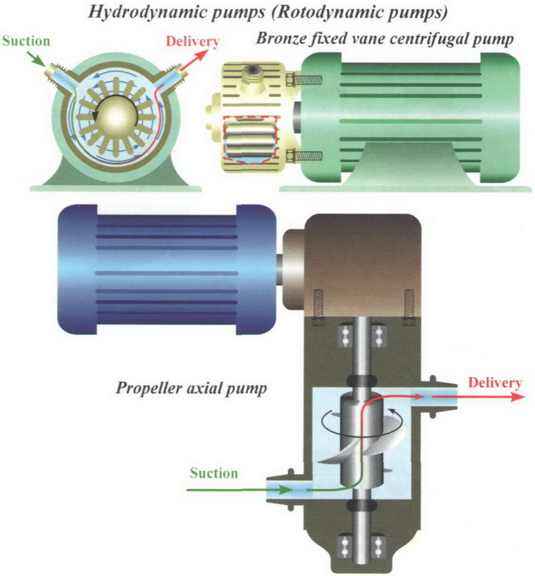


Figure 3.2

These pumps are employed in oil hydraulics only to boost displacement pumps subjected to cavitation.

A second group of rotodynamic pumps is suitable for high flows (15 – 250 l/min) with pressures ranging between 5 and 50 bar. If any liquid has to be forced out to a specific distance, the end of the delivery hose must be equipped with a very small nozzle; pressure is generated when the fluid encounters the narrowing section in the final jet. The internal form of the jet affects the fluid flowing out: a tiny and circular opening makes water reach a considerable distance depending on its flow/pressure, whereas if a small helical cylinder is put inside the narrow opening, the liquid flows out in the shape of a very wide cone (rose).

Such a pump is required also to transport fluids through long hoses because the overall resistance the fluid encounters along its path (bends, drive couplings, shutoff valves, branches, level increase) entails an overall significant pressure.

This type of pumps is widely used in many industries, for instance in the farming industry for phytochemicals spraying, liquid fertilisers and weed-killers, water mains, street sweepers, industrial water pumping, water jet cleaners, fire alarms.

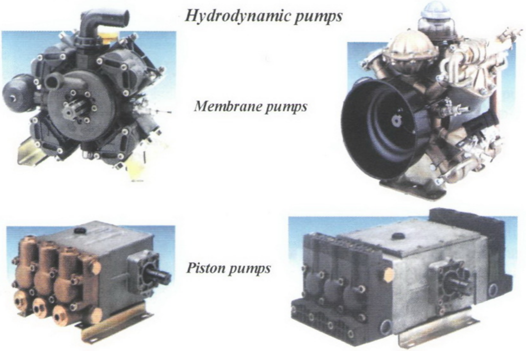


Figure 3.3

Most rotodynamic pumps designed to perform these tasks are *membrane* pumps, that is pumps made up of one or more rubber membranes that are operated alternatively by a oil bath eccentric shaft and equipped with non-return valves; otherwise, rotodynamic pumps can be *piston* pumps, whose operating principle is similar to displacement piston pumps but, unlike them, they need wide internal leakages (Figure 3.3).

DISPLACEMENT PUMPS OR VOLUMETRIC PUMPS – CONCEPTS AND FORMULAS

The volume of the fluid held in displacement pumps varies during the single sequence of each pumping from a minimum next to zero to a maximum that in theory is equivalent to the displacement of the pumping element itself; this is why the word ‘volumetric’ is used.

The substantial pressures involved in fluid power demand volumetric generators to which we are now going to refer only as ‘pumps’.

In order to respond to system requirements, the pump can have a fixed displacement (gears, lobes, screws, vanes, axial pistons, radial pistons) or a variable displacement (vanes, axial pistons, radial pistons).

Key concepts

We can apply the principles of hydraulics to a simple circuit consisting of a prime mover, a pump, a shutoff valve, hoses and a tank assuming for the moment there is no internal resistance.

As shown in Figure 3.4 on the left, since the valve V is open, the fluid is sucked from the tank and flows freely without resistance in theory. The system flow is therefore as high as possible, which means it is equivalent to the maximum flow the pump can generate while pressure is equal to zero. If the hose is blocked by closing the valve V, there is no flow and pressure tends to soar to infinity; actually, in the latter case (Q = 0, p = Max) it is essential to add a relief valve so as not to exceed the maximum pressure the pump can endure, as well as an overload cut-out if there is an electric motor in order to prevent them from being destroyed.

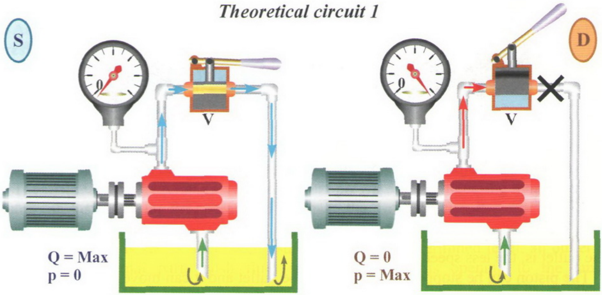


Figure 3.4

Yet, an oil hydraulic circuit does not make sense without at least one actuator. Consider the simplest circuit, made up of the motor/pump unit, an adjustment valve, an actuating cylinder and connection hoses (Figure 3.5).

During the initial standby phase of the motor/pump, the load keeps down the cylinder piston via the rod. The fluid flow of the motor/pump, started by the start control, is bucked by the cylinder piston that sustain the opposite force F of the load; the ensuing pressure offsets the force F of the load and the overall internal resistance; the piston starts moving up until it goes beyond the outlet connected to the unloading hose. At this point the fluid enter this outlet and it is unloaded into the tank; pressure decreases and the load pushes the piston downward; however, the short upward movement stops as soon as the piston closes the outlet, thus promoting the cyclic oscillation of the load along a distance that is equal to the distance travelled by the piston to the outlet.

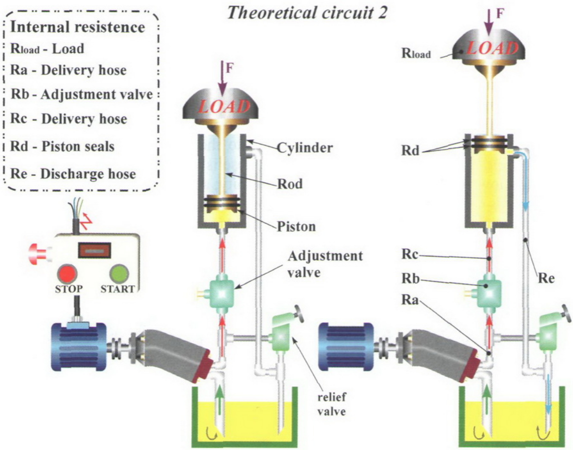


Figure 3.5

The flow adjustment valve regulates the translation speed of the cylinder: the smaller the outlet is, the less speed is.

The piston can be stopped so as to block the outlet and it can move down very slowly only if internal leakages inside the pump occur between delivery and suction.

The overall pressure inside the circuit is equivalent to the sum of the internal resistances the parts oppose:

- R_a = Internal resistance of the hose between the pump and the adjustment valve
- R_b = Internal resistance of the adjustment valve
- R_c = Internal resistance of the hose between the valve and the cylinder
- R_d = Internal resistance of piston seals
- R_e = Internal resistance of the delivery pipe
- R_{load} = Force exerted by the load

The load opposes a resistance that is equal to its weight F, which exerts a pressure on the piston that is equivalent to the ratio between F and the surface S of the lower face of the piston:

$$p_{load} = \frac{F}{S} = \frac{daN}{cm^2} \text{ (bar)}$$

Assuming the load has a force F of 1000 daN and a piston cross-section of 8 cm^2, pressure is:

$$p_{load} = \frac{1000}{8} = 125 \text{ bar}$$

If the sum of internal resistances ($R_a + R_b + R_c + R_d + R_e$) generates a pressure p_r of 45 bar, the overall pressure p inside the circuit is:

$$p = p_{load} + p_r = 125 + 45 = 170 \text{ bar}$$

As there is no resistance R_e during the upward movement of the rod, the overall pressure is slightly lower in this phase.
We stress again that this circuit (with a stop button, without the pressure control valve, the cylinder upper head and a regular upward movement) *serves only as an example* and systems cannot actually be equipped with it.

As we are often going to mention the **relief valve**, it is worth explaining briefly how it works; a more detailed description is provided in the chapters on adjustment valves.
The main task of the relief valve is to prevent pressure from increasing more than the pump can sustain during the operational phases of the hydraulic circuit.

Once the valve is set, as soon as pressure reaches a predetermined level, an outlet is opened and internal parts push fluid surplus through this opening to the tank; consequently, pressure cannot exceed that level (this operation reflects what happens between the piston and the outlet of the cylinder without the upper head we previously mentioned).

In order to make pressure drop in the whole system, the pump is put in standby by adding a simple circuit to the Vent connection of the relief valve (but not all of them are equipped with it) (see paragraph above –Intermittent operation–).

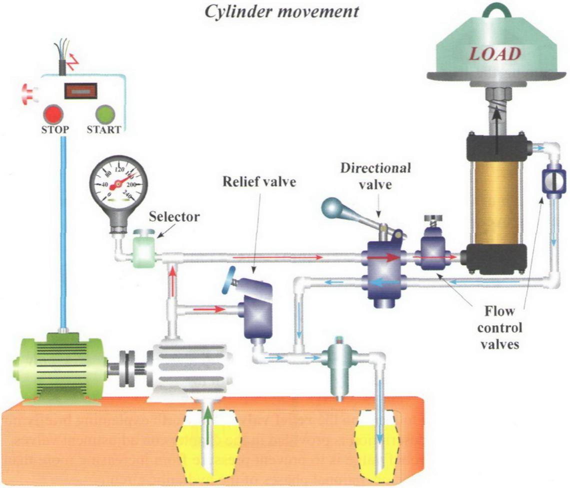


Figure 3.6

Figure 3.6 shows the circuit described above and consisting of a motor pump, flow control valves (speed of rod translation) and a cylinder (this time with both end caps!), plus a relief valve, a directional valve, suction and discharge filters and the manometer equipped with a selector.

The pump has to meet system demands: as a matter of fact, it must sustain not only the operating pressure but also peak pressures, as well as providing a constant flow, avoiding overheating and excessive noise, being long-lived.

Consider a system made up of three actuators A, B, C connected to the same pump, which is equipped with a relief valve; the actuators are controlled by a single directional valve (Figure 3.7) and have the following characteristics:

- ✓ Cylinder A – Piston cross-section = 50 cm^2, rod stroke 100 mm, facing an opposite force of 1000 daN – p_A = 1000/50 = 20 bar.
- ✓ Cylinder B – Piston cross-section = 50 cm^2, rod stroke 100 mm, facing an opposite force of 5000 daN – p_B = 5000/50 = 100 bar.
- ✓ Cylinder C – Piston cross-section = 50 cm^2, rod stroke 150 mm, facing an opposite force of 15000 daN – p_C = 15000/50 = 300 bar.

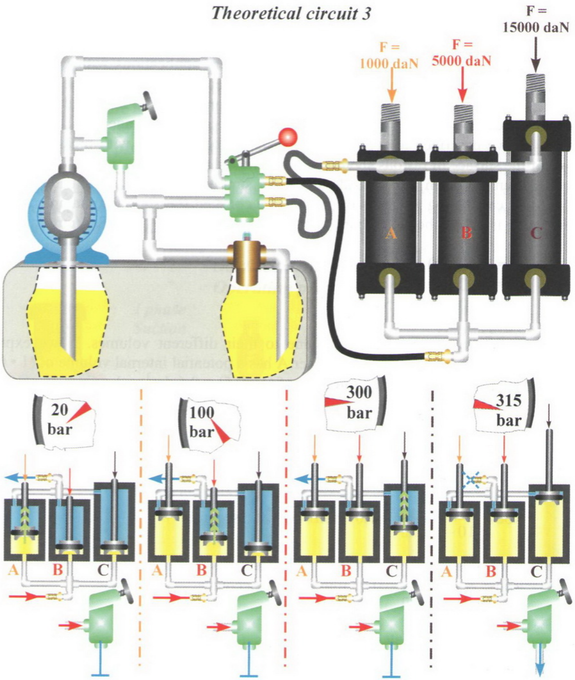


Figure 3.7

The pump is started, fluid pressure begins going up to 20 bar, which is the pressure needed to offset the force cylinder A opposes. Piston A moves up while B and C do not move because they do not have enough pressure. Pressure remains at 20 bar throughout the path and it starts soaring up to 100 bar (the level required to move piston B) only when piston A touches the upper end cap. When B reaches the upper end cap, the manometer displays 300 bar and piston C performs the whole cycle.

When C reaches its final position, pressure should further increase, but the relief valve, set to as little as 5 ÷ 10% higher than 300 bar (this rate depends on possible problems, like additional friction, etc.) keeps it constant until rods starts moving back. Although they have different strokes, flow does not affect system parameters and rod speed is the same and constant in all of them, unless flow control valves are added.

This circuit too is purely an example; actually real applications with two or more cylinders are unlikely to have actuators without directional valves for their handling. Yet, the situation described is real because two or more actuators subjected to different forces and equipped with a directional valve are moved simultaneously; in order to avoid what we described above, the system should be provided with pressure control valves or the surface of pistons should be dimensioned properly.

In the example taken into consideration, the simultaneous start of actuators can be achieved by calculating the areas of pistons at a constant pressure, for example 100 bar (relief valve is set to about 110 bar); hence, pistons surfaces S_A, S_B, S_C, (cm^2 = daN/p) are:

- ✓ $S_A = 1000/100 = 10\ cm^2$
- ✓ $S_B = 5000/100 = 50\ cm^2$
- ✓ $S_C = 15000/100 = 150\ cm^2$

The upward movement is different due to their different volumes. If we express strokes in dm and sections in dm^2, cylinder A has a potential internal volume of (1 • 0.1) = 0.1 dm^3 (one-tenth of a litre), cylinder B of (1 • 0.5) = 0.5 dm^3 and cylinder C of (1.5 • 1.5) = 2.25 dm^3.

If we neglect volumetric efficiency and assuming the pump pumps 10 l/min at a pressure of 100 bar (i.e. it fills a volume of 10 dm^3 per minute), the time rods take to travel their strokes is:

- ✓ Rod_A: $\dfrac{0.1}{10-1/\min} = 0.01\ \min \bullet 60 = 0.6$ seconds

- ✓ Rod_B: $\dfrac{0.5}{10\ \ 1/\min} = 0.05\ \min \bullet 60 = 3$ seconds

- ✓ Rod_C: $\dfrac{2.25}{10\ \ 1/\min} = 0.225\ \min \bullet 60 = 13.5$ seconds

Working principle of displacement pumps

Displacement pumps are equipped with a reciprocating or rotary operating system. In the first case, the pumping part or parts are the pistons, which fit inside their cylindrical chambers; pistons are positioned in a radial or axial manner around the rotating shaft connected to the prime mover or, in manual pumps, operated by a lever; rotary pumps can be gear, screw, lobe or vane pumps. Unlike vacuum technology, oil hydraulics does not allow the use of suction systems based on Venturi principle.

The cycle of a pump is divided into two phases:

I Phase -**Suction**- *(Inlet phase)*
II Phase –**Delivery-** (or Compression) *(Output phase)*

In the inlet phase, the pumping part, be it a piston, a gear tooth or another rotating part, creates a vacuum in the suction chamber that helps the fluid flowing up due to the atmospheric pressure that acts on the free surface of the tank.

In the output phase, once suction is completed and the chamber is fully filled with the fluid, the pumping part pushes the fluid to the outlet connected to the system hose.

The pump is often on standby (the relief valve releases the fluid through the Vent connection); as a result, pressure is almost zero during the delivery phase, also know as 'compression'. Meanwhile the fluid keeps on flowing inside the pump (from suction to draining) so as to lubricate sliding parts.

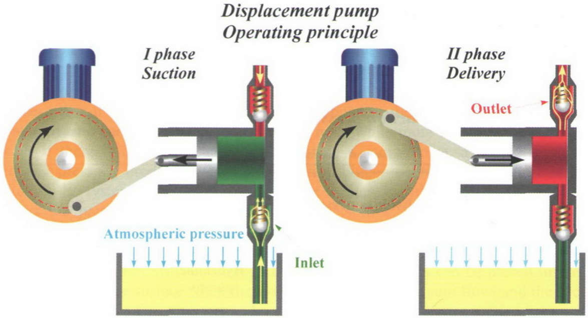


Figure 3.8

The two phases (suction-delivery) alternates immediately in piston reciprocating pumps, whereas in rotary pumps, as we are going to see later on, the fluid flows through an inactive part of the stator between suction and delivery.

The operating principle shown in Figure 3.8 can be misleading: it is the operating principle not only of piston pumps, but also of all reciprocating and rotary pumps.

The sine qua non is the hermetic sealing of inactive openings. This means the outlet must be closed during suction so as to avoid the collapse of the vacuum and prevent the fluid already delivered from flowing back into the chamber; if the inlet was open during the delivery, the fluid would have a very easy way out to the tank. Rotary pumps do not have this drawback because there is no direct phase alternation; it is essential in piston pumps to add non-return valves on the openings or fixed or rotary distributors (depending on the type of pump) synchronised with piston movements.

Suction

Suction in all displacement pumps is a crucial phase affected by a number of important physical and mechanical conditions. Sometimes their internal design itself, which is conceived so as to sustain substantial flows and ensure high pressure and efficiency, does not promote an efficient suction; high speeds (500 – 2500 rpm on average) make it impossible to add dynamic seals between the seat and the moving part of the pump because they would wear out very quickly and create such a friction that the prime mover would have to supply a huge power. The lack of seals promotes a longer working life of the system on the one side, but on the other it undermines suction because of the leakage between moving and fixed parts of the pumps.

If suction is inadequate (little fluid flows to the pumps), an additional pump, known as **booster pump**, should be added; this extra pump can ensure an adequate flow to the pumping part of the main pump although its operating features are lower and generate pressure only due its flow forces.

On the other hand, excessive suction pressures would inevitably lead to aeration and cavitation, which already jeopardise the mechanical structure of the pump.

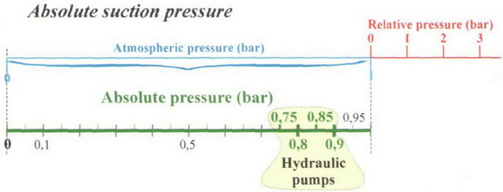


Figure 3.9

Consider a pump with a suction absolute pressure of 0. 85 bar.

We stress again that this is the sum of relative and absolute pressure (see chapter 1) and in practice 0.85 absolute bar *are not equivalent* to 1 – 0.15 relative bar, but to an atmospheric pressure tending to the vacuum and therefore it is equal to 0.85 bar i.e. – 0.15 relative bar (Figure 3.9).

When it is not operational, the fluid held in the tank and inside the submerged part of the suction hose cannot flow up to the pump because atmospheric pressure acts also inside the hose. Once the system is started, the fluid can flow up thanks to the vacuum generated in the pump chamber and in the upper part of its hose as far as the free surface (0.85 absolute bar in this case). If we ignore the various pressure drops like friction, narrowing, fluid density and hose height, the thrust that makes the liquid reach the pump is equivalent to the difference between the atmospheric pressure of 1 bar and suction absolute pressure (0.85 bar):

Thrust the fluid experiences = $p_{\text{atmospheric}} - p_{\text{suction absolute}}$ (bar) = 1 – 0.85 = 0.15 bar

By decreasing the absolute pressure (i.e. shifting it towards the vacuum), for instance to 0.3 bar, the thrust, that is now equal to 0.7 bar, is quite higher, but the ensuing rise in temperature due to a strong friction would entail intolerable cavitation conditions.

Also the parameters of the suction hose are very important. A small internal cross section, a bend, a drive coupling or any type of narrowing promote friction with a subsequent overheating; too long a hose holds a column of liquid whose weight exerts a force that is stronger than thrust. The height between the pump and the fluid free surface should be as small possible; usually suction hoses do not exceed 80 cm and have a larger cross section than delivery hoses. Hose manufactures dimension outlets and inlets depending on the characteristics of the pump and they specify the maximum suction speed of the fluid; the coupling hose size depends on these dimensions: a larger cross section promotes suction but the drive coupling could bring about localised pressure drops, while the consequences of a hose with a small cross section can easily be figured out!

There is still the problem of the filter positioned at the lower end of such a hose: contaminating particles are sucked if there is none while more localised pressure drops occur if there is one. As usual, the solution is a compromise: a course mesh filter reduces pressure drops and prevents at least bigger contaminating particles from entering the system. The system must include a filter over the discharge hose so as to blocks the particles that entered fluid during its flow, thus preventing them from flowing into the tank.

A pump that is not efficient enough under these conditions can be placed inside the tank, under the free surface when the system reaches its maximum flow, and the suction hose is positioned a few centimetres from the tank bottom: a shorter suction hose avoids substantial pressure drops; a flange holding the shaft connects the pump to the prime mover, which is obviously attached to the upper surface of the tank (Figure 3.10).

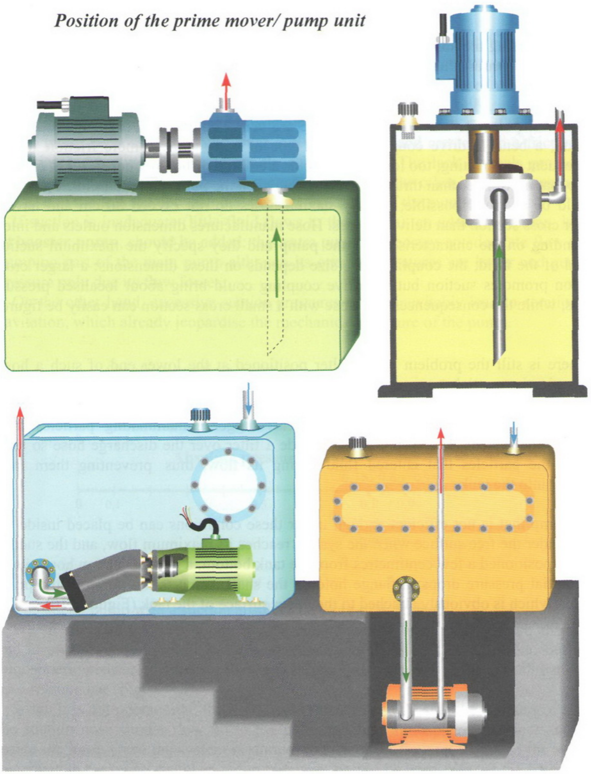


Figure 3.10

In order to exploit not only atmospheric pressure but also fluid weight, the pump should be positioned outside the lower part of the tank or even some dozens of centimetres below the tank if we want to exploit the speed of fall of the fluid.

The height of the suction hose h_{as} (cm) can be calculated from Bernoulli's formula assuming v (m/s) is the average speed of the fluid between the tank and the pump, p_{as} (bar) is the maximum absolute pressure of suction (the pump specifications include it), pc (daN/cm^{2}m) is the pressure drop per linear metre of the hose and ρ (kg/dm^3) is the fluid density:

$$h_{as} = \left(\frac{p_{as}}{\rho} - \frac{v^2}{2g} \right) \bullet \frac{1}{1 + \frac{pc}{\rho}}$$

Compatible viscosity

The pumps is the part that is most affected by the viscosity grade of the fluid employed. The *optimum viscosity* grade, that is the ideal combination of a fluid and a pump working properly, can be achieved only at a specific temperature that depends on the type of the system; in general, it ranges between 35 and 45 °C and in other words it is the temperature the system sustain most of time; the oil or any other fluid employed must have an average viscosity (at 40 °C) taking into consideration the peak and bottom temperature (see ISO VG tables in chapter 2).

Too viscous a fluid entails, on the one side, an increase in pressure drops in the suction hose resulting in cavitation and, on the other, overheating due to the effort of moving parts; a fluid with low viscosity avoids most cavitation but its poor grip results in internal leakages with the ensuing poor volumetric efficiency, wear and high temperatures.

Manufacturers specify in the rating plate the *minimum and maximum viscosity* the pump can endure, that is the tolerable viscosity range beyond which the parameters of components are no longer guaranteed, because it is difficult to determine the exact optimum viscosity (it is affected by many factors) and viscosity limits must be known. For instance, in a pump with a stated range of 16÷200 cSt, at start viscosity at any environmental temperature must not exceed 70 cSt and it must be at least 30 mm^2/s with exceptional running at high temperatures.

These viscosity limits are often unsuitable in some operational conditions, especially for self-propelled machines. A snowplough must be started at a temperature far below 0 °C, oil in a mower that is small so as to fit to a farm tractor exceeds the maximum operational temperature quickly. Some special devices, like heat exchangers placed in the outlet hose or tank preheating electrical resistances, reduce fluid temperature. In addition, special vane pumps are available on the market: they can be started at very high fluid viscosity (500÷600 cSt at start) and their revolution brings the oil to the working temperature, although it cannot guarantee the immediate pressurisation of the vehicle.

Cavitation

Vapour tension in fluids (see Chapter 2), i.e. the point at which a liquid changes into gas due to pressure and temperature, entails the phenomenon known as 'cavitation'.

Different types of anomalies generate some obstacles that impede the fluid flow, thus causing both pressure drops with the resulting temperature changes and the decrease (towards the vacuum) in absolute pressure in the whole suction area.

Vapour tension in a fluid with such a pressure and temperature creates tiny cavities (hence the word 'cavitation') filled with gas, mainly air, released by the liquid.

The low pressure found where the liquid flows from the suction area to the delivery area promotes the increase in size of the bubbles, which are super-compressed by the fluid itself once they are in the area subjected to system pressure.

This causes pressurised cavities to collapse, which results in very strong tangential stresses affecting the solid parts of the pump. In practice, these imploded particles are projected towards the different parts at a very high speed and they cause a stress that is similar to the stress generated by tiny bullets that hit the materials and erode them (Figure 3.11). By releasing oxygen in the fluid, the air found inside it leads to metal oxidation and early oil ageing.

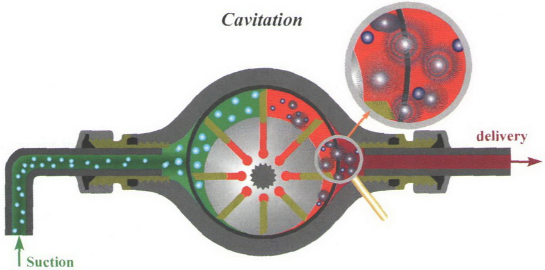


Figure 3.11

A phenomenon that is partially similar to cavitation occurs if there is a substantial amount of air generated by foam inside the tank; some foam bubbles experience the same phenomenon of cavitation in suction, while others move beyond the pump chamber and enter the circuit, thus increasing the overall fluid compressibility.

The main causes for cavitation are the improper dimensioning of the suction hose and high fluid viscosity, while the inclusion of air is accounted for by not only foam in the tank but also faulty drive couplings on the suction hose itself. The following table details the different causes for cavitation and air inclusion.

CAUSES FOR CAVITATION AND INCLUSION OF AIR DURING SUCTION	
Cavitation	Inclusion of air
High viscosity grade of the fluid	The level of fluid inside the tank is lower than the standard level
Faulty connections between the hose and the pump	Faulty dynamic seals of the pump (if any)
High revolution speed (rpm) of the pump	The suction hose is porous or has a hole between the pump and the free surface
Clogged suction filter	Poor seal between the pump and the suction hose
Poor lubrication inside the pump	Connection pipes are not hermetic or fixed properly
Tank temperature below the average	Faulty drive coupling welding on the suction hose
Small internal cross section of the suction hose	Discharge hose end above the free surface
Narrowing of the suction hose	Foam inside the tank due to the discharge hose
Malfunctioning tank relief valve	
Fluid density is not suitable for that specific pump	

Effects of cavitation

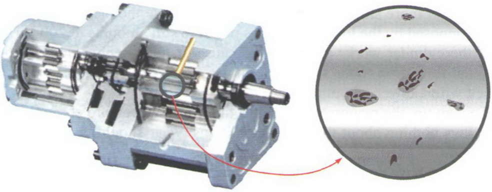


Figure 3.12

Cavitation causes much noise, vibration and the erosion of the internal parts of the pump (Figure 3.12). This phenomenon in the system can be detected by an increase in noise and poor efficiency; yet, we must stress that cavitation is not often responsible for these two problems.

Delivery

The fluid moves from the suction area to the delivery area by due to the inversion of the motion of each piston or the revolution of gears, screws or vanes. When the circuit is closed, there is no flow and pressure exceeds the maximum tolerable level; as a matter of fact, the fluid cannot flow because of piston end-stroke or a directional closed centre valve.

In fixed displacement valves, the relief valve only can prevent pressure from exceeding the maximum level specified by the pump manufacturer; on the other hand, displacement in variable displacement pumps is reduced to almost 0 by special adjustment devices covered in the next chapter and by using relief valves too, which by the way are needed in case of possible malfunctions.

When the system is on, the oil held in that part of the cylinder previously blocked by the valve is unloaded and the ensuing pressure on the positive face of the piston outweighs the force the load opposes, thus starting its translation (for the time being we will consider cylinders because of their easy operating principle, while motors needs more explanations even if they are based on the same principle).

Unfortunately, the pump is often considered the main culprit for malfunctioning systems. The most common situation occurs whenever slowdowns in the load translation speed are detected.

A worn-out actuator is subjected to substantial leakages between the pressurised and discharge areas because of both scoring on the liner due to dirty oil and dynamic seals that had to be replaced long before.

Assume a load of 10000 daN is translated by a cylinder whose piston has a surface of 50 cm^2 and the pump has a flow of 8 l/min.

The ensuing pressure is: $p = F/S = 10000/50 = 200$ bar.

If we assume piston leakages are such as to allow an oil leak from one chamber to another of 3 l/min under working conditions, and the prime mover works properly, the load, albeit more slowly, keeps on moving because the piston manages to outweigh the force opposed by the load; this means pressure still amounts to 200 bar.

If the cylinder works properly, the cause of this phenomenon can be the pump: the internal flow drop (poor seals) still allows a pressure of 200 bar on the actuator piston, albeit with a lower oil outlet.

To sum up, the manometer displays the working pressure correctly in both cases and this says nothing about the pump conditions: wear should be assessed before replacing any part.

What we have described so far is an anomaly occurring under constant operational conditions; yet, dynamic changes due to the resulting hydraulic power must be taken into account in systems even when they are fully efficient.

Hydraulic power is the product of flow and pressure.

If for the time being we neglect efficiencies and avoid the definition in kW, HP, etc., hydraulic power N can be defined as:

$$N = p \cdot Q$$

In a real circuit, the prime mover is dimensioned according to the ideal performance under standard working conditions and a sudden rise in pressure due to unexpected factors demands more power: rotational speed decreases to the detriment of flow, which is affected by the number of revolutions of the pump. Actually, different motors react in different ways to these increase in energy: three-phase asynchronous motors experience a slowdown and an increase in power, petrol engines too experience a speed fall and torque increase; when specific limits are exceeds, the magnetothermic switch turns on in asynchronous motors while petrol engines stop. Diesel engines instead stop almost immediately for not so large increase in power demand too.

Displacement

Figure 3.8 (*Operating principle of displacement pumps*) shows clearly that the overall liquid that is sucked is equivalent to the cylinder internal volume or rather to the product of the cylinder cross section and the overall length travelled by the piston.

Since pump displacement is practically stated in **cm³/revolution**, assuming a single piston pump with a cylinder cross section S of 500 mm^2 and a piston stroke of l of 20 mm, its *theoretical displacement* c is:

$$c = S \bullet l = \frac{50 \bullet 20}{1000} = 10 \text{ cm}^3$$

In multi-piston reciprocating pumps, the product of the displacement of a pumping part and the total number of pistons is equal to the over all theoretical displacement: the theoretical displacement of a pump with 9 axial pistons, each of them with a volume of 10 cm^3 is $(10 \bullet 9) = 90$ cm^3. The theoretical displacement of rotary pumps depends on the number of meatus: the displacements of these pumps is equal to the product of a single meatus (space between moving parts; for instance, the meatus of a gear pump and a vane pump are the space respectively between two teeth and the relative stator surface and two vanes and the relative stator surface) and the total number of meatus.

There is a reason why we stressed the expression ‘theoretical displacement’. The actual volume that is sucked and delivered is *never equivalent* to the theoretical displacement, better known as **geometric displacement**, which is always affected by leakages, the inability to suck the whole volume due to the high number of revolutions, air inclusion, fluid compressibility, the resistance opposed by the suction filter and the various pressure drops.

Theoretical flow is equal to the product of geometric c and the revolutions per minute rpm of the pump; the **actual flow Q** in dm^3/min (litres per minute) is affected by volumetric efficiency η_v.

$$Q = \frac{c \bullet \text{rpm} \bullet \eta_v}{1000} = \text{dm}^3/\text{min} = \text{l/min}. \text{ Therefore: } c = \frac{Q \bullet 1000}{\text{rpm} \bullet \eta_v} = \text{cm}^3$$

Efficiency

The **volumetric efficiency η_v** of a hydraulic pump is the ratio between theoretical flow Q_t and actual flow Q measured at the outlet by means of flow transducers. It depends on the characteristics of the system, the pressure and speed of the test, the fluid employed (density, viscosity, temperature, compressibility, etc.). Like all types of efficiency, a percentage or a decimal number indicates it.

$$\eta_v = \frac{Q}{Q_t}$$

The **mechanical efficiency η_m** of a hydraulic pump is the ratio between the actual pressure p measured at the outlet and the theoretical pressure p_t; theoretical pressure is calculated by taking into consideration the actual driving torque recorded with a torsion meter on the driving shaft (p_t = actual torque / theoretical displacement). It is affected by the same parameters as volumetric efficiency.

$$\eta_m = \frac{p}{p_t}$$

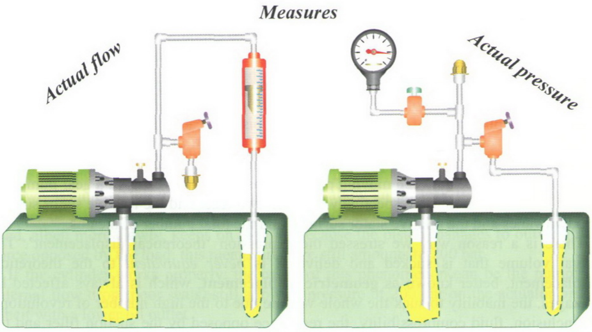


Figure 3.13

The **overall efficiency η_g** of a hydraulic pump is the ratio between the hydraulic power delivered and the mechanical power used under specific conditions of pressure, number of revolutions and type of fluid.

$$\eta_g = p \cdot Q \,/\, (\text{actual torque} \cdot \text{rpm}) = p \cdot Q \,/\, (p_t \cdot \text{theoretical displacement} \cdot \text{rpm}) = \eta_m \cdot \eta_v$$

In brief, he overall efficiency is the product of the mechanic efficiency η_m and volumetric efficiency η_v.

$$\eta_g = \eta_v \cdot \eta_m$$

Manufacturers specify the theoretical displacement, the maximum working pressure and the peak pressure (Figure 3.13).

Power

Mechanical power P in a pump is transformed into hydraulic power N by applying the following formulas:

N = Hydraulic power (kW)
P = Mechanical power (kW)
M = Twisting moment or torque ($N \cdot m$)
c = Displacement (cm^3/revolution)
Q = Flow (dm^3/min or l/min)
p = Pressure (bar)
η_v = Volumetric efficiency
η_m = Mechanical efficiency
η_g = Overall efficiency
rpm = Rotational speed (revolutions/min)

Absorbed torque

$$M = \frac{c \cdot p}{20 \cdot \pi \cdot \eta_m} \quad (N \cdot m)$$

Absorbed mechanical power

$$P = \frac{M \cdot rpm}{9740} \quad (kW)$$

Hydraulic power

$$N = \frac{2 \cdot \pi \cdot 10^{-3} \cdot M \cdot rpm}{60} = \frac{Q \cdot p}{600 \cdot \eta_g} \quad (kW)$$

Actual rotational speed

$$rpm = \frac{Q \cdot 1000}{c} \cdot \eta_v \ (\text{rotations/min})$$

Using the units that the SI no longer allows, that is Cavalli Vapore CV or hp and Horse Power HP:

1 CV = 0.735 kW 1 kW = 1.36 CV 1 HP = 0.745 kW 1 kW = 1.34 HP

$$\frac{Q \cdot p}{450 \cdot \eta_g} \quad (HP)$$

The number of revolutions per minute, measured in 'rpm' due to the US/UK system, is often referred to also as the consonant n and it is known also as 'engine speed' or simply 'speed'.

USA System

In the United States of America flow is often measured in *gallons per minute* (GPM), displacement in CCR *cubic centimetres per revolutions* (cm^3/rev) or in CIR *cubic inches per revolutions* (in^3/rev).

$$Q = \frac{d \cdot rpm \cdot Eff_v}{231} \text{ (GPM)}$$

Where:
Q = Flow (GPM)
d = Displacement (in^3/rev)
1 G = 231 in^3
G = US gallon= 3.78 litres
Since η stands for 'efficiency':
η_v or Eff_v: Volumetric Efficiency
η_m or Eff_m: Mechanical Efficiency
$\eta_g \rightarrow \eta_{oa}$ or Eff_{oa}: Overall Efficiency

Since Americans measure pressure in psi (pounds/square inch – lbf/in^2), following formula is often used in the United States for Hydraulic Horsepower:

$$HP = GPM \cdot psi \cdot 0.000583$$

1 GPM = 3.78 dm^3/min 1 dm^3 / min= 0.265 GPM

Maximum pressure and flow

The specifications manufacturers provide along with the pump indicate the working pressure and the maximum or peak pressure; the latter can be sustained only for short whiles followed by standbys and the manufacturer itself establishes both of them. Higher pressures would lead to poor performances quickly and inevitably.

The minimum and maximum rotational speed rpm specified by the manufacturer sets a speed limit below the pump cannot guarantee the ratio between flow and pressure, as well as a limit above which mechanical stress, leakages, friction resulting in temperature rise, loss of fluid properties and the likelihood of cavitation cannot guarantee proper working conditions.

Pre-filling

The installation of a new or overhauled pump on the system needs pre-filling, i.e. the manual filling inside the internal parts of the fluid with the same properties as the fluid held in the tank, using clean pre-filtered fluid to pump manufacturers' recommendations.

As in some type of pumps, especially piston pumps, this fluid does not reach the pumping parts, air bleeding must be carried out as soon as the system is started and before it is operational. In order to avoid dangerous oil squirting, bleeding should not be performed directly on the delivery hose but on the bleed valves or, in the absence of these valves, on the valves next to the actuating cylinder.

ASSEMBLIES

The pump or pumps needed to meet the requirements of the whole system are chosen depending on the characteristics of the system, or rather on the requirements of the actuators. The generating part or parts must therefore be connected and firmly assembled first between them and secondly to the prime mover. In addition, the ensuing natural vibrations should be dampened as they increase due to the wear of components after several working hours.

Coaxial or multistage pumps

There is a myriad of oil hydraulic systems that require two or more pumps on a single circuit. The trend in stationary applications has long consisted in connecting the different pumps to a single prime mover in order to save resources in the installation of two or more motors and their drive couplings, bell housings, clamps, vibration dampeners, as well as to cut back on expensive electric connection. The logic in mobile applications is clear: it would be nonsense to devise an endothermic motor for traction and another motor for operational circuits, with a few exceptions.

The need for coaxial pumps (multistage), i.e. placed in a series on a common axis and directly connected one to another, prompted manufacturers to bring out some standard products that were suitable for a series and mechanical connection (Figure 3.14). Such a pump can work alone or in connection to other pumps in a coaxial manner after a very easy and quick intervention. The back cover of the first pump, connected to the prime mover, and intermediate pumps needs removal, while the last pump keeps its original assembly.

In theory, there are no limits to the number of coaxial pumps if there is a very powerful prime mover; actually, systems are devised for a limited number (usually two, three or four) because of different factors like the maximum torque of the prime mover, the number of revolutions of each pump (obviously it is the same for all the pumps), the simplicity of the circuit, vibrations…

Figure 3.14

Pump assembly is various because it is possible to connect pumps of the same type and different flow/pressure, for instance two, three or four gear pumps, one or two piston pumps, and a final gear pump, a variable displacement vane pump and a fixed displacement pump, etc.

Two coaxial pumps, that is a vane or piston pump and a gear pump, are typically used in stationary applications, in those circuits where servo-controlling solenoid valves need a dedicated hydraulic circuit (gear pump) and in self-propelled machines hydrostatic drive where an auxiliary gear pump provides oil boosting to closed circuit.

Coaxial pumps are quite suitable for a system with two or more hydraulic motors, each of which needs a specific pressure and/or flow, because it is not always possible or economical to change the parameters via control valves; two coaxial pumps with different displacements can serve as many actuators with different dimensioning but the same speed required.

Drive couplings

An improper alignment of pump/motor shafts entails overheating and quick wear of the bearings of both parts, which causes more vibrations, noise, less efficiency and flow; the pump is doomed to be short-lived under these conditions.

Except for rigid drive couplings, the design of the drive couplings on the market promotes axial and radial alignment and it tolerates misalignments of about two grades and shaft loosening of few millimetres. Still, it is important not to abuse this tolerance: the alignment must be performed very carefully in view of misalignments due to settlement that could occur during the operation.

Mechanical interventions are therefore difficult and they involve the use of rulers, callipers and feeler gauges as well as the assembly instructions provided by manufacturers.

Anyway, a thorough analysis of the characteristics of the system is carried out before the installation so as to set the criteria for the choice of the ideal drive coupling. The factors that affects this choice are, on the one hand, power characteristics, rotational speed and torque and, on the other, the simplicity of the lubrication and of the replacement of the elastomer, as well as bores, the design of shafts (splined or feather key shaft) and special connections to endothermic motors.

A drive coupling for hydraulic pumps is mainly made up of female couplings for each shaft and the flexible part in the middle fixed to the couplings by screws or various gears, which must allow elastometer flexibility in any case (Figure 3.15).

Flexible drive couplings

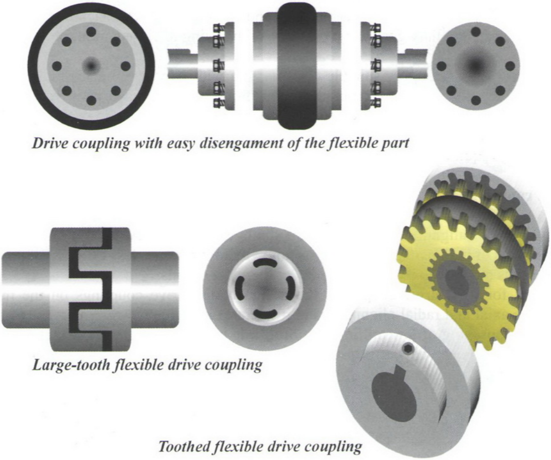


Figure 3.15

The central part made of metal, nylon, polyamide or rubber can be replaced, in some types, by disassembling the pump or the motor and removing the lateral bosses first and then the central or, in other types, by simply loosening the screws between the bosses so as to replace the central part.

The choice of drive coupling dimensioning depends on the pump torque. For instance, if a motor has a power of 80 kW and it is connected to a pump of 45 kW (obviously other devices too are connected to the motor), the matched joint for that motor is too rigid.

Endothermic motors allow not only to be connected to the shaft but also flywheel or pulley coupling (Figure 3.16).

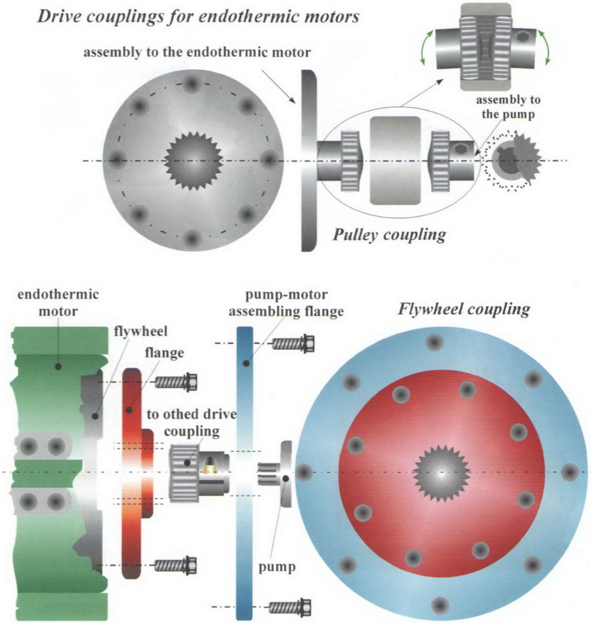


Figure 3.16

Bell housings

The bell housing allows a safe and quick assembly of the prime mover to the pump (Figure 3.17). This simple object, made up of a hollow truncated cone that can be moulded at the manufacturer's option in order to make the brand recognisable, ensures the connection of the motor alone to the tank or the support bed; the pump remains solidly connected to the bell housing itself. The connector screws of the motor and the pump pass through the holes at the ends of the bell housing (Figure 3.18).

Bell housings and drive couplings

Figure 3.17

Bell housings

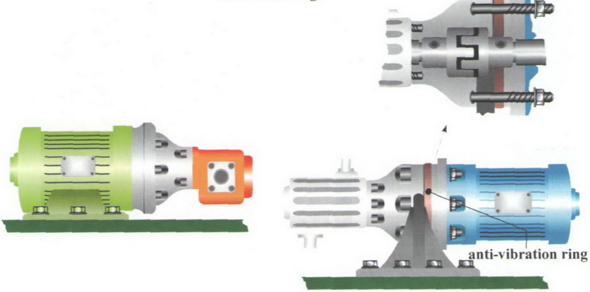
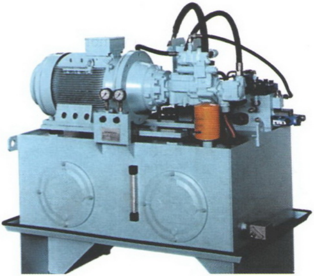
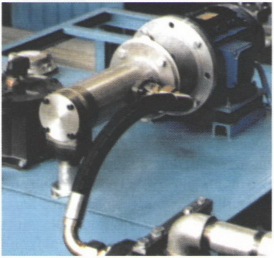


Figure 3.18

Bell housings have an internal cavity where the drive coupling can be positioned. The assembly can be simplified by a flange fixed to the tank and directly connected to the bell housing instead of anchoring it to the motor.

Bell housing couplings





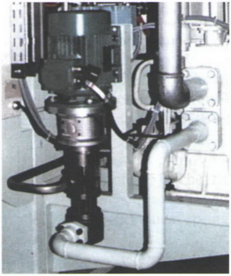


Figure 3.19

Vibration dampening

The vibrations the motor pump generates during its operations damage the system and cause noise exceeding the limits. There is an increase in the vibrations that already occur in recent or even new applications as components wear out and they must be dampened as much as possible by means of plugs and flexible fairleads.

A simple bell housing cannot reduce the oscillations of the generating unit and if vibrations are particularly strong it is possible to resort to bell housings with separate bell housing and almost the same dimensioning, in which the groove of a shaped rubber ring responds to the needs of this case.

The flange and the clamp bolts of the support surface must be covered with anti-vibration plugs, the hoses connected to the pump and the circuit must be uphold by internally flexible collars and horizontal plates provided with rubber thru-bulkheads (Figure 3.20).

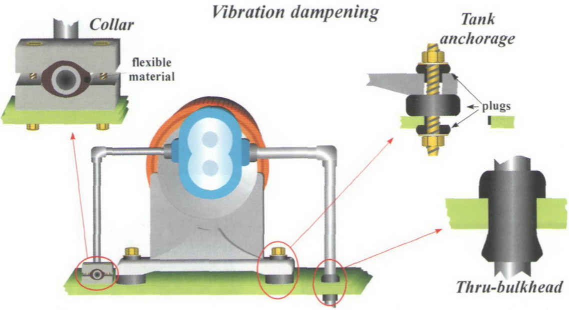


Figure 3.20

Chapter 4

OIL HYDRAULIC PUMPS

Oil hydraulic pumps are classified first as fixed or variable displacement pumps.

Even if pressure reaches the maximum limit and the prime mover can generate the power needed, flow in fixed displacement pumps is constant as long as the rotational speed of the mechanical generator stays the same. Flow rate changes when the number of revolutions on the transmission shaft declines: flow diminishes as revolutions per minute decrease and vice versa; pressure does not change (under specific conditions and if the load is constant).

Special mechanisms inside the pump casing in variable displacement pumps decrease displacement from the maximum limit to zero flow (see chapter 5). The external intervention can be carried out automatically or manually by means of a hand wheel. In the first case, the dedicated controllers can be controlled with a mechanical spring, load sensing, electromechanical or electroproportional system.

Variable flow pumps (not all types are suitable) that can reverse the flow are available: their outlet becomes the inlet while their prime mover has a constant direction, which is the prerequisite for closed circuit applications like wheel and track drive or hoist and winch handling.

FIXED DISPLACEMENT PUMPS

The hydraulic pumps described in this paragraph can have only a fixed displacement for a number of mechanical reasons.

Manual lever pumps

Manual pumps (Figure 4.1) consist of a piston sliding inside its cylinder and operated by a lever that can be moved by hand or with a spring return pedal during suction depending on the applications.

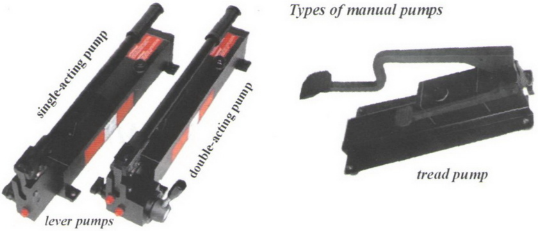


Figure 4.1

Like any other piston pump, their operating principle is based on the reciprocation of suction and delivery (Figure 4.2).

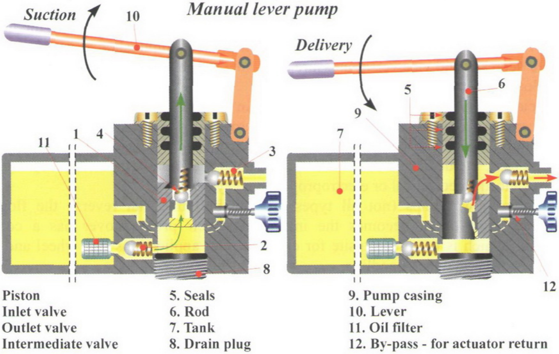


Figure 4.2

By pulling the lever (10) upward, the rod (6) moves the piston (1) during suction: the inlet valve (2) opens allowing the oil to flow into the cylinder; the intermediate valve (3) prevents the delivered fluid from flowing back into the pump casing (9).

At the upper dead centre, delivery is started by pushing the lever downward: the inlet valve (2) closes preventing the fluid to flow back into the tank, the intermediate valve (3) allows the fluid to flow during the delivery. Seals (5) block leakages between the

External gear pumps

Apart from manual pumps, which are used (albeit widely) only for simple flying applications, external gear pumps are the simplest and most common oil hydraulic pumps operated by a motor. Their success is accounted for by a number of advantages, like their extreme lightness, mechanical simplicity, wide-range viscosity tolerance, optimum suction, their wide range of flow rates, adaptability to any position and space and, last but not least, their cost, which makes them one of the cheapest types on the market.

Yet, these pumps have some drawbacks too, i.e. a design focused on sustaining around 150 – 280 bar, their unsuitability for high flow applications, rather loud noise and a rather poor overall efficiency.

Nonetheless, the strong demand for these hydraulic generators prompts manufacturers to carry out research and improve their products by employing special materials, accurate heat treatments, minimum coupling tolerance between wheels and surface precise finish in order to reduce the drawbacks mentioned above.

External gear pumps (Figure 4.5) are essentially made up of two twin gearwheels geared with each other that are held in a totally smooth stator housing so as to prevent leakages between moving and fixed parts; the 8-shaped bearings, held in the stator along with gears, counterbalance side hydraulic thrusts by means of dedicated seals.

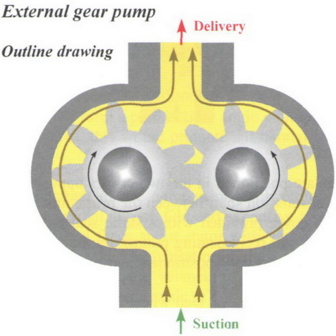


Figure 4.5

smaller) helps reaching the maximum pressure. Special stainless steel systems for water applications are available.

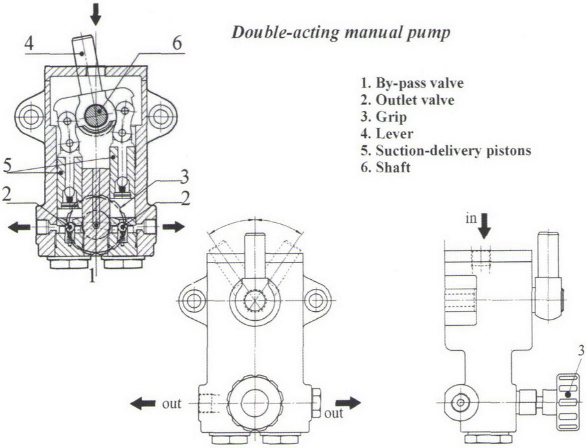


Figure 4.4

The following list includes only few applications of manual pumps:

- ✓ Jacks for general lifting purposes (short stroke)
- ✓ Pullers
- ✓ Bolt cutters
- ✓ Parallelogram platforms
- ✓ Emergency pump in series with automatic pump (for instance, if there is no tension the machine must be handled for safety reasons)
- ✓ Tensioning of steel ropes
- ✓ Trolley jacks (essential in repair shops)
- ✓ Portable cranes (for short-range movements of engine test benches, containers, etc.)
- ✓ Small presses for occasional deep drawing or bending
- ✓ Flexible hose fittings
- ✓ Benchtop pipe cutters, bending machines and tube closures
- ✓ Wine torques (for small-sized businesses because large-sized companies use complex pneumatic systems)
- ✓ Laboratory tests on pressures up to 2000 bar

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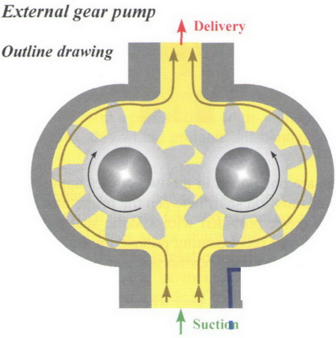


Figure 4.5

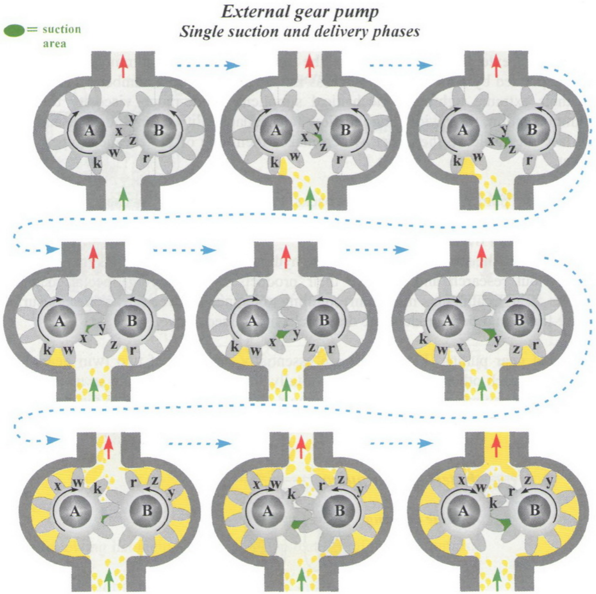


Figure 4.6

Bearings or bushings are positioned in the two covers opposite the stator where gear spindles revolve; the transmission shaft between the prime mover and the pilot wheel is in the hole of the front cover provided with a static seal (see Figure 4.9 and Figure 4.10). Like in most rotary and reciprocating pumps, the inlet has a larger bore than the outlet and both of them are usually positioned one opposite the other on the stator housing.

When the motor is started (Figure 4.6), lead wheel A, firmly connected to it, makes the driven gear B move into the opposite direction. The rotation of the teeth that come out of the mesh entails a vacuum whose volume is equal to the space between the two teeth; in other words, as shown in the Figure, the tooth x that was previously in the space between teeth y and z rotates clockwise, thus clearing a space (previously inexistent, hence a vacuum) equal to this volume.

During this very short phase (rotational speed can range from few hundreds revolutions per minute to 3000 rpm), the fluid pushed by the atmospheric pressure on the free surface of the tank fills the space between the teeth w and k of gear A and it is driven clockwise to the outlet. In the following phase, the vacuum generated by the tooth y allows teeth z and r of gear B to drive anticlockwise the oil to the right part of the stator. Obviously, these phases alternate so as to conclude the revolution.

When the actuating circuit is operated, delivery pressure should be the same throughout the whole upper chamber; actually, the inevitable leakages cause the pressure of the whole active part (oil transferring chambers) to be proportionally distributed and to decrease from the outlet to the first space after suction.

The pressurised fluid inside the pump subjects gearwheels to considerable radial loads (Figure 4.7).

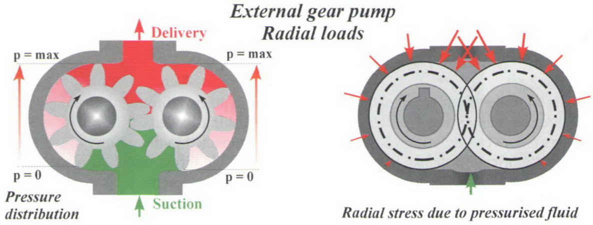


Figure 4.7

Another problem is the accurate positioning of gear plane faces vis-à-vis cover surface plates. If they do not form a perfect right angle with tolerances even less than 2-3 thousandths of millimetres, this would promote substantial leakages, early wear and the ensuing seizing.

That is why it is important to use components with more suitable tolerances and methods to limit axial and radial thrusts. Old and expensive types of gear pumps (made up of perfectly smooth lateral faces and a stator with very precise shim adjustments) were replaced by radial and axial compensation pumps, also known as **self-balanced** pumps, which consist of a casing and balancing bushings made of aluminium alloy, treated steel gearwheels and front seals made of reinforced nitrile mixture or viton.

Axial and radial compensations are made possible by placing two balancing or compensation bearings opposite the plane faces of the gearwheels and axially floating between the covers and the gears themselves.

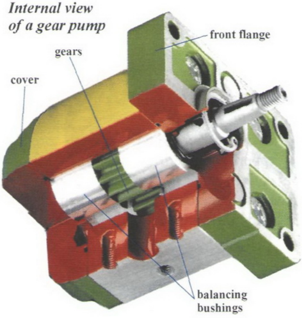


Figure 4.8

In **axial and radial balance**, the pressurised fluid, pushed through tiny and accurately measured openings between the outlet and the bushings, exerts a thrust on their two back parts (cover sides). This keeps them solidly connected to the wheel but it allows an adequate lubrication thanks to an accurate design of the thrust force. As a result, gears perfectly mesh with bushings while the lubricating film prevents the faces of the parts from wearing out; the spindles solidly connected to the gearwheels do not need more bearings and the single bearing, if used, is positioned over the end of the transmission shaft (Figure 4.10).

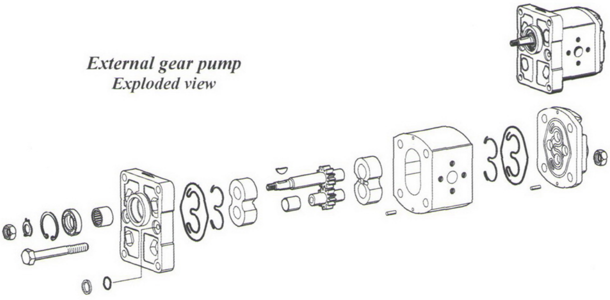
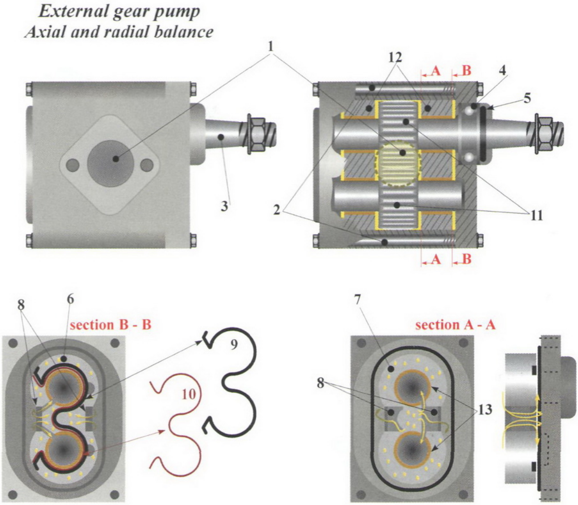


Figure 4.9



1. Outlet	**5. Oil seal unit**	**9. Balancing seal**
2. Tie rods	**6. Cover side bushing**	**10. Anti-extrusion ring**
3. Drive shaft	**7. Gear side bushing**	**11. Gears**
4. Bearing	**8. Balancing openings**	**12. Selected bushings**
		13. Bronze bearings

Figure 4.10

Radial balance design must also take into account the fact that teeth in mesh cannot fully expel oil. As a result, tiny fluid drops are 'squeezed' between the engaging wheels, thus entailing (depending on the incompressibility of the liquid) local overpressures that act in a radial manner vis-à-vis gear shafts. In addition, during gear disengagement, the volume between the teeth suddenly increases even before the contact with the sucked fluid.

Consequently, the central part of bushings must have some interstices in order to discharge this fluid; these are the only points where delivery and suction areas come into contact and the overpressure-prone fluid discharges in micro-areas subjected to early vacuum (Figure 4.11).

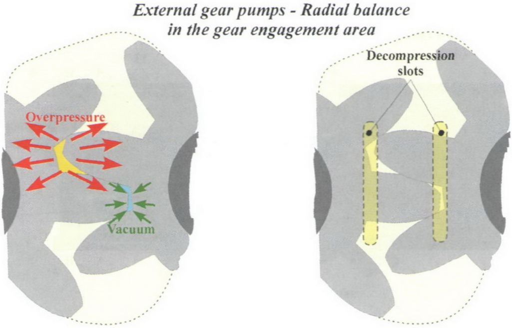


Figure 4.11

Figure 4.12

Bushing openings (Figure 4.12) play another important role as they allow leakages to pass from the delivery area to the suction area where the leaked fluid mixes with the fluid from the tank.

The 3-shaped seal sets the balance area and separates the suction area from the delivery area. It is supported by an anti-extrusion ring, with the same shape as the seal, so as to avoid the extrusion of the seal parts where it is not supported due to play.

The pump leading gear is generally set to revolve clockwise; anticlockwise revolution occurs when the back cover is disassembled and wheels are inverted (i.e. the leading wheel is replaced by the driven wheel and vice versa).

This operation must be carried out carefully in order to avoid forcing and the position of the balancing bushing vis-à-vis the housing must be marked so as not to reassemble it wrongly; its positioning vis-à-vis the inlet and the outlet is essential as well (Figure 4.13).

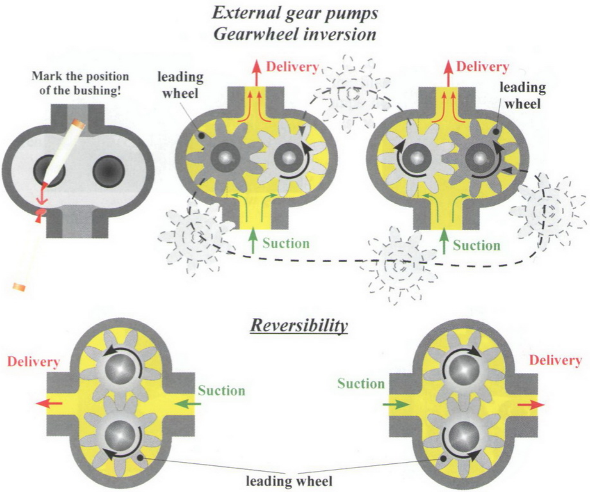


Figure 4.13

Some systems need the direction reverse of the prime mover but their wheels cannot be inverted; under these circumstances gear pumps known as **reversible** pumps are employed because their inlets can turn into the outlets and vice versa as they have the same bore.

Standard gear pumps (Figure 4.14) usually have the following characteristics:

- Working pressure up to 250 bar
- Peak pressure up to 280 bar
- Rotational speed 500 ÷ 3000 rpm
- Absolute suction pressure 0.75 ÷ 0.85 bar
- Fluid viscosity 15 ÷ 150 cSt
- Overall efficiency 0.75 ÷ 0.85
- Contamination sensitivity
- Loud noise at high pressures

Some versions have *two or three pairs of gears*, which allow them to have only one inlet but to exploit more outlets, thus avoiding the coaxial system.

External gear pumps

Figure 4.14

The type we are now going to describe should not be mistaken with the double staggered gear pumps. We can see in Figure 4.6 (Single suction and delivery phases) that the fluid flowing out of the upper outlet cannot be delivered uniformly. As a matter of fact, the oil held in each meatus is alternately driven by the right and left teeth, thus causing a pulsating delivery along with slightly variable pressures, noise and vibrations. Luckily, the cycle is not remarkably pulsating, so it does not have the trend shown in the left part of Figure 4.15: as previously explained, the expulsion of the last drops from a specific tooth coincide with the beginning of the first phase of a tooth of the other wheel. Actually, the curve trend in standard pumps depends on the number of gear teeth: the more teeth a gear has, the less it pulsates; yet, a high number of teeth cause different types of technical problems.

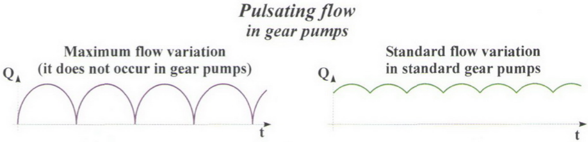


Figure 4.15

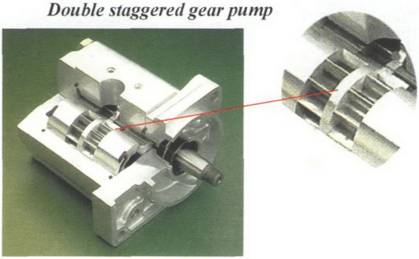


Figure 4.16

In double staggered gear pumps (Figure 4.16), the teeth of the second pair of gears are offset by a half tooth spacing (a tooth of the second pair is arrayed in the middle of the space between two teeth of the first one) and divided from the other by a sheet with a clearance hole. Leading wheels are axial and solidly connected to a single transmission shaft and obviously the casing has a single outlet. The double pair of staggered gears reduces pulsations by about 70 % during delivery.

If noise is too loud and yet we want to use this very type of pump, we can resort to external helical gear pumps: they are quite noiseless but more expensive and they can sustain a lower nominal pressure than standard pumps because of the substantial loads between the bushings and the teeth (Figure 4.17).

A famous international company has recently launched a special gear pump that partially meets the need for *variable flow*. A single casing holds two separated external gear pumps that share delivery and suction hoses; the parts are mutually axial and connected to the single transmission shaft. If the hydraulic circuit requires a limited flow, the flow is generated by a single pump, while the other pump is on standby due to its dedicated valve. When more flow (hence more power) is needed, a Load Sensing signal switches the valve of the second pump thus increasing the flow. However, this device cannot be classified as a variable flow pump because flow distribution, albeit automatically, is affected by the displacement of the two pumping parts: if both of them, at a certain rpm, pump 10 l/min, system flows can amount to as little as 10 or 20 l/min.

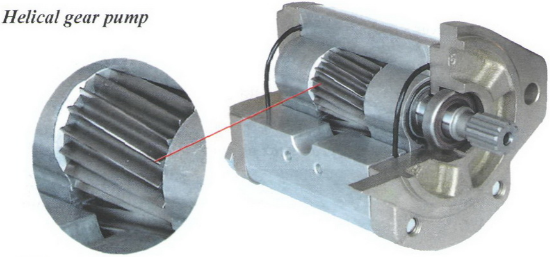


Figure 4.17

Gerotor pumps

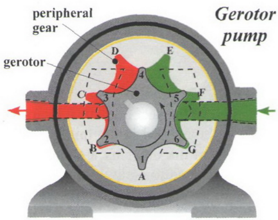


Figure 4.18

Gerotor pumps consist of a central (external-toothed) male gear, solidly connected to the transmission shaft and an internal-toothed (female) gear that can revolve and that is held only on the seat of the pump housing. The internal male gear (usually made up of not more than 6-8 teeth) always has one less tooth than the peripheral female gear; for instance, if an internal male gear (the real Gerotor) has six external teeth, the gearwheel will have seven internal teeth. This is why a specific Gerotor tooth meshes with its corresponding internal tooth of the female gear of the external wheel only once during the whole revolution.

Assume the pump is revolving and tooth 1 of the Gerotor is fully engaged with the internal tooth A (Figure 4.18). During the first half of the revolution, since the female gear has larger teeth, a wider and wider space originates between the gears resulting in the vacuum needed during suction. During the second half the space shrinks progressively and oil can be driver into the outlet. Obviously the same cycle occurs as each tooth alternates: after 1/6 of revolution tooth 2 meshes with B, then 3 meshes with C and so on.

Gerotor flanging
employed in auxiliary and booster pumps and mounted on the main pump

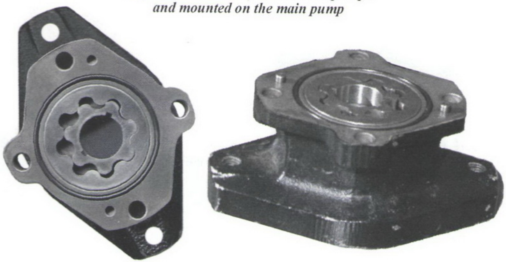


Figure 4.19

Gerotor pumps cause little noise, need almost no maintenance, are light and quite small, have little pulsating flow, few components and an easy and compact casing. They are suitable for revolutions ranging between 500 and 3500 rpm, their volumetric efficiency is low but adequate for the specific functions and the recommended viscosity range is 20-180 cSt. Since Gerotor pumps cannot sustain more than 50 ÷ 70 bar, they are confined to lubrication circuits and to serve as booster pumps directly mounted on the main pump (Figure 4.19).

Some of the newest pumps, which are still at an experimental stage, managed to reach working pressures above 150 bar with an overall efficiency of 75% thanks to the enhanced compensation of the axial play.

Another interesting pump that is similar to a Gerotor pump under some respects but that cannot be classified as Gerotor (it has more standard-shaped teeth) has long been available and it is employed in many applications.

Under other respects, this hydraulic generator is more similar to standard internal gear pumps (see next paragraph), yet it cannot fall into this category because it does not have a component known as 'prefill' and it has much lower working pressures.

This component, referred to as '*low-pressure internal gear pump*' in manufacturer catalogues, is characterised by a peculiar innovation. An insert made of a deformable material is embedded in the external part of each tooth of the peripheral wheel; the insert balances the tolerance between the teeth of the ring wheel and the central wheel solidly connected to the transmission shaft, thus ensuring a firm tightness between suction and delivery chambers (Figure 4.20).

Low-pressure internal gear pump

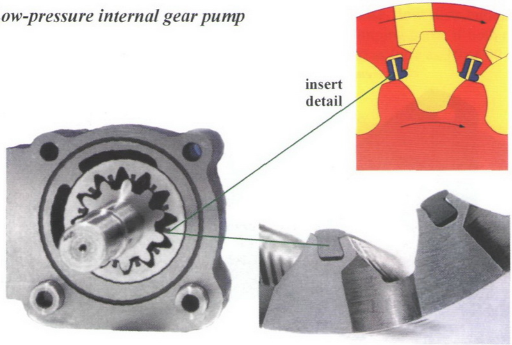


Figure 4.20

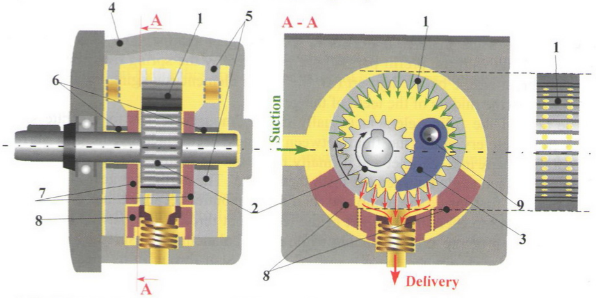
Low-pressure internal gear pumps are available in different versions with displacement ranging between 32 and 200 cm^3/rev, pressures between 60 and 125 bar and a speed between 400 and 3500 rpm. Both volumetric and overall efficiency amount to about 90% under optimum conditions, absolute suction pressure ranges between 0.6 and 0.8 bar, it is sensitive to fluid contamination (inserts subjected to wear). They are really noiseless and their viscosity range is very interesting.

Internal gear pumps

The main features of internal gear pumps are noiselessness, limited flow pulsations, higher pressures than similar external gear pumps, high volumetric and overall efficiency. Their main drawbacks that affected their popularity are, above all low flow rates and a more complex design than external gear pumps (resulting in higher costs).

Present versions have accurate axial and radial balance, as well as an enhanced prefill part; they can reach working pressures above 300 bar, volumetric efficiency above 95%, overall efficiency above 90%, rotational speed between 400 and 3500 rpm (or 300 l/min at about 1500 rpm in large versions with 200 cm^3).

Internal gear pump



1. Peripheral wheel
2. Internal wheel
3. Prefill part

4. Pump casing
5. Bearings
6. Bronze bearings

7. Axial balance
8. Radial balance
9. Prefill part pivot

Figure 4.21

Internal gear pumps

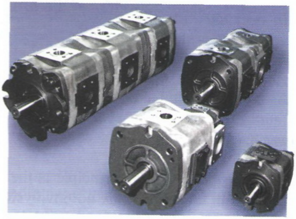


Figure 4.22

The clockwise revolution of the internal gear (2) (Figure 4.21), solidly connected to the transmission shaft and equipped with bronze bearings (6) mounted on flexible bearings (5), meshes with the peripheral gearwheel (1), which entails fluid suction almost in the same manner it occurs in external gear pumps. The engagement and position in line with the prefill part (3) are promoted by the spring over the outlet that exerts a thrust on the balancing part (8); the prefill element, swinging around the pivot (9), moves to the middle of the teeth of the gears.

The task of the filling element (usually called "moustache") is to separate suction areas from delivery areas: the fluid between the gear teeth, held by the gear surfaces, cannot flow back into the inlet and it finally flows out through the outlet after travelling a short rotational angle between the teeth themselves. The peripheral gear (1) allows the fluid to flow, first inside the pump and then into the outlet because it has clearing holes that correspond to each internal part of the teeth (root throat). The parts (7) adhering to the wheel (1) promote axial balancing while radial balancing is ensured by the parts (8) moved by the spring when the pressurised fluid is sent through tiny clearances in the lower part.

Lobe pumps

Two gears, one of which is axially connected to the prime mover, are rigidly assembled respectively to each lobe. When the lobes revolve, they suck the fluid and drive it to the outlet (Figure 4.23).

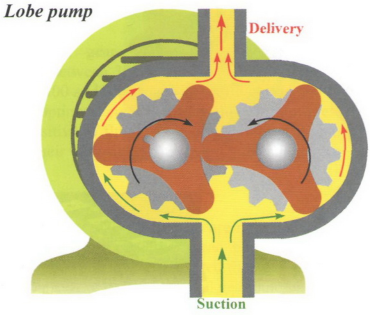


Figure 4.23

All in all, the operating principle is the same as for external gear pumps (note that gearwheels in lobe pumps are not involved in the hydraulic operation because they just transmit the rotary motion from one lobe to the other): as lobes revolve in opposite directions at the inlet, this causes a vacuum that makes the fluid flow in; the fluid is then driven alternately by the left and right lobe to the outlet.

In order to avoid leakages, lobe coupling must be very precise, which means these pumps are employed only in circuits with very decontaminated fluids so as to prevent lobe erosion. In addition, these pumps cannot sustain high pressures and are subjected to rather strong pulsations due to the large space between lobes. This is why lobe pumps are rarely used in oil hydraulics and cheaper external gear pumps are employed instead.

Screw pumps

What follows is a list of the remarkable characteristics of screw pumps (Figure 4.24):

- ✓ Easy manufacturing process
- ✓ Working pressure up to 120 and peak pressure up to 200 bar
- ✓ Up to 5000 rev/min
- ✓ Very noiseless
- ✓ Little vibration
- ✓ Wide range of the grade of kinematic viscosity
- ✓ Almost no pulsations
- ✓ Medium-high flow rates
- ✓ Absolute suction pressure equal to 0.9 bar
- ✓ Overall efficiency of around 75% under optimum working conditions
- ✓ Compatible with most hydraulic liquids
- ✓ Long-lasting
- ✓ Medium cost

Some of these characteristics are, on the one side, a plus for this type of pumps and, on the other, a drawback because they limit their fields of application. For instance, high flow rates drop dramatically when pressure goes up, even if it does not reach high levels and it compares only with the characteristics of gear pumps.

Screw pumps are better then external gear pumps in some respects: they are less noisy and pulsating, fewer vibrations occurs, they are compatible with similar or more viscous fluids, they last longer and have higher flow rates at medium pressures. Conversely, external gear pumps are more compact and compatible mechanically speaking, they are manufactures by many companies in a number of versions with different flow rates; from the economic point of view, although their price is not exorbitant, screw pumps are more expensive than similar gearwheel pumps.

These units are found in the versions with two, three or five rotors. In two-rotor pumps, a rotor is axial vis-à-vis the prime mover and it moves the other rotor via gears; In three- (Figure 4.25) or five-rotor pumps, the central leading screw transmits the revolution directly to the others. Three-screw pumps are usually employed in oil hydraulics.

The profile of driven screws is opposite to the profile of the leading screw and, like in gear pumps, the volume generated by the revolution of the screws in the inlet creates a vacuum that helps the fluid to flow up; subsequently, the fluid is translated to the outlet by the particular profile between the leading and driven screws.

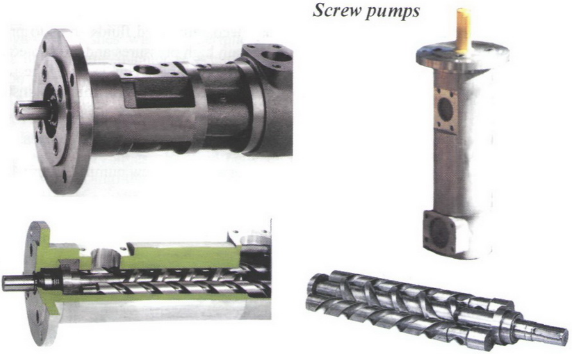


Figure 4.24

Screw pump - Outline drawing

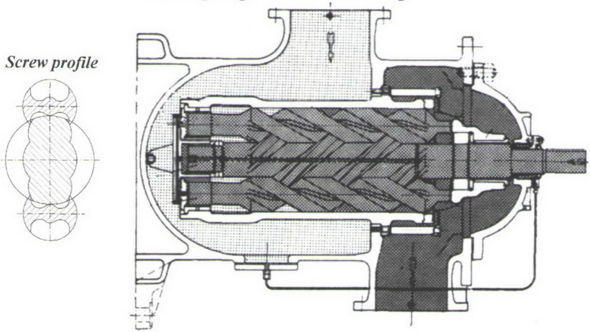


Figure 4.25

Balancing is not problematic because the pressurised fluid in motion fills the interstices between the screws and the pump casing, thus generating compensating radial forces; the translation is axial and the fluid found between the rotors avoids friction. A very accurate design of threads prevents the fluid from being squeezed between the parts in mesh (see gear pumps) and it also ensures the most constant flow rate among volumetric pumps; the slight pulsating flow virtually results from the compressibility of the liquid inside the outlet.

FIXED AND VARIABLE DISPLACEMENT PUMPS

The hydraulic pumps described so far are only fixed displacement pumps due to their design. Other types can be manufactured in fixed or variable displacement versions depending on their operational peculiarities.

Fixed displacement vane pumps

Fixed displacement vane pumps are popular among many self-propelled machines manufacturers and widely used in medium-power stationary applications because of their design simplicity, good value for money, reasonable efficiency and the ability of balanced versions to reach higher average pressures than other rotary pumps.

The essential parts of a vane pump (Figure 4.26) are the splined rotor solidly connected to the drive shaft, rectangular vanes arranged in a radial manner inside the stator and free to slide into the slots and the stator, which is eccentric vis-à-vis the rotor and mounted on the internal wall of the casing.

The operating principle of any vane pumps is the vane shifting into the slot, from the seat towards the eccentric stator. When the prime mover is started, the vanes at rest inside the slots tend to move towards the stator wall due to the centrifugal force. The effect of the centrifugal force should be considered carefully, except for some old versions where vanes were directly connected to the stator through springs inside the splines (nowadays they are still essential in systems that need quite few revolutions).
The centrifugal force is not enough to set vanes into motion below 600 revolutions per minute (a few dozens more or less depending on the materials and the design; the adhesion force of the vanes to the stator affects their life at a rotational speed above 2500 rpm.

In any case, the centrifugal force alone cannot guarantee a high efficiency, especially at rather low speeds (around 800-1000 revolutions). Vanes adhere to the eccentric stator thanks to the pressurised oil sent to their back head through a clearance inside the rotor that connects the outlet to the vane seats. During the inlet the vanes cause a vacuum that sucks the fluid from the tank, while during the delivery the liquid is sent to the outlet.

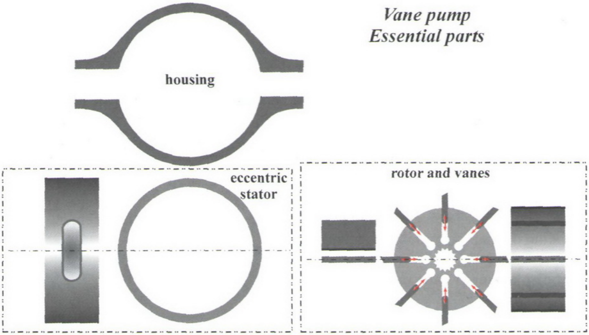


Figure 4.26

In order to ensure a perfect tightness with the stator, vanes have a sharp-edged end; the vertex must be positioned towards the direction of the revolution, as shown in Figure 4.27. If the vertex was in the opposite direction, fluid pressure would act on the vane end pushing it towards the seat.

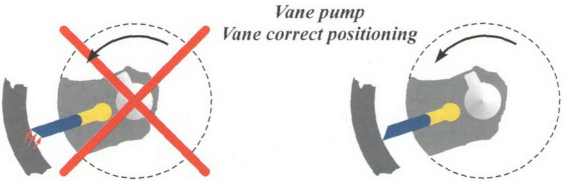


Figure 4.27

If we observe the revolution of a single vane in obsolete fixed displacement versions starting from the lower part (Figure 4.28), we can see that suction covers half round angle while the other 180° are devoted to delivery. This design promotes an excellent translation of the fluid but the whole right part, subjected to circuit pressure, triggers a substantial radial thrust on the rotor with the ensuing load unbalance.

As we are going to see later on, this system is still applied to variable displacement vane pumps that cannot be modified due to their design, yet better-balanced systems are preferred in fixed displacement versions.

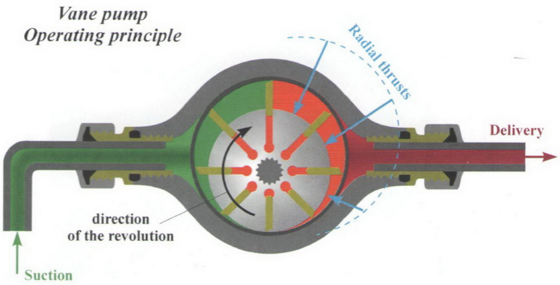


Figure 4.28

Parts of balanced vane pump

Figure 4.29

The balance of vane pumps is made possible by two diametrically opposed and mutually connected outlets and as many inlets. In this manner, the radial forces acting on an outlet are offset by the forces acting on the other (Figures 4.30 and 4.31). As a result, the hydraulic thrust counterbalance most of the radial unbalance and it promotes the dimensioning reduction of bearings.

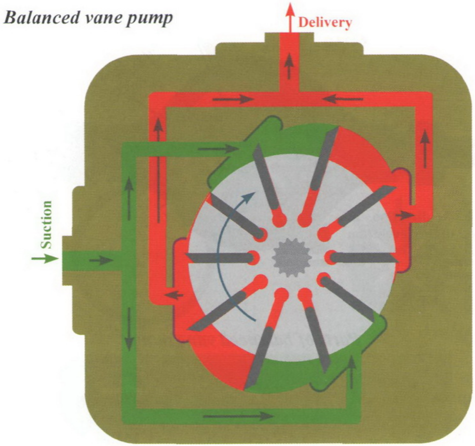


Figure 4.30

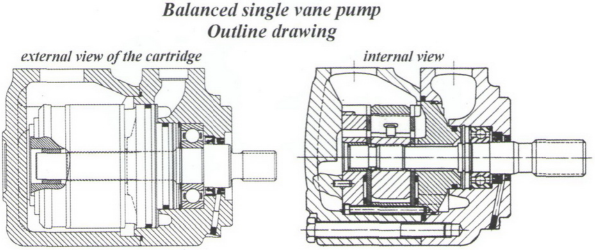


Figure 4.31

Vane pumps are usually set to rotate clockwise. Depending on the manufacturer (the revolution is usually limited by the position of the pins), anticlockwise revolution occurs by rotating the unit made up of vanes, stator and rotor (the direction of the revolution in all types of pump is viewed by the shaft end).

Vanes, rotor and stator unit of a balanced pump

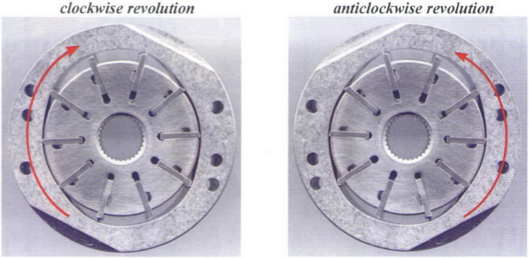


Figure 4.32

The unit (Figure 4.32) made up of vanes, rotor, stator, bushes and control heads is usually referred to as 'cartridge' (Figure 4.33).

Balanced vane pump cartridge

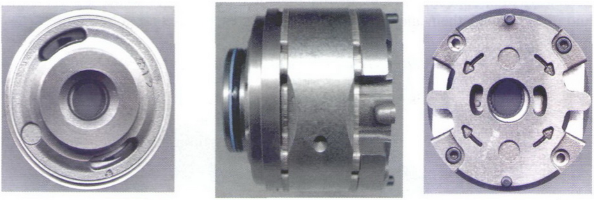
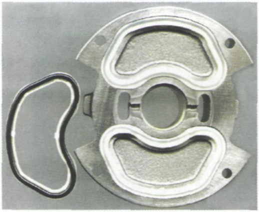


Figure 4.33

Bushes and heads, found on both sides of the rotor/stator unit, guarantee an adequate suction/distribution of the fluid and the axial balance of the system.
In particular, brass intermediate bushes or wear plates, as well as distribution slots, distribute through splines the thrust fluid over the bases of each vane and (in pin or intravane vanes – see later on) in their clearances. The pressurised fluid flows into the two basins, fitted with proper seals and an anti-extrusion ring, in the internal parts of the *pressure plates*, that channel it into the splines of the bushes (Figure 4.34).

Unlike the pressure in the fluid film between the bushes and the rotor/stator unit, the pressure in these basins offsets the pump axial loads.

Balance vane pump
Heads and cartridge opposing bush



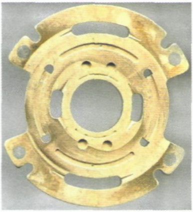


Figure 4.34

In everyday language, standard pumps with single inlet and outlet are referred to as '*round*' and radially balanced pumps as '*square*' (Figure 4.35) due to their shape.

Vanes must be balanced in order to ensure high performances. An effective system consists in putting **double vanes** with opposed sharp edge in the slots (Figure 4.36). The tightness between vanes and the stator virtually doubles and, in order to reduce the ensuing friction, opposite cavities are manufactured on the sides so that the pressurised fluid flows through them and, only in the delivery area, it offsets stator pressure.

External view of a balanced vane pump

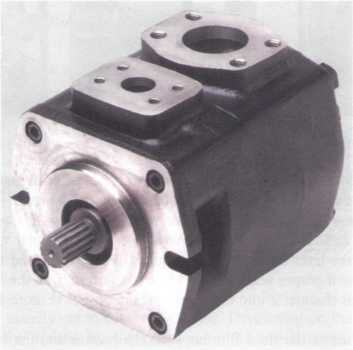


Figure 4.35

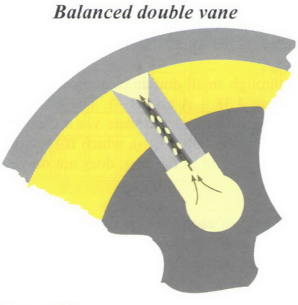


Figure 4.36

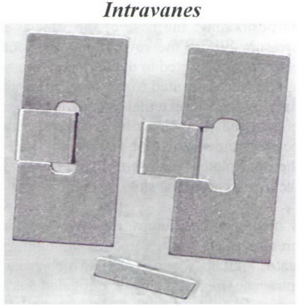


Figure 4.37

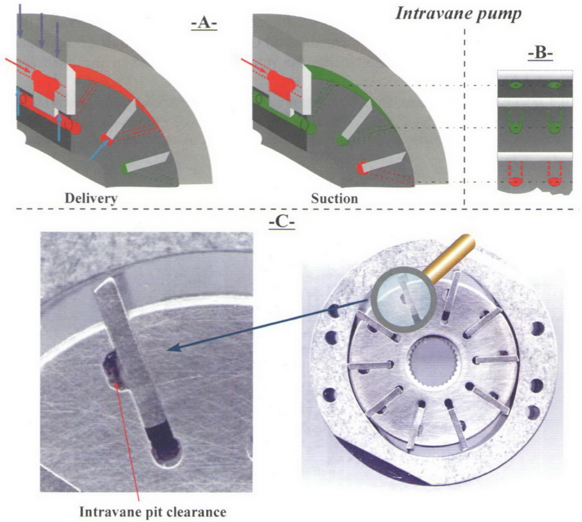


Figure 4.38

The pressure at the base of the vane in **intravane** systems (Figure 4.37) depends on its positioning: the base of the vane is alternately subjected to suction and delivery pressure during the revolution. The slot between the small insert and the vane is constantly subjected to delivery pressure.

The fluid is routed to the base of the vane through small double clearances between its seat and the external part of the rotor (Figure 4.38 – A and B –). The constantly pressurised fluid flows to the cavity between the insert and the vane via a clearance between the slot and the plane face of the rotor (Figure 4.38 – C –), which is routed by the wear plate. The insert, driven by the pressurised fluid to the ring, does not move in the lower part of the slot.

In single vane pumps, the thrust fluid at the base of each vane is always at delivery pressure. This ensures an optimum tightness, but it also results in a strong friction on the stator wall in suction areas, which makes it wear out quickly. This phenomenon in intravane pumps is quite mitigated because the vane in the delivery area is fully subjected to the working pressure while tightness during suction is guaranteed by the pressurised fluid found only in the cavity.

The operating principle of intravanes is similar to pin vanes. The pressurised fluid is sent to the base of the vane via two channels on the vane itself; the thrust pin pressure is constantly ensured by a channel inside the rotor and connected to the bush (Figure 4.39).

Like intravanes, the vane is not subjected to delivery pressure during suction and the pin makes it adhere to the stator. The lack of splines feeding the cavity and radial holes on the rotor that are typical in intravanes ensure more sturdiness to the hollow vane pump.

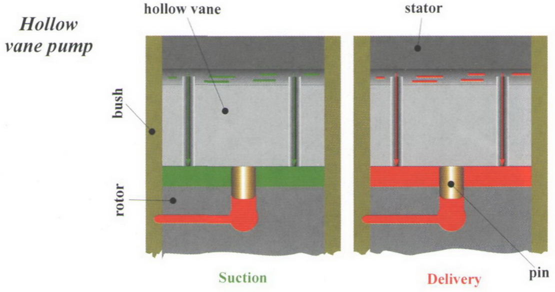


Figure 4.39

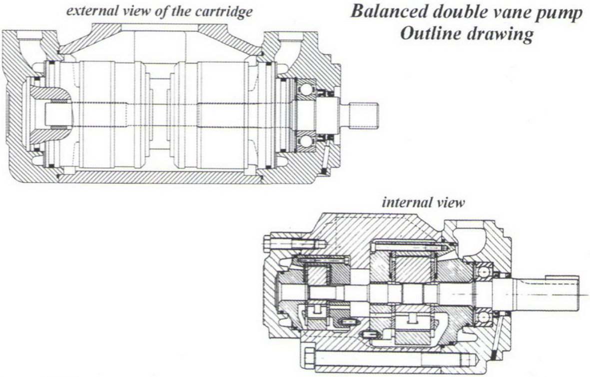


Figure 4.40

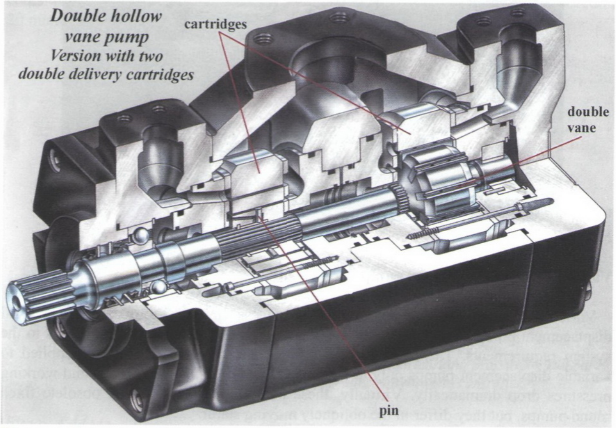


Figure 4.41

Balanced vane pumps are widely employed in self-propelled machines, especially in the double version equipped with one inlet and two outlets with different flow rates (for instance, earthmoving equipment).

More sophisticated versions with double hollow vane have the advantages of the pin thrust as well as the excellent tightness of the two vanes (Figure 4.41).

The most advanced versions of balanced vane pumps can be operated at maximum working pressures of 250 – 280 bar and sustain 1800 revolutions – 200 bar at 300 l/min, speed between 600 and 3000 rpm, overall efficiency of 0.85 and absolute suction pressure of 0.85 bar. The recommended oil viscosity ranges from 15 to 54 cSt under normal working conditions.

These particular types of hydraulic generators can also be started at a temperature well below zero. As a matter of fact, the pump can be started with a fluid viscosity of 800 cSt, with appropriate precautions recommended by manufacturers, like initial low speed. Obviously, under these conditions the whole system is operational when it warms up.

Double vane pump for mobile application

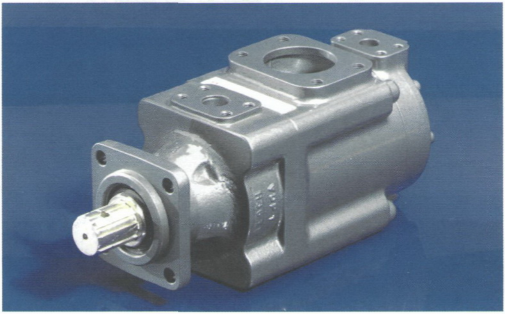


Figure 4.42

Variable displacement vane pumps

Variable displacement vane pumps are based on the same operating principle as fixed displacement pumps and they are the type of pumps whose flow rate best adapts to the system requirements. The balancing systems previously described can be applied to variable displacement pumps only to a limited extent, therefore efficiency and working pressures drop dramatically. Virtually, these pumps are similar to the obsolete fixed round pumps, but they differ in the obliquely moving stator.

Variable displacement vane pump Zoom in on the stator rotor vane unit

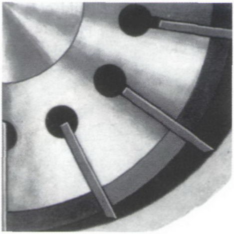
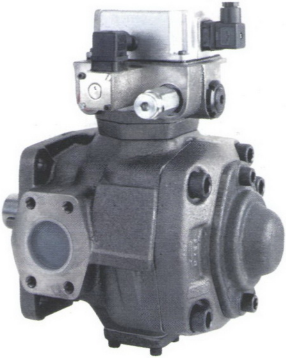


Figure 4.43

The versions equipped with a spring mechanical controller can sustain a working pressure of 80 - 100 bar with flow rates up to 150 l/min at 1500 rpm (displacement 100 cm^3) while an hydraulic servo-control (see next chapter) or an electrohydraulic controller (Figure 4.44) replacing the spring ensures working pressures of 150 bar. The recommended viscosity at 50 °C ranges between 25 and 45 mm^2/s, the fluid temperature must be between –10 °C and 70 °C, the primer (effect of the centrifugal force on the vane operation) occurs at 800 rpm and the maximum speed is 1800 rpm. Fluid contamination sensitivity demands decontaminations with filter meshes not exceeding 20 micrometres.



Large vane pump
Variable displacement

displacement = 100 cm^3
maximum working pressure = 150 bar
electrohydraulic controller

Figure 4.44

It is recommended that the pump should not be installed without the relief valve because once the maximum working pressure is exceeded (for instance when the directional valve is on the closed centres), it takes rather a long time to obtain the zero displacement position.

Figure 4.45

Small vane pump
Variable displacement

displacement = 16 cm³
working pressure = 210 bar
peak pressure = 250 bar
hydraulic controller

Special variable displacement pumps made of special materials and with more accurate balances, albeit with a low displacement (8 ÷ 16 cm^3), can sustain nominal pressures of 210 bar and peak pressures up to 250 bar (Figure 4.45).

The stator ring (3) can move and slide diagonally vis-à-vis the transmission shaft, but it is held back by both the force the controller spring exerts (9) and the control pin (6) connected to the outlet via a small channel (Figure 4.46).

The sliding block (2) can be adjusted so as to balance the pump hydraulically; this must be carried out when the system is operated by comparing the pressures displayed by the manometers positioned according to the manufacturer instructions (for example, with a hydraulic controller a manometer is connected to the outlet and the other manometer to the controller clearance). By acting on the sliding block balancing adjustment screw (2), there is an ideal balance as long as the differential pressure is equivalent to the pressure printed in a dedicated table in the use and maintenance instructions provided by the manufacturer.

Flow rate and pressure are respectively maximum and minimum when the stator ring is in the right position marked as x (Figure 4.47); the stator is held back by the controller spring (9) that opposes the force exerted by the spool (6). An increase in pressure inside the circuit results in an opposite and higher force than the spring force in the pin (6), with the ensuing left translation of the stator. The higher the system pressure is, the more the ring shifts.

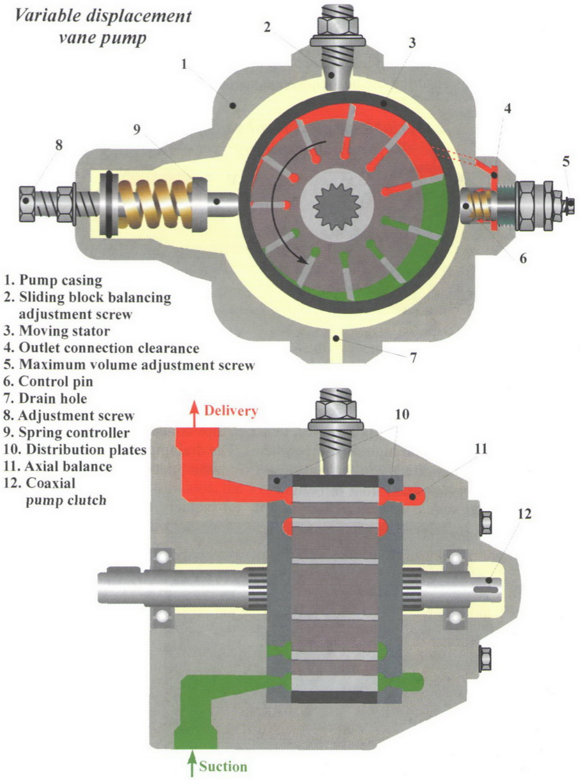


Figure 4.46

When the stator ring is in the left position *y*, the pressure of the system has exceeded the maximum level: the piston of a specific cylinder is at the end of the stroke or the directional valve is at an intermediate position with closed centres. In theory the pressure is maximum, the flow rate is zero and little power is absorbed, that is as much power as it is needed to keep the rotor/vanes unit sliding and overcome internal resistances. Actually, there is no flow in the user circuit, but it continuously circulates throughout the pump. The fluid leaked through the sliding parts flows back into the tank via a drainpipe, thus ensuring the heat exchange needed and an excellent lubrication.

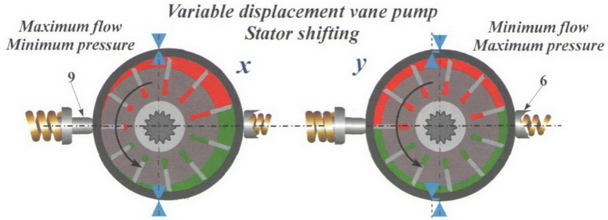


Figure 4.47

Some fundamental steps must be taken in order to install the pump in the tank correctly. As many manufacturers recommend, it is advisable to position it with the controller placed upward (see Figure 4.48); motor pumps are usually mounted below the fluid level in high-flow and high-pressure systems. The drainpipe must not be connected to the output hose, but directly to the tank, away from the inlet and below the minimum level in order to prevent foam from forming.

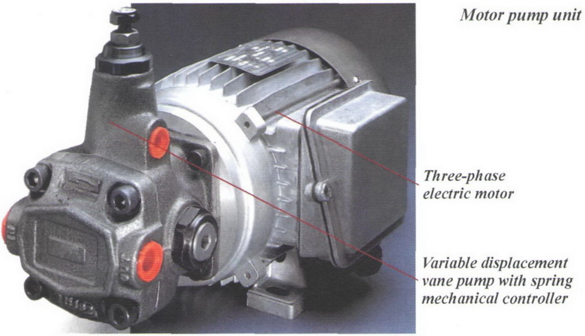


Figure 4.48

Besides the measures on shaft alignment and antivibration plugs that have already been described, other actions should be taken, like adding an air bleed and, after the installation, prefilling the pump via the drain hole (Figure 4.49).

Installation of a variable displacement vane pump

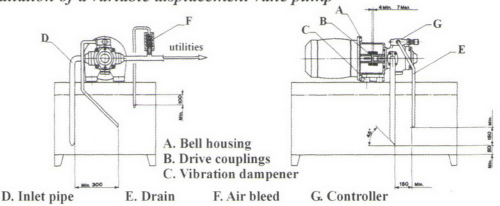


Figure 4.49

The typical stationary application of variable displacement pumps is on machine tools that require pressures of about 120 – 130 bar (hydraulic compensator), 70 – 80 bar (mechanical controller) or, in many cases, much lower pressures.

Piston pumps – Preliminary remarks

Considerable flow rates, optimum efficiency, the ability to reach high pressures and, in variable displacement versions, the availability of many and precise controllers make piston pumps the best pumps for fluid power. Piston pumps guarantee higher performances than the pumps described so far, but they are more expensive, more complex and need more maintenance.

Besides manual pumps, piston pumps can mainly be classified as radial and axial pumps. The cylindrical pumping parts in radial piston pumps are arranged in the shape of a star; in some cases, they are rotary and held in a block solidly connected to the transmission shaft, while in others they form a fixed radial block and are moved by an eccentric shaft. Axial piston pumps have different pistons aligned and arranged in the shape of a circle; the cylinder block can be in line with the axis of the shaft connected to the prime mover and it is either rotary and solidly connected to the shaft or fixed while the shaft rotates. The so-called inclined plate ensures the alternation of suction/pumping. In bent axis pumps, the plate is perpendicular to the drive shaft and the transmission of power to the pumping block is carried out by connecting rods, cardan joint or angle gear.

The phenomenon of pulsating flow dealt with in the paragraph on rotary pump is even more troublesome in reciprocating piston pumps because the design of their pistons cannot allow a totally constant flow.

Flow oscillation depends on the number of pumping parts. If by 'cycle' we mean the whole stroke of a piston (suction and delivery), the delivery cycle in a manual pump with a single pumping part (single-acting pump) does not occur during the first half (suction), whereas during the second half the flow rate soars from zero to the maximum level and then diminishes to zero again. Flow in double-acting pumps (two pistons with opposite motions) is incessant but very pulsating (Figure 4.50).

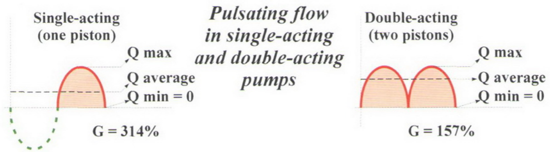


Figure 4.50

The more right phase timed pistons there are, the less the flow oscillates (Figure 4.51).

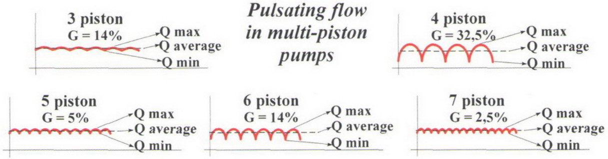


Figure 4.51

The ratio between the flow variation ($Q_{max} - Q_{min}$) and the average flow rate $Q_{average}$, is the irregularity grade G.

$$G = 100 \bullet \frac{Q_{max} - Q_{min}}{Q_{average}} \ (\%)$$

Given a single-acting manual pump that has an average flow of 3.18 l/min and a maximum flow of 10 l/min, the irregularity grade G is:

$$G = 100 \bullet \frac{10 - 0}{3.18} = 314\%$$

If we compare the oscillating trend of three-piston pumps whose G is 14 and four-piston pumps whose G is 32.5 or six-piston pumps whose G is 14 and seven-piston pumps whose G is 2.5, it is clear that the irregularity is less substantial in odd-numbered piston pumps because flows are more overlapping.

Radial piston pumps - Fixed displacement

Radial piston pumps with **rotary radial block and eccentric reaction ring** are made up of a series of (usually odd-numbered) pistons, arranged in a radial manner and held in a block solidly connected to the prime mover via a cruciform joint. Piston ends are attached to ball socket joints fixed to the sliding blocks resting on the reaction ring (Figure 4.52). This ring can revolve freely on a large bearing fixed to the pump casing; the distributor shaft, on the same axis as the drive shaft, is solidly connected to the rotary block.

When the pump is revolving, the centrifugal force in the suction area (and the possible pressure of the fluid if there is a booster pump) and the pressure in the delivery area push pistons towards the reaction ring that avoids the friction of the sliding blocks because it rotates on the bearing.

Some clearances inside the rotary distributor shaft connect the inlets and the outlets to their pistons, so that the fluid is sucked into the cylinders travelling the extent stroke during the first 180° of each revolution and then it is driven out via the outlet and the pressure port during the second half.

In radial piston pumps equipped with a **fixed radial block** and an **eccentric drive shaft**, the transmission shaft is eccentric vis-à-vis the pistons held by an adequately dimensioned spring.

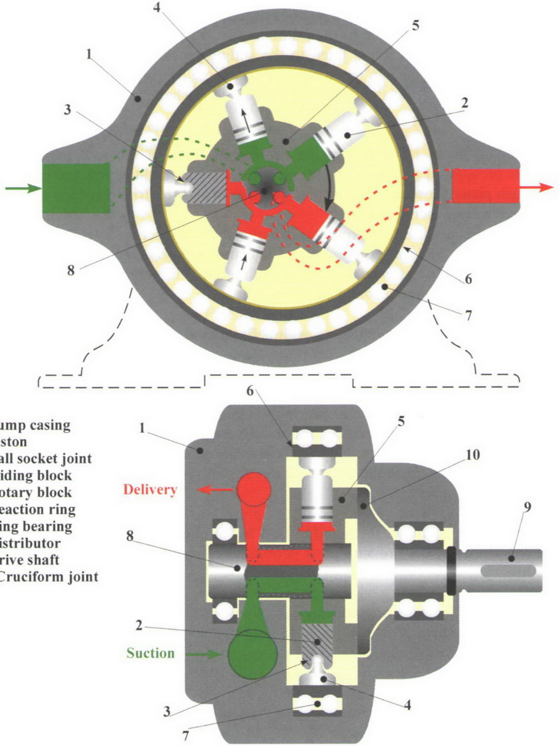
In the versions with a **bearing on the eccentric shaft**, the lower ends of each piston do not wear out thanks to the bearing. The shaft revolution causes the pistons to reciprocate, thus promoting suction and delivery (Figure 4.53). Suction involves the cross drains of the distributor that is coaxial vis-à-vis the drive shaft; the fluid inside the pump casing flows from the side clearings to the pistons and, if the suction valve is open, it fills its cylinder.

During the following phase, the piston presses on the fluid, thus switching on the delivery valve that allows the liquid to flow out through dedicated drains and the outlet. The connection of the pressure ports permits to connect more devices on the auxiliary openings.

This design demands a small number of low-bore pistons because the double ball bearing on the eccentric shaft can sustain only limited dynamic loads.

As a result, in order to reach higher pressures and flow rates, it is essential to replace the ball bearing on the eccentric shaft with **hydrostatic sliding blocks**. The fluid film between the eccentric shaft and the mirror-like part of the sliding blocks serves as a bearing absorbing the radial stress due to the pressurisation of the connected pistons. In this version, the pump, except for the sliding blocks, built and operated with the same characteristics as the previous one, can sustain better dynamic loads; this promotes a higher number and bore of pistons with the ensuing increase in pressure and flow (Figure 4.54).

Radial piston pump
Rotary radial block



1. Pump casing
2. Piston
3. Ball socket joint
4. Sliding block
5. Rotary block
6. Reaction ring
7. Ring bearing
8. Distributor
9. Drive shaft
10. Cruciform joint

Figure 4.52

Radial piston pump
Fixed radial block - Bearing on the eccentric shaft

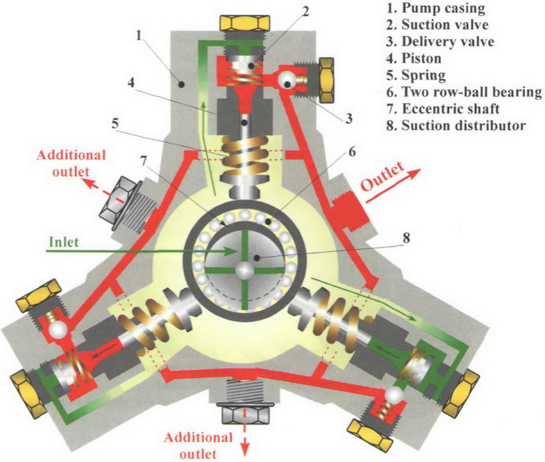
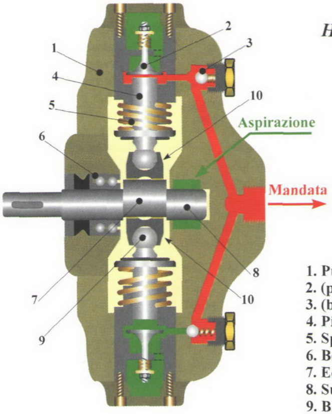


Figure 4.53

Radial piston pumps equipped with hydrostatic sliding blocks ensure flow rates up to 200 l/min, working pressures of 450 bar and peak pressures of 600 bar; rotational speed ranges between 100 and 2000 rpm, overall efficiency amounts to about 0.9, suction pressure is 0.9 absolute bar and the recommended viscosity range is 15 ÷ 50 cSt.

More than one piston row can be assembled in a single block in small and medium versions (Figure 4.55) so as to be able to choose between a single high-flow outlet and more outlets with different pressures and flows. Some companies manufacture pumps equipped with up to six rows.

Radial piston pump
Fixed radial block
Hydrostatic sliding block



1. Pump casing
2. (poppet) Suction valve
3. (ball) Delivery valve
4. Piston
5. Spring
6. Bearing
7. Eccentric shaft
8. Suction distributor
9. Ball joint
10. Hydrostatic sliding block

Figure 4.54

Radial piston pump
Double star

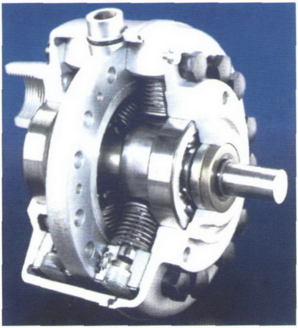


Figure 4.55

Radial piston pumps - Variable displacement

Variable displacement radial piston pumps are similar to rotary radial block pumps, but they differ in the reaction ring that can move (Figure 4.56). Their adjustment and operating principles are similar to variable displacement vane pumps: stator ring shifting is affected by the delivery pressure versus the force exerted by the spring of the hand wheel adjuster or the compensator.

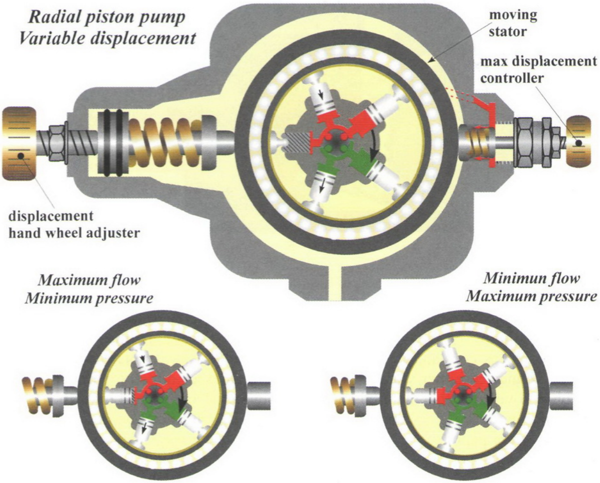


Figure 4.56

Axial piston pumps - Fixed displacement

Axial piston pumps are more popular than radial piston pumps and have different designs that mainly fall into two categories, depending on whether the cylinder block is in line with he drive shaft or bent axis type.

It is important to specify if the cylinder block is rotary or fixed in in-line piston pumps; piston reciprocation is made possible by a round plate inclined vis-à-vis the axis of the transmission shaft.

Swash plate pumps with fixed cylinder block are made up of a swash plate solidly connected to the transmission shaft (hence they revolve together) and a series of (ideally odd-numbered) pistons in axial line with the same shaft and firmly held in the pump casing. Suction and delivery valves ensure that the fluid flows from the tank to the actuators.

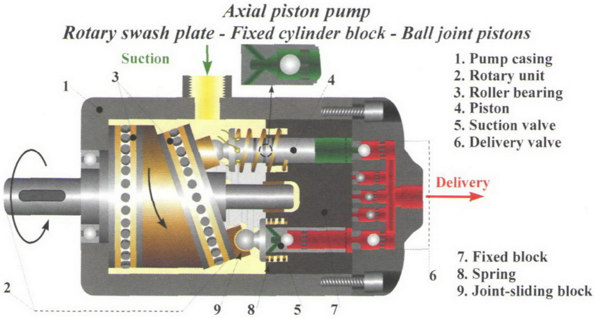


Figure 4.57

The revolution of the swash plate makes pistons move alternately; pistons are held by springs constantly connected to the plate. The swash plate and the pistons can be connected either via a ball joint/sliding block or directly, which means the piston is covered with a material adequately tempered and shaped into a hemisphere in order to reduce the contact surface. Likewise, the positioning of the return springs on the pistons depends on the design and suction and delivery valves often have no springs.

In the version of Figure 4.57, pistons are connected to the plate through a ball joint/sliding block and a roller thrust bearing, while the return spring placed in the front part of the pump casing pushes on the base of the ball joint. When the shaft is revolving, the angle of the plate makes pistons move. Oil flows through the space between the plate and the pistons and fills the sucking pistons, whose front end has some clearances connected to the internal space; in the following phase, the ball of the suction valve clogs the internal cavity of the front part, thus allowing the fluid to flow into the outlet through the delivery valve.

A different version of this pump has rounded front piston ends (less friction) connected to the swash plate through a spring placed inside the piston and back end of the pump casing (Figure 4.58).

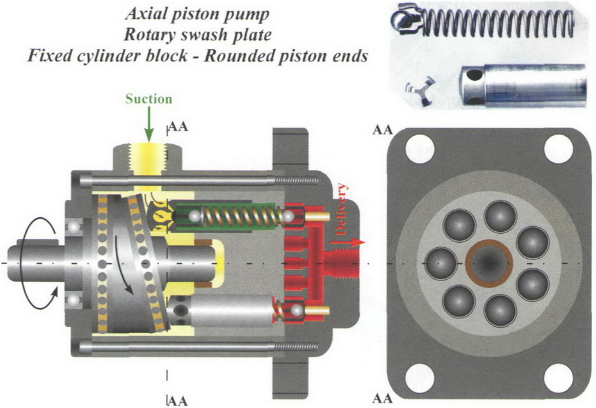


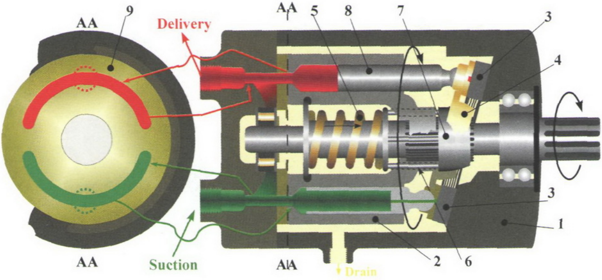
Figure 4.58

Suction/delivery valves (which are not found in bent axis versions or versions with rotary cylinder block), the long path of the fluid in the space inside the pump casing (this ensures abundant lubrication and excellent heat exchange) result in high performances with pressures up to 500 bar, revolutions between 1000 and 3000 rev/min, $\eta_g = 0.9 \div 0.95$ and suction pressures of 0.9 absolute bar. These generators are not much affected by dirty oil and require a viscosity range between 30 and 70 cSt.

Swash plate pumps with rotary cylinder block, whose mechanical operation is similar to the previous ones, differ in the cylinder block because it is rotary and rigidly connected to the transmission shaft. The swash plate is fixed on the pump casing and the slipping with the pistons is made possible by the ball joint/sliding block. The preload spring, whose task is to guarantee a solid contact between the sliding blocks and the plate, is usually twisted around the drive shaft (in the back part inside the pump) and it is held by a washer on the pump casing and another washer over the swash plate connected with some pins to a ring placed on the transmission shaft. The force on the sliding blocks is transmitted by the inclined ring (Figure 4.59).

The hydraulic principle of these pumps and fixed cylinder block pumps is not the same because when the pistons rotate suction and delivery are sent to their back end through the fixed intermediate bush (also known as distributor plate or wear plate) provided with suction and delivery slots ending in correspondence with the dead ends (maximum and minimum piston stroke). The openings of the head, the same as the bush at the beginning, end with the inlet and outlet; the bush is not found in many versions because the friction of the cylinder block is largely limited by the fluid film.

Axial piston pump
Fixed swash plate - Rotary cylinder block



1. Pump casing
2. Rotary block
3. Fixed swash plate
4. Sliding block holder
5. Preload spring
6. Spring pin
7. Thrust ring
8. Piston
9. Distribution bush

Figure 4.59

The fluid for the hydrostatic sustenance of the sliding blocks and internal lubrication (Figure 4.60) is pushed into a clearance in the front piston head, the joint and the sliding block; tiny drains in the sliding block drive the fluid into the pump casing.

Piston - Ball joint/Hydrostatic sliding block

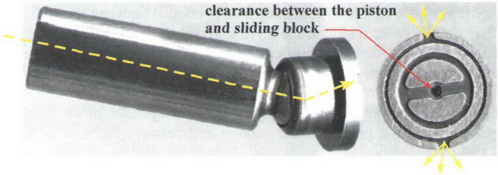


Figure 4.60

Bent axis piston pumps have a plate perpendicular to the drive shaft; the piston unit usually has an inclination of 25° and it is revolved by a cardan shaft, a system of connecting rods or a pair of bevel gears (Figure 4.61).

Axial piston pump - Bent axis

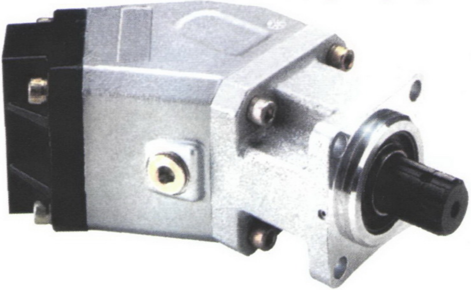


Figure 4.61

All the versions of these pumps need a quite robust dimensioning of the axial bearings of the transmission shaft because the cylinder block revolves without a mechanical support: the strain of the whole component affects roller bearings. The hydraulic performances of these pumps are similar to swash plate pumps with rotary cylinder block, but bent axis piston pumps are more robust; in addition, they are more expensive and obviously they cannot be mounted in tandem.

The plate in the versions with **connecting rods** is rigidly connected to the transmission shaft; the hemispheric seats for the connecting rods are on the face opposite to the shaft. These seats are made up of a small cylinder with ball ends; the plate ball has a larger diameter than the ball fit into the internal seat of the piston. The cylinder block can rotate thanks to the torsion movement on the pistons. The hydraulic principle is the same as swash plate pumps with rotary cylinder block, piston reciprocation results in the inlet/outlet phases and a bush with slots channels the fluid in the inlet/outlet.

A single-ball joint is in its seat at the centre of the plate and it presses on the preload spring that makes the cylinder block adhere between the plate and the distribution bush. In some types, the spring twisted around the small piston of the central joint presses between the upper part of the rotary block and a ring before the ball, while in others the preload spring is inside the cylinder block opposite to the small piston (Figure 4.62).

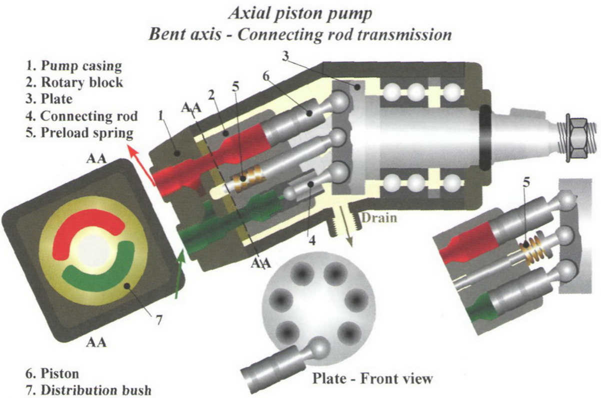


Figure 4.62

A more efficient solution to make the cylinder block rotate is the **cardan joint** (rotary shaft with joints at both ends - Figure 4.63) because it provides more traction than small pistons alone as it helps the spindle/preload spring. The preload system is now placed inside the cylinder block and the cardan joint end (Figure 4.64).

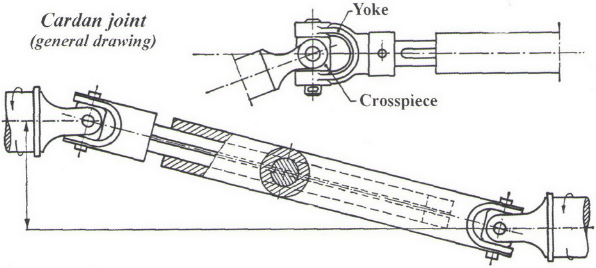


Figure 4.63

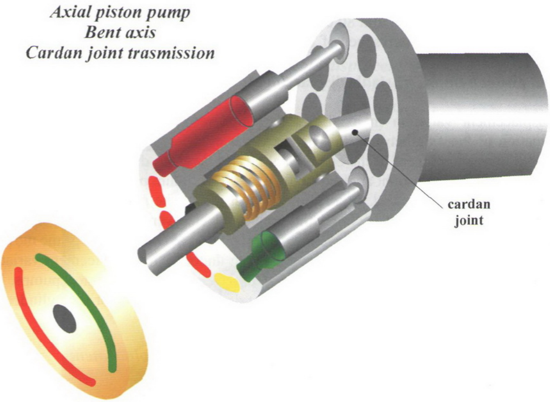


Figure 4.64

The plate and the upper part of the cylinder block in the versions with **bevel gears** become respectively the leading and driven gearwheels (Figure 4.65).

Figure 4.65

Their pistons are simpler as they have neither cylinders nor ball joints. Drain clearances directly connected to the inlet are manufactured inside the aluminium pump casing.

Bevel gear pumps, which are widely employed for commercial vehicles, usually have displacements between 40 and 80 cm^3/rev, working pressures of 350 bar, peak pressures of 400 bar and a maximum speed of 2000 ÷ 2500 rpm.

Axial piston pumps - Variable displacement

Displacement variation in axial piston pumps results from the adjustment of the inclination of the axis between the plate and the cylinder block. The variation must be possible in every moment of the operational phases and with displacements ranging from zero (pistons forming a right angle with the plate = maximum pressure, minimum flow) to the maximum angle allowed. In addition, it must always be possible (closed circuits) to reverse the flow (the inlet becomes the outlet and vice versa), always under the same conditions of variability without reverting the revolution direction of the prime mover.

Although every type of axial piston pump can undergo the displacement variation, it is usually performed in in-line or bent axis *rotary cylinder block* piston pumps. Figure 4.67 and Figure 4.68 briefly show the shifting device of the plate or the cylinder block; controllers are dealt with more in detail in the next chapter .

In-line axial piston pump - Variable displacement

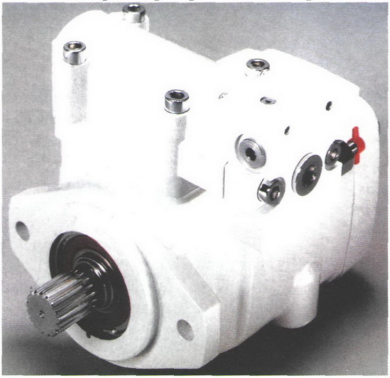


Figure 4.66

The design of variable displacement **bent axis** pumps differs from fixed displacement pumps only in the larger space of the mobile cylinder block.

In Figure 4.67 flow variation is performed manually through a hand wheel/worm screw. When the angle is maximum, flow too is maximum: the less the angle between the cylinder block and the axis of the transmission shaft is, the less the flow rate is; flow is zero as long as the two axis are in line. Flow direction can be reversed by reversing the angle: the former inlet becomes the outlet.

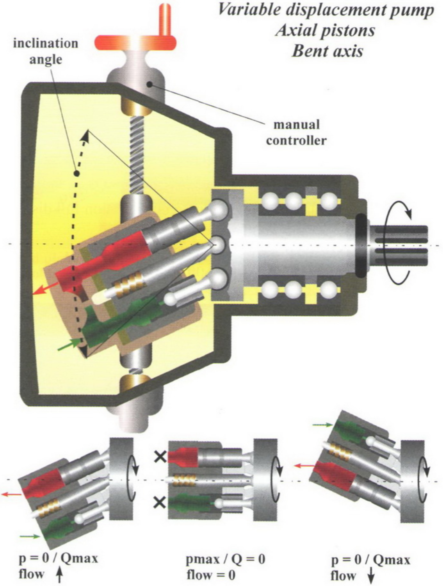


Figure 4.67

Assuming 25° is the maximum angle to which pump maximum displacement c_{max} corresponds, during the variation of the angle α, the displacement of the pump c_x can be determined by means of the following formula:

$$c_x = c_{max} \bullet \frac{\sin \alpha}{\sin 25°}$$

Given a bent axis pump whose maximum displacement c_{max} at 25° is 150 cm^3/rev, nominal pressure is 180 bar, volumetric efficiency $\eta_v = 0.9$, overall efficiency $\eta_g = 0.85$ and rotational speed of 1750 rpm, if the inclination is changed to 12°, its displacement $c_{12°}$ is:

$$c_{12°} = 150 \bullet \frac{\sin 12°}{\sin 25°} = 73.8 \text{ cm}^3/\text{rev}$$

The flow rates Q_{max} at maximum inclination and $Q_{12°}$ at 12° are:

$$Q = \frac{c_{max} \bullet rpm \bullet \eta_v}{1000} = \text{dm}^3/\text{min} = \text{l/min}$$

$$Q_{max} = \frac{150 \bullet 1750 \bullet 0.9}{1000} = 236.3 \text{ l/min} \quad Q_{12°} = \frac{73.8 \bullet 1750 \bullet 0.9}{1000} = 116.2 \text{ l/min}$$

Motor power has to be calculated by taking into consideration full displacement:

$$N = \frac{Q_{max} \bullet p}{600 \bullet \eta_g} \text{ (kW)} = \frac{236.3 \bullet 180}{600 \bullet 0.85} = 83.4 \text{ kW}$$

Variable displacement bent axis pump (inclination in one direction only for open circuits)

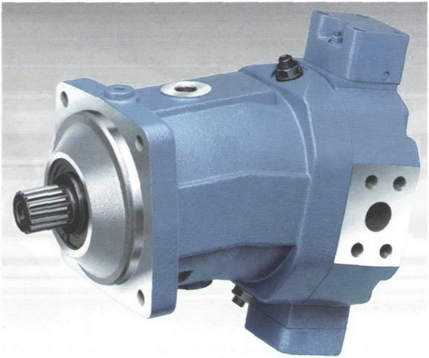


Figure 4.68

Also variable displacement pumps with rotary cylinder block in line with the drive shaft have very few differences in respect to the fixed type, like the mechanism that performs the inclination of the plate (Figure 4.69).

They are quicker than bent axis versions in the transients because the disk has less inertia than the cylinder block, they are cheaper to manufacture and can be mounted in tandem.

Variable displacement pump
Axial pistons - Rotary cylinder block

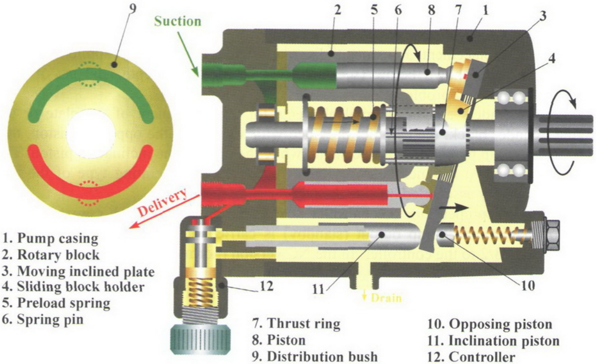


Figure 4.69

Variable displacement pump - Axial pistons
Internal view

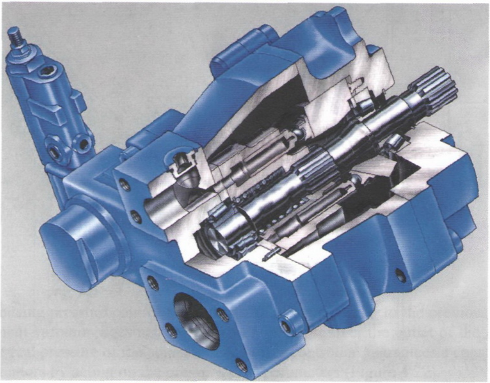


Figure 4.70

Displacement adjustment circuits

Except for manual control setting, displacement variation is performed by automatic controllers, which adjust pressure or flow, or all the parameters altogether, which keep power constant.

The main displacement adjustment diagrams, which are usually applied to axial piston pumps, are shown below (see next chapter for an in-depth analysis of each compensator and their problems).

The two cylinders are the internal elements for the adjustment of the plate or the moving cylinder block: the cylinder provided with a spring is the opposing piston, that keeps the pump in maximum displacement, while the other is the inclination piston. The valves connected to the inclination cylinder represents the operating type of the real controller, directly assembled outside the pump casing.

The pressure compensator or controller keeps the delivery pressure constant, albeit with flow variations. Pump output (standby) is carried out by adding a monostable solenoid valve 3/2 to the controller. In the NC version (normally closed solenoid valve), the solenoid excitation operates the standby; the NO (normally open solenoid valve) is excited during the operational phases and immediately starts the pump output then the voltage supply switches off (Figure 4.71).

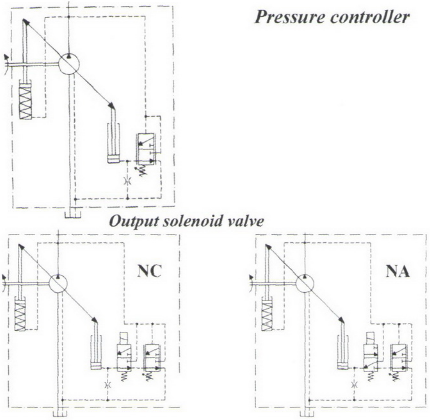


Figure 4.71

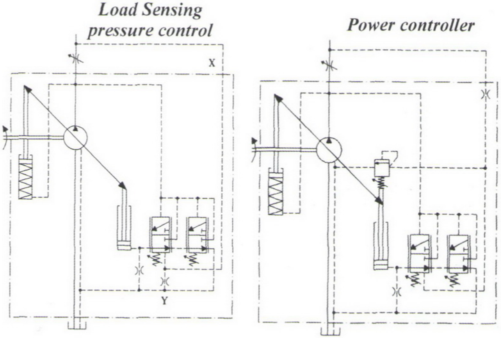


Figure 4.72

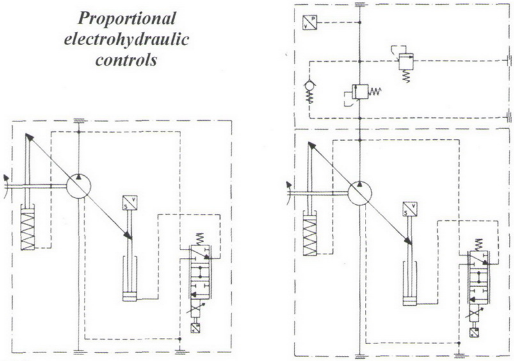


Figure 4.73

Load sensing pressure control performs tasks that are similar to the previous one, but displacement variation does not depend on the pressure over the outlet of the generator but on the real pressure of the actuator. The power controller guarantees a constant force on the actuators by acting on the pressure/flow parameters (Figure 4.72).

The electronic proportional control ensures better results (Figure 4.73). Transducers on the swash plate and the valve (depicted in the drawing on the rod of the inclination piston and next to the solenoid) guarantee high performances. More transducers and different hydraulic valves can adjust pressure/flow parameters in many different ways.

Axial piston pump
Variable displacement - Electroproportional adjustment

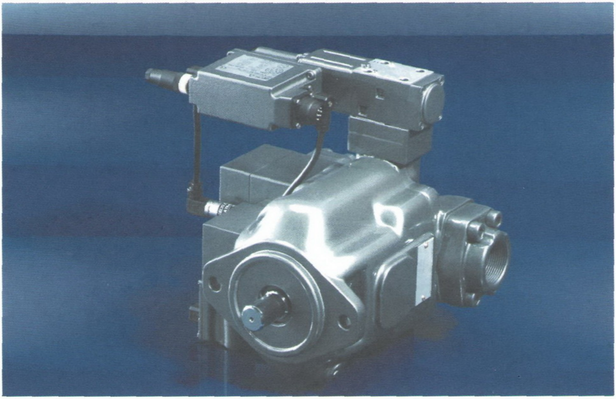


Figure 4.74

Integrated motor pump

The motor/pump unit can be installed as a single block in special applications. This special design has two important advantages: less space is needed as the coupling has no bell housing and the prime mover is cooled down because the oil is sucked through the motor itself (Figure 4.75). Yet, we must stress that in this device the heat resulting from the efficiencies of the circuit and the electric motor is yielded to the oil, which warms up quickly. This unit is therefore suitable for intermittent operation unless a good heat exchange is guaranteed in continuous operation.

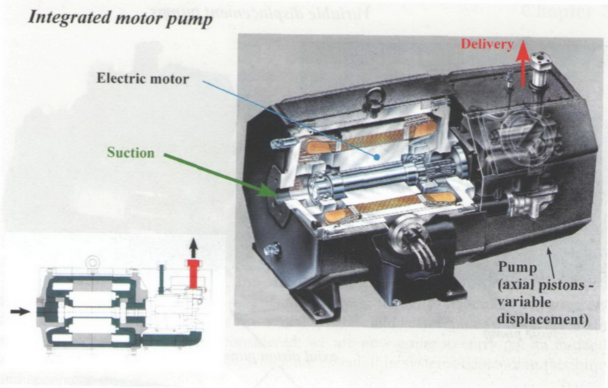

Integrated motor pump
Delivery
Electric motor
Suction
Pump
(axial pistons -
variable
displacement)

Figure 4.75

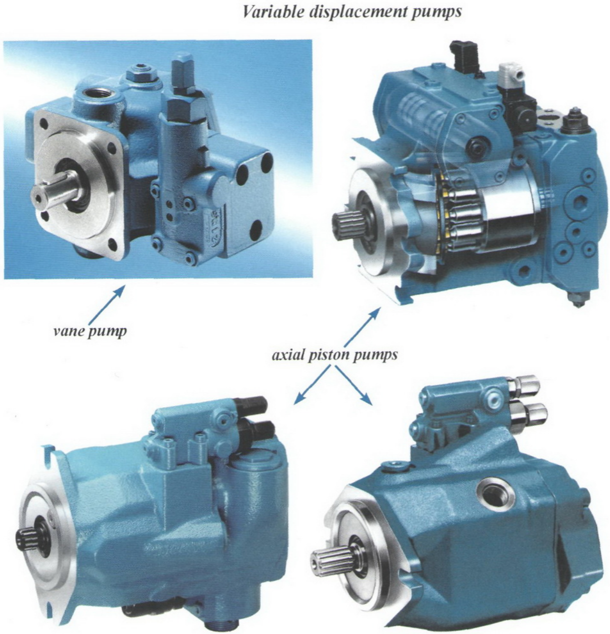


Figure 4.76