TYPES AND PERFORMANCE OF PUMPS AND COMPRESSORS

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Contents

Summary

The present chapter attempts to briefly describe an important application of fluid flow in turbo machines with emphases on pumps and compressors as two of the more important machines that are used in our everyday life .These can be as small as a fist or as large as hydro power plants.

The book makes distinction between the various types based on theory of operation such as centrifugal Pumps and Positive Displacement Pumps. Classifications based on Mechanical Efficiency, Net Positive Suction Head – NPSH are highlighted. The more general affinity laws are briefly described with some solved examples.

An important phenomenon that occurs frequently in pumps, namely cavitation is analyzed .Pumps operations and performance characteristics are also discussed. Compressors are briefly described as they bear the same basic features as pumps with more description of performance and designs.

1. Classifications of Pumps

1.1. Selecting between Centrifugal Pumps and Positive Displacement Pumps

Pumps are in general classified as Centrifugal Pumps or Positive Displacement Pumps.

1.1.1. Centrifugal Pumps

The centrifugal pump produces a head and a flow by increasing the velocity of the liquid through the machine with the help of a rotating vane impeller.

The centrifugal pump can be classified as:
- End suction pump
- In-line pump
- Double suction pump
- Vertical multistage pump
- Horizontal multistage pump
- Submersible pumps
- Self-priming pumps
- Axial-flow pumps
- Regenerative pumps

Positive Displacement Pumps

The positive displacement pump operates by alternating of filling a cavity and then displacing a given volume of liquid. The positive displacement pump delivers a constant volume of liquid against varying discharge pressure or head.

The positive displacement pump can be classified as:
- Reciprocating pumps
- Power pumps
- Steam pumps
- Rotary pumps

Selecting between Centrifugal or Positive Displacement Pumps

Selecting between a Centrifugal Pump or a Positive Displacement Pump is not always straight forward.

Flow Rate and Pressure Head

The two types of pumps behave very differently regarding pressure head and flow rate:
- The Centrifugal Pump has varying flow depending on the system pressure or head
- The Positive Displacement Pump has more or less a constant flow regardless of the system pressure or head. Positive Displacement pumps generally give more pressure than Centrifugal Pumps.

Capacity and Viscosity

Another major difference between the pump types is the effect of viscosity on the capacity:
- In the Centrifugal Pump the flow is reduced when the viscosity is increased
- In the Positive Displacement Pump the flow is increased when viscosity is increased

Liquids with high viscosity fill the clearances of a Positive Displacement Pump causing a higher volumetric efficiency and a Positive Displacement Pump is better suited for high viscosity applications. A Centrifugal Pump becomes very inefficient at even modest viscosity.

Mechanical Efficiency

The pumps behave differently considering mechanical efficiency as well.
1. Changing the system pressure or head has little or no effect on the flow rate in the Positive Displacement Pump
2. Changing the system pressure or head has a dramatic effect on the flow rate in the Centrifugal Pump

Net Positive Suction Head - NPSH

Another consideration is the Net Positive Suction Head NPSH.
1. In a Centrifugal Pump, NPSH varies as a function of flow determined by pressure
2. In a Positive Displacement Pump, NPSH varies as a function of flow determined by speed. Reducing the speed of the Positive Displacement Pump, reduces the NPSH

Affinity Laws

The turbo machine affinity laws are common used for pumps and fans to calculate capacity, head and power consumption when changing speed or wheels diameters

The Affinity Laws of a centrifugal pump or fan express the effect on capacity, head and power consumption of a pump or fan due to
• Different speed - revolutions per minute (rpm)
• Geometrically similarity - impeller diameter

Changing the Pump or Fan Speed (rpm)

Changing the speed (rpm, rev/min) of the pump or fan:

Capacity

$$Q_1/Q_2 = N_1/N_2 \tag{1}$$

Head

$$H_1/H_2 = \left(N_1/N_2\right)^2 \tag{2}$$

Power

$$P_1/P_2 = \left(N_1/N_2\right)^3 \tag{3}$$

Where

Q = capacity
H = head
P = power
N = speed

If the speed increase with 10%
- The volume increase 10%
- The head increase 21%
- The power increase 33 %

<u>Note!</u> This means that if we want to increase the volume flow in an existing system with 10% we have to increase the power with 33%. In general the motor and the power supply must be changed.

Changing the Impeller Diameter

When effectively changing the impeller diameter from D_1 into D_2 then:

Capacity (Discharge) in l/s becomes

$$Q_1/Q_2 = \left(D_1/D_2\right) \tag{4}$$

Head

$$H_1/H_2 = \left(D_1/D_2\right)^2 \tag{5}$$

Power

$$P_1/P_2 = \left(D_1/D_2\right)^3 \tag{6}$$

Where

D = impeller diameter

Example 1- Pump Affinity Laws

When Pump Speed Changed (Impeller Size Constant):

Assuming the initial flow rate as 100 GPM, the initial head is 100 ft, the initial power is 5 bhp, and the initial speed is 1750 rpm and the final speed 3500 rpm.

The final flow capacity can be calculated by transforming 1.1:
$$Q_2 = Q_1 N_2 / N_1 = \left(100 \text{ GPM}\right)\left(3500 \text{ rpm}\right) / \left(1750 \text{ rpm}\right) = 200 \text{ GPM}$$

The final head can be calculated by transforming 1.2:

$$H_2 = H_1 (N_2 / N_1)^2 = (100 \text{ ft})((3500 \text{ rpm}) / (1750 \text{ rpm}))^2 = 400 \text{ ft}$$

The final power consumption can be calculated by transforming 1.3:

$$P_2 = P_1 (N_2 / N_1)^3 = (5 \text{ bhp})((3500 \text{ rpm}) / (1750 \text{ rpm}))^3 = 40 \text{ bph}$$

When Pump Impeller Diameter Changed (Pump Speed Constant)

The impeller diameter is changed from 8" (inch) to 6" (inch).

The final flow capacity can be calculated by transforming 1.4:
$$Q_2 = Q_1 D_2 / D_1 = (100 \text{ GPM})(6") / (8") = 75 \text{ GPM}$$

The final head can be calculated by transforming 1.5:
$$H_2 = H_1 (D_2 / D_1)^2 = (100 \text{ ft})((6") / (8"))^2 = 56.25 \text{ ft}$$

The final power consumption can be calculated by transforming 1.6:
$$P_2 = P_1 (D_2 / D_1)^3 = (5 \text{ bhp})((6") / (8"))^3 = 2.1 \text{ bph}$$

What is Cavitation?

Cavitation occurs in a fluid flow system when the local static pressure is below the vapor pressure. Cavitation - a common problem that can be observed in pumps and control valves - causing serious wear and tear and damage. Under the wrong conditions, cavitation will reduce the components life time dramatically. According to the Bernoulli Equation this may happen when the fluid accelerates in a control valve or around a pump impeller. The vaporization itself does not cause the damage; the damage occurs when the vapor almost immediately collapses after evaporation when the velocity is decreased and pressure increased. Figure 1 and 2 show examples of suction side and discharge side cavitation in pumps.
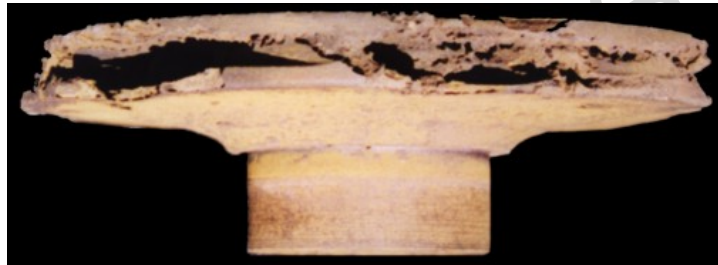
Figure 1. Suction side Cavitation

Figure 2. Discharge side Cavitation

How to Avoid Cavitation

Cavitation can in general be avoided by increasing the distance between the actual local static pressure in the fluid - and the vapor pressure of the fluid at the actual temperature. This can be carried out by:

I) Reengineering of Components Initiating High Speed Velocity and Low Static Pressure

Cavitation and damage can be avoided by using special components designed for the actual rough conditions:

- Conditions as huge pressure drops can - with limitations - be handled by Multi Stage Control Valves
- Difficult pumping conditions - with fluid temperatures close to the vaporization temperature - can be handled with a special pump - working after another principle than the centrifugal pump.

II) Increasing the Total or Local Pressure in the System

By increasing the total or local pressure in the system, the distance between the static pressure and the vaporization pressure is increased and vaporization and cavitation may be avoided. The ratio between static pressure and the vaporization pressure, an indication of the possibility of vaporization, is often expressed by the Cavitation

Number. Unfortunately it may not always be possible to increase the total static pressure due to system classifications or other limitations.

Local static pressure in the component may then be increased by lowering the component in the system. Control valves and pumps should in general be positioned in the lowest part of the systems to maximize the static head. This is common for boiler feeding pumps receiving hot condensate (water close to 100 ^{o}C) from a condensate receiver.

III) Reducing the Temperature of the Fluid

The vaporization pressure is highly dependable of the fluid temperature. Water, our most common fluid, is an example (see Table 1)

Temperature (^{o}C)	Vapor Pressure (kN/m^{2})
0	0.6
10	1.2
20	2.3
30	4.3
40	7.7
50	12.5
60	20
70	32.1
80	47.5
90	70
100	101.33

Table1. Vapor pressure of water as a function of temperature

As we can see - the possibility of evaporation and cavitation increases dramatically with the water temperature. Cavitation can be avoided by locating the components in the coldest part of the system. By example it is common to locate the pumps in heating systems at the "cold" return lines. The same is the situation for control valves. Where it is possible they should be located on the cold side of heat exchangers.

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Bibliography

Balje, O.E. (1981). Turbomachines. A guide to design, selection and theory. John Wiley and Sons, New York. [A comprehensive reference on turbo machinery].

Sabersky, R.H., Acosta, A.J. and Hauptmann, E.G. (1989). Fluid flow (3rd edition), Chapters 12 and 13. Macmillan Publ. Co. [A very well written reference on fluid flow basics and problems].

Stodola, A. (1927). Steam and gas turbines. Volumes I and II . McGraw-Hill, New York [A legendary text for all times to be read for detailed information on fluid flow and turbo machines].

Khalil, E.E., (1990) Power Plant Design, Gordon and Breach, [A very well written reference on power plant applications where pumps and compressors details are discussed].

Richards J. (1894). Centrifugal pumps: an essay on their construction and operation, and some account of the origin and development in this and other countries. The Industrial Publishing Company. p. 40–41. http://books.google.com/books?id=013VAAAAMAAJ&pg=PA41. [A classical historical reference on centrifugal pumps worthy of reading].

Biographical Sketch

Essam E.Khalil received B.S.(Hon.) from Cairo University in Mechanical Engineering in 1971 and M.S. degree in 1973, in Mechanical Engineering from Cairo University; He then received his DIC in 1976 and Ph.D. in 1977, from Imperial College of Science and Technology, London University, U.K. He has been a faculty member at Cairo University and is currently Professor of Mechanical Engineering, Cairo University. Over 40 years of experience in design and simulation of energy related systems for terrestrial and aerospace applications. Published over 465 papers in journals and conference proceedings on combustion, energy, fluid flow, CFD and indoor air quality control. Developed and delivered advanced courses in Mechanical power engineering, heat transfer applications, gas turbine combustors, and terrestrial energy. He is Fellow of AIAA, Fellow ASME and Fellow ASHRAE. In 2009 he received the ASME George Westinghouse Gold Medal and in 2010 he was awarded the AIAA energy systems Award. He was honored in 2011 as the recipient of the ASHRAE Distinguished service award and Sustained Services award from AIAA. He has been elected as convener of ISO TC205 WG2 and his interests include power systems, Computational Fluid Dynamics and sustainable development.