

ANALYSIS PERFORMANCE CHARACTERISTICS OF CENTRIFUGAL PUMPS

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Centrifugal pump characteristics at different pump operating speeds are important for the successful operation of pumps in various areas of technology. In paper the pump characteristics are discussed to calculate the characteristic functional curves $H = f(Q)$, $P_w = f(Q)$, $P_f = f(Q)$, $\eta = f(Q)$ of given type of pump, specific work and flow of fluid at different frequencies of rotation, to demonstrate the connection of centrifugal pumps in serial and parallel, to product map of characteristic curves with an ISO efficiency curve and to verify the similarity rules for pumps. Using calculation focused analyses the behavior of individual characteristic curves of the pump, to find how the ISO efficiency curve can be applied for the determination of the most efficient frequency of a pump rotation. Based on calculation will be decided when to connect the pumps in series or in parallel.

KEYWORDS

map of characteristics curves, efficiency, power, NSPH

1 INTRODUCTION

Pumps are relatively simple machines with no special designing and are readily available in most developing countries. Besides, their installation, commissioning and maintenance are easy and cheap [Haidary 2013]. The transfer of energy from the mechanical rotation of the impeller to the motion and pressure of the fluid is usually described in terms of centrifugal force, especially in older sources written before the modern concept of centrifugal force as a fictitious force in a rotating reference frame was well articulated. The concept of centrifugal force is not actually required to describe the action of the centrifugal pump. In the modern centrifugal pump, most of the energy conversion is due to the outward force that curved impeller blades impart on the fluid. Invariably, some of the energy also pushes the fluid into a circular motion, and this circular motion can also convey some energy and increase the pressure at the outlet [TLC 2016]. Since 1930, the main challenge in pump usage was the selection of a proper pump for a small hydro-site [17]. As the operating performance of a centrifugal pump understood the connection between the flow rates supplied by the pump with the delivery head prescribed by the plant. During the evaluation of the operating performance it must be considered that the pump is not an isolated machine, but it is always integrated into a complete plant and all parts of the plant have to be considered. The cooperation of the pump with the consumer can be described by the characteristics of pump

and plant. Characteristics of turbo machineries represent the functional connection between different machine and operating parameters. In centrifugal pumps the parameters Q , H , η , P , NPSH (Net Position-Suction Head) have the greatest importance. For the representation of the characteristics frequently values such as delivery head H , power P , efficiency η , or the NPSH – value are laid over the volume flow rate. Geometry sizes such as impeller diameters D , position of the peeler or the machine number of revolutions n are often used as parameters. In order a pump to run correctly, without wasting energy or sustaining internal damage, its characteristics must be suitable for the intended service. Therefore, the correct measurement of characteristics for selection of the pump is the first step guarantee efficient and appropriate performance, reliable operation and a reasonably long life [5]. Some parameters that influence the pump characteristic curves and the pressure evolution during the startup period were studied.

2 MATERIALS AND METHODS

A centrifugal pump is a rotor dynamic pump that uses a rotating impeller to increase the pressure of a fluid. This pump is constituted by rotor inside a carcass. The fluid enters the pump impeller along or near the rotating axis and is accelerated by the impeller flowing radially outward into a diffuser or volute chamber (casing), and subsequently into the downstream piping system. It works by converting the kinetic energy into potential energy measurable as the static fluid pressure at the outlet of pump. In Fig. 1 a typical behavior of the characteristic curves of centrifugal pumps is shown [Haidary 2013].

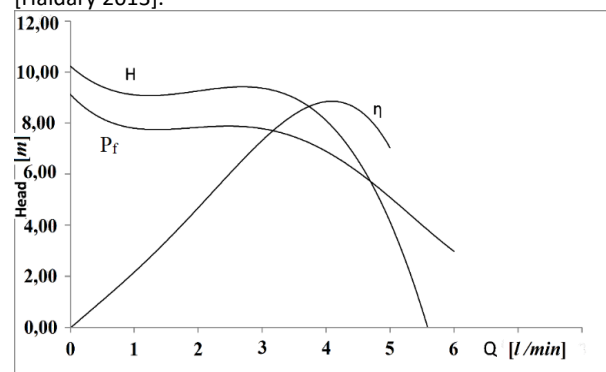


Figure 1. Typical characteristic curves of centrifugal pump [1] H – is net head P_f – is mechanical power curve and η – is efficiency curve

Many centrifugal pump Fig. 2, form configurations have been developed for different applications, from single to multistage, axial or radial split, horizontal or vertical rotor position. By far the most common pump type is the horizontal one, used in a wide range of pressures up to 400 [bar] and temperatures up to 450 [°C], while vertical ones are used for a more restricted range of applications, mainly in low NPSHA conditions and very low temperatures [TLC 2016]. The energy is imparted to the liquid by means of centrifugal force and so the pressure generated is proportional to the square of the speed [Benra 2016].

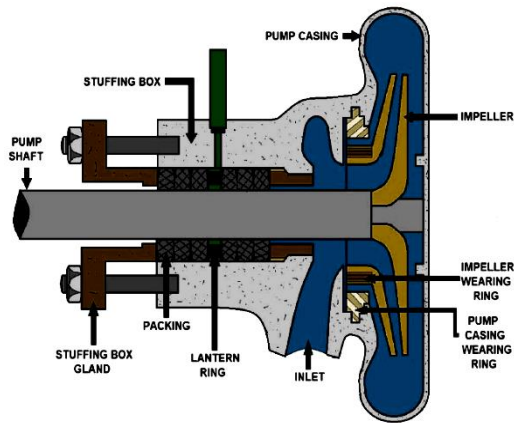


Figure 2. Centrifugal pump components [2]

Centrifugal pumps are used for large discharge through smaller heads. To achieve larger specific work, more pumps are connected in series; however, if the aim is larger flow, they can be connected in parallel [Haidary 2013]. Centrifugal pumps are divided into three general categories [TLC 2016]: axial pump, gear pump and peripheral flow pump.

3 AXIAL FLOW PUMP

In these pumps, fluid enters axially into the propeller through a ring of pipes. When passing through the propeller (rotor), the blades give a whirl movement in the fluid, which is eliminated at the outlet before entering the unloading pipe. The axial flow pump Fig. 3, adapts better under conditions, where great unloading and sewers pumping are some of the typical application. In Fig. 4 a typical behavior of the characteristic curves of axial pumps is shown [Haidary 2013].

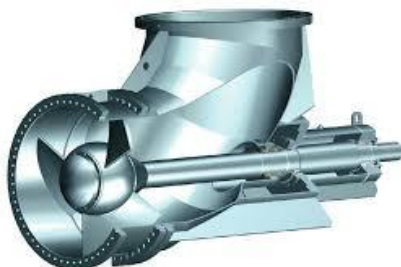


Figure 3. Axial pump [14]

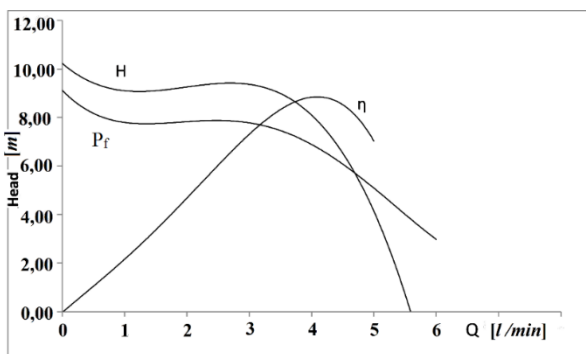


Figure 4. Characteristic curves of an axial curve [1]

4 GEAR PUMP

The gear pump of positive displacement has a carcass of a fused piece and two brass runners in the shape of straight gears rotating with some interstices next to each other Fig. 5. In such

a way the fluid enters the suction area is caught in the spaces between the adjacent teeth and turns until it is shot out through the outlet area [Haidary 2013].

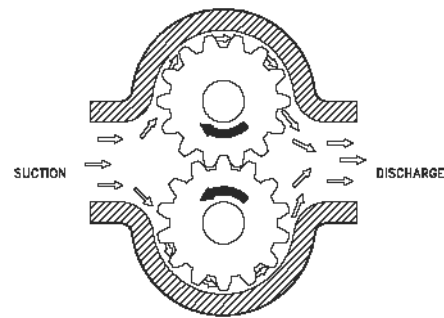


Figure 5. Gear pump [1]

The main disadvantage of gear pump is that very close interstices between the end of the rotor and the carcass are needed. Any waste or corrosion in this area caused by the materials pumped reduces the efficiency of the pump. In Fig. 6 a typical behavior of the characteristic curves of gear pump is shown [Haidary 2013].

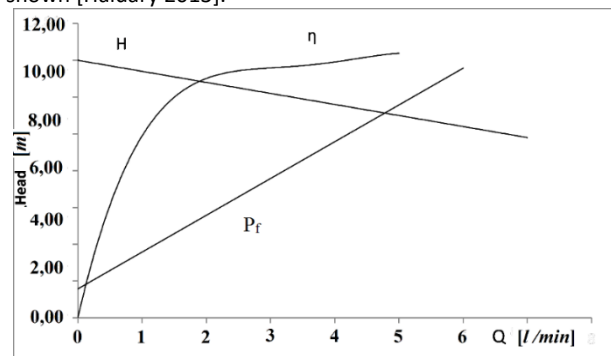


Figure 6. Characteristic curves of gear pump [1]

5 PERIPHERAL PUMP

The peripheral pump, also known as regenerator or turbine pump, consists of a runner of straight blades in an annular carcass and an axis of activation on two lubricated ball bearing. The front cover and the sealed carcass are fused.

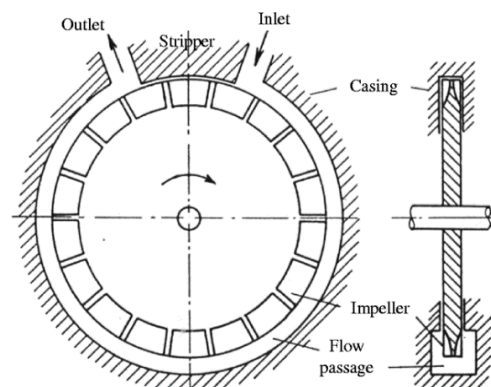


Figure 7. Peripheral pump [1]

The device, which belongs to the rotative mechanical type, is self-lubricated. The aspiration is connected directly to the collection tank and its drive is connected to the pipe selection and the measuring system. The action of this pump is described in Fig. 7. In Fig. 8 shows a typical behavior of the characteristic curves of peripheral pump [Haidary 2013].

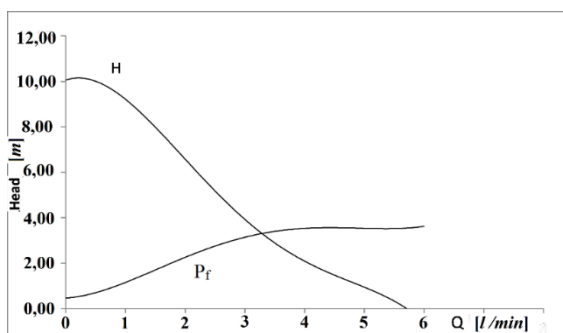


Figure 8. Peripheral pump characteristic's curves [1]

6 THEORETICAL INTRODUCTION

The term pump is used for a machine that adds energy to the fluid; the machine that extracts energy is called turbine. Selection of the correct pump for a certain application is essential for a satisfactory and efficient operation. Basically, the pump increases the fluid load between its inlet and outlet. Considering the stationary flow and omitting the viscous, for monomeric height, H (net head), representing the specific work of a pump from the Bernoulli equation result:

$$H = \left(\frac{P}{\rho g} + \frac{w^2}{2g} + z \right)_2 - \left(\frac{P}{\rho g} + \frac{w^2}{2g} + z \right)_1$$

Where indexes 1 and 2 represent the inlet of the pump and the outlet of the pump, respectively. P is pressure, w is velocity, ρ is density and g is gravity. Under normal condition, the inlet and outlet section of any pump are equal and considering that the fluid entering the pump is similar to the one that comes out, leads to the conclusion that the inlet and outlet speed of the fluid are the same. The variation of the potential energy between the inlet and outlet of the pump can be neglect ($z_1 = z_2$); so the net head H , is essentially proportional to the static pressure increment [Haidary 2013].

7 NET HEAD

A Q-H curve or pump curve shows the head H as a function of the flow Q . The flow Q is the rate of fluid going through the pump. The pump started and runs with constant speed. Q equals 0 and H reaches its highest value when the valve is completely closed. The valve is opened gradually and as Q increases H decreases. H is the height of the fluid column in the open pipe after the pump [Blistan 2012]. The plant characteristic indicates the delivery head, which is necessary for delivering the fluid against the existing resistances in the piping for any flow rates [Benra 2016].

$$H = \frac{P_2 - P_1}{\rho g}$$

Where P_1 and P_2 shows the pressure drop of the pump system. Or specific work ϵ_w done by the pump is:

$$\epsilon_w = Hg = \frac{P_2 - P_1}{\rho}$$

8 PUMPING POWER

The power imparted into a fluid will increase the energy of the fluid per unit volume. Thus the power relationship is between the conversion of the mechanical energy of the pump mechanism and the fluid elements within the pump [Larralde 2010]. The power curves show the energy transfer rate as a

function of flow. Distinction is made between three kinds of power [Chalghoum 2016]:

- Supplied power from external electricity source to the motor and controller.
- Shaft power transferred from the motor to the shaft.
- Hydraulic power transferred from the impeller to the fluid

The power given to the fluid P_w is denominated hydraulic power or output power and is calculated as:

$$P_w = \rho g Q H$$

The mechanical power P_f given to the pump by the activator motor is denominated control power or input power and can be calculated as:

$$P_f = \omega T = \frac{2\pi}{60} n T$$

$$P_f = \frac{2\pi}{60} n T$$

The ω is the angular axis speed in $[\text{rad} \cdot \text{sec}^{-1}]$, n frequency of rotation in (r.p.m., revolutions per minute) and T the torque in the axis in $[\text{Nm}]$, to obtain the power in $[\text{W}]$ [Haidary 2013].

9 PUMP EFFICIENCY

Pump efficiency η is defined as the ratio of the power imparted on the fluid by the pump in relation to the power supplied to drive the pump. Its value is not fixed for a given pump; efficiency is a function of the discharge and also operating head. For centrifugal pumps, the efficiency tends to increase with flow rate up to a point midway through the operating range and then declines as flow rates rise further [Blistan 2011]. Pump efficiency decreases with time because of wear. A well-designed pump usually comes with a diametric clearance of $0,2 \div 0,4 \%$. However, as long as it remains below $0,6 \div 0,8 \%$ its effect on efficiency remains negligible. When the clearance starts to increase beyond these values efficiency starts to drop drastically. For equal operating conditions, the rate of wear depends primarily on the design and material of the wear ring. Generally, for noncorrosive liquids, the resistance to wear increases with the hardness of the sealing surface material [Sahoo 2009]. The maximum benefit of installing an energy-efficient machine will only be carried out if performance levels can be maintained for long periods of time between overhauls. Performance degradation is inevitable; however, a combination of good hydraulic and mechanical design can have a positive impact in this area and prolong optimum efficiency for much longer periods of time [Graham 2007]. The pumping efficiency, η is defined as the ratio between the hydraulic power and the mechanical power, that is, the quotient between the power given to the fluid and the power given to the pump by the activator motor [Haidary 2013]:

$$\eta = \frac{P_w}{P_f} = \frac{\rho g Q H}{\frac{2\pi}{60} n T} \quad (2)$$

If there are no losses, $P_w = P_f$, and the efficiency is 100 %, but it is not the case in practice. (3)

If two pumps of the same geometric family (pump 1 and 2) are operating in homologous points, the relation between their flows, monomeric heights and powers are as follows:

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1 \left(\frac{D_2}{D_1} \right)^3};$$

$$\frac{H_2}{H_1} = \left(\frac{n_2}{n_1}\right)^2 \left(\frac{D_2}{D_1}\right)^2$$

$$\frac{Q_2}{Q_1} = \frac{\rho_2 n_2}{\rho_1 n_1} \left(\frac{D_2}{D_1}\right)^3$$

These are the similarity rules that can be used to estimate the effect of fluid change, speed or size of any dynamic turbo machine (pump or turbine) from a geometrically similar family. D is the diameter of the impeller, n is rotational speed [Haidary 2013].

10 NET POSITIVE SUCTION HEAD (NPSH)

The NPSH [m] parameter shows the difference between the actual pressure of a liquid in a pipeline and the liquid's vapor pressure at a given temperature. NPSH is an important parameter to be taken into account when designing a circuit: whenever the liquid pressure drops below the vapor pressure, liquid boiling occurs, and the final effect will be cavitation. Vapor bubbles may reduce or stop the liquid flow as well as damage the system [Wang 2016]. Measuring centrifugal pump cavitation performance curve and capturing cavitation flow evolution in the impeller entrance are the major research parts [Febo 2016]. Centrifugal pumps must have a certain amount of absolute fluid energy at the pump inlet/suction to function correctly. We designate this energy, expressed in feet or meters, using the metric system, as the net positive suction head required or NPSHR [Febob 2016]. The net head of suction, NPSH (Net Position-Suction Head), is the load available at the inlet of the pump to avoid the liquid's cavitation or evaporation. NPSH has two parts: NPSH required (NPSHR) and NPSH available (NPSHA). (NPSHR) is a function of the pump and it is defined as:

$$NPSH_R = \left(\frac{P_1}{\rho g} + \frac{w_1^2}{2g} - \frac{P^*}{\rho g}\right)$$

Where P_1 and w_1 are the pressure and the [Sentyakov2013] at the inlet of the pump and P^* is the transported liquid's steam pressure. Velocity is calculated as:

$$w = \frac{Q}{A}$$

Where A, [m²] is the circular surface area of the impeller which is calculated as:

$$A = \frac{\pi}{4} D^2$$

Where D, [m] is impeller diameter.

NPSHA is a function of the system and must be calculated. If the inlet of the pump is in height z_1 above a tank whose free surface is under pressure P_a , we can use the Bernoulli equation to the NPSHA as:

$$NPSH_a = \left(\frac{P_a}{\rho g} - z_1 - h_{f1} - \frac{P^*}{\rho g}\right)$$

Where P_a is atmospheric pressure, h_{f1} is the load loss between the tank and the inlet of pump [Haidary 2013].

11 PUMP CURVES

A pump curve describes the operation of a pump for a range of flows at a defined speed. Many design elements affect the shape of the pump curve, and most of these cannot be changed by the user. As a result, centrifugal pumps are usually selected

from the manufacturer's available designs to match the system requirements. An engineered or assembled-to-order pump can be specified, and the manufacturer can often provide a pump performance characteristic well suited to the specific application depending on the type of pump. This condition will obviously force an arrangement to be found to fulfil that basic requirement with the highest possible efficiency. Several options could be considered, such as choosing a pump whose Q-H characteristic curve is flat enough to satisfy the conditions imposed by the process; using several pumps in parallel or serial. It may happen that the option adopted entails the pump running in a capacity range wider than the recommended (80 ÷ 110)% of the flow at the best efficiency point (BEP), losing not only efficiency but also reliability, as shown in Fig. 9, Fig. 10. Additionally, the minimum safe flow recommended by manufacturers for the pump should be clearly stated in order to prepare the system to ensure the safe, continuous operation of the pump when the process flow demand is lower than that value [Warman 2014].

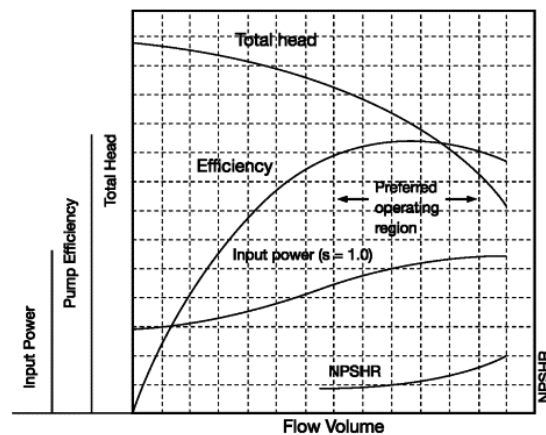


Figure 9. Curve of centrifugal pumps [5]

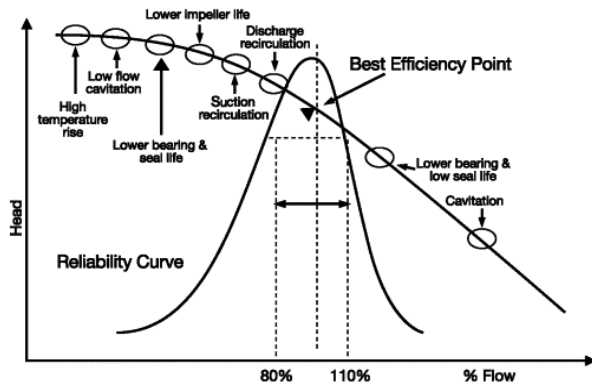


Figure 10. Adverse effects of operating away from the BEP [5]

A common problem of electrically driven pumps occurs when the pump is to be installed in a country using a different electric voltage and/or frequency from that used in the country of the manufacturer.

Lastly, the actual efficiency occurring at the operating point of the pump and the maximum efficiency the pump can achieve should be compared; unreasonably large differences should not be allowed. This comparison is made by intersecting the system characteristic curve with the pump's head curve in order to establish what the operating point of the pump will be and compare it with the BEP [Edibo 2014].

For production a map of characteristic curves $H = f(Q)$ and the Iso-efficiency curve „Figure 11“, the following procedure is used:

- Record of H with Q dependence for the whole range of revolution.
- Record of the efficiency with the flow dependence.
- For determining the flow in the last curve we selected the efficiency and the frequency of rotation from data.
- We determined the net head from the graph of $H = f(Q)$
- For the same efficiency determined the flow for another value of revolution that indicate it in the H with Q curve.
- Then we joined points in the H with Q graph for the same efficiency.

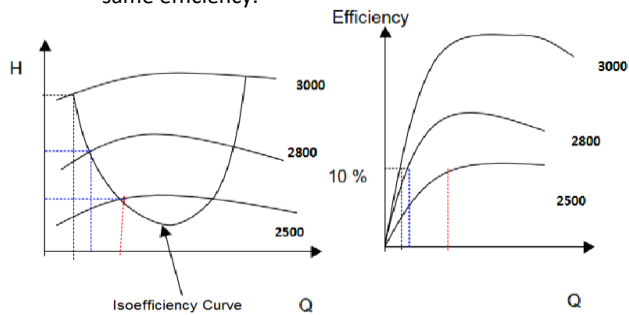


Figure 11. A map of characteristic curves [1]

12 EXPERIMENTAL DATA MEASUREMENT

Operational versatility is an outstanding feature of centrifugal pumps that has contributed to their extended use in a wide range of applications. But, although a centrifugal pump can serve many different operating conditions, it will not provide suitable and satisfactory performance in all of them. One of the main difficulties for the analysis of centrifugal pump operative performances comes from how the data is presented [Satterfield 2013]. For the determination of the characteristics of a centrifugal pump the measurement of the volume flow rate is of elementary importance. Experiments recorded values with different frequency are set in Tab. 1. We measured these data in Kabul Polytechnic University laboratory uses plant has four pumps (centrifugal, gear, axial and peripheral) each pump is independent and autonomous. In each pump there are two pressure taps, one for the discharge and another one for the suction. These taps are connected to their corresponding (pressure sensors (SP1-Sp8) which are connected to the electronic interface and PC. The vales and pipes disposition allows a quick selection of any of the pumps to be tested. The pumps discharge is independent and each one has a flow regulation membrane valve.

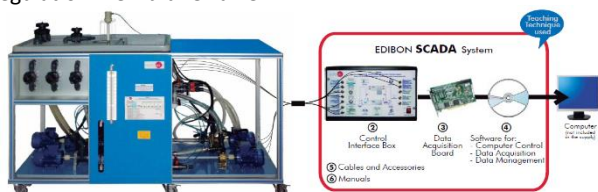


Figure 12. Scheme of multi pump testing bench in Kabul polytechnic university [15]

Electronic Interface and PC with a control and measuring software for direct reading of frequency of rotation and torque [Nm] and of the variation of the frequency of rotation by a frequency converter used and we write experimental values from PC in Tab. 1. The test procedure for measuring data of the given type pump (centrifugal pump) observed that the flow reached by the pump, both, suction, P01, and discharge, P02, pressure and also the torque T, [Nm]. Repeated steps 5÷7 until

covering the whole flow rang or until the discharge pressure of 0, 3 bars reached. It is shown in Tab. 1, in 2000 revolutions per minute.

Table 1. Frequency of rotations of pumps to be used

3000 (rpm)	Number	1	2	3	4	5
	Flow rate [$L \cdot min^{-1}$]	19,84	18,7	16,36	7,16	6,23
	Suction pressure [Bar]	-0,004	-0,03	-0,03	-0,01	-0,01
	Discharge pressure [Bar]	0,83	0,86	0,89	0,94	0,94
	Torque [Nm]	1,05	0,88	0,86	0,84	0,82
2800 (rpm)	Number	1	2	3	4	5
	Flow rate [$L \cdot min^{-1}$]	16,96	16,3	12,35	7,08	4,49
	Suction pressure [Bar]	-0,03	-0,03	-0,02	-0,01	-0,01
	Discharge pressure [Bar]	0,76	0,77	0,79	0,83	0,85
	Torque [Nm]	0,86	0,85	0,84	0,82	0,8
2500 (rpm)	Number	1	2	3	4	5
	Flow rate [$L \cdot min^{-1}$]	15,3	14,89	14,58	13,7	11,81
	Suction pressure [Bar]	-0,03	-0,03	-0,03	-0,02	-0,02
	Discharge pressure [Bar]	0,6	0,61	0,61	0,62	0,63
	Torque [Nm]	0,63	0,69	0,63	0,67	0,65
2000 (rpm)	Number	1	2	3	4	5
	Flow rate [$L \cdot min^{-1}$]	12,02	11,68	11,42	10,65	8,59
	Suction pressure [Bar]	-0,02	-0,02	-0,02	-0,02	-0,01
	Discharge pressure [Bar]	0,38	0,39	0,38	0,39	0,4
	Torque [Nm]	0,5	0,47	0,53	0,47	0,5

To demonstrate centrifugal pumps connection in series and in parallel we start the data acquisition again, we fully open the valve associated with the centrifugal pump, and we set the frequency of 3000 (rpm). Then we rewrite the volume flow rate of water, Q [$L \cdot min^{-1}$], Hydraulic High, H [m], P_w [W] and P_f [W] see measured values from connection of pumps parallel and serial is shown in Tab. 2.

Table 2. Pumps frequency of rotations

connection	one pump	Two pumps Parallel	Two pumps serial
Flow rate [$L \cdot min^{-1}$]	18,52	33,46	27,61
Head [m]	9,3	9,41	9,08
Power output [W]	-0,03	-0,03	-0,03
Power input [W]	0,86	0,87	0,84

13 PRACTICAL PART

First we found t_{av} , to obtain the average temperature we measured $t_1 = 21$ [°C] at the beginning and $t_2 = 23$ [°C] at the end of the experiments, then $t_{av} = 22$ [°C]. According to average temperature density of water was chosen from physical properties table of water 997,9 [$kg \cdot m^{-3}$]. Each measured value calculated only for the number of rotation 3000 (rpm). Net head measured 8,5 [m], the hydraulic power or output power and is calculated 27,57 [W], the mechanical power or input power obtained 329,8 [W]. The pumping efficiency η that is the quotient between the power given to the fluid and the power given to the pump by the activator motor 8,36 was calculated. NPSHR is a function of the pump and it calculated 0,69 [m]. For calculating of velocity w, [$m \cdot s^{-1}$] we used flow rate equation and for circular surface area A we obtain 0,46 [$m \cdot s^{-1}$] and 0,0007 [m^2] respectively. For each measured value of Q in „Table3“, calculated the net head, output power, input power and pumping efficiency from equations (2-4-6 and 7). Measurement

is continued for (2800–2500, and 2000) until five steps revolutions per minute respectively.

Table 3. Calculated characteristics values

	Number					
		1	2	3	4	5
3000 (rpm)	Flow rate [$l.min^{-1}$]	19,84	18,7	16,36	7,16	6,23
	Head [m]	8,52	9,09	9,40	9,70	9,70
	Power output [W]	27,57	27,73	25,08	11,33	9,86
	Power input [W]	329,86	276,46	311,01	307,87	257,61
	Efficiency	8,36	10,03	8,06	3,68	3,82
2800 (rpm)	Number	1	2	3	4	5
	Flow rate [$l.min^{-1}$]	16,96	16,3	12,35	7,08	4,49
	Head [m]	8,07	8,17	8,27	8,58	8,78
	Power output [W]	22,33	21,73	16,67	9,91	6,43
	Power input [W]	252,16	249,23	246,30	240,43	234,57
	Efficiency	8,85	8,72	6,76	4,12	2,74
2500 (rpm)	Number	1	2	3	4	5
	Flow rate [$l.min^{-1}$]	15,3	14,89	14,58	13,7	11,81
	Head [m]	6,43	6,53	6,53	6,53	6,64
	Power output [W]	16,06	15,88	15,55	14,61	12,79
	Power input [W]	164,93	180,64	164,93	175,40	170,16
	Efficiency	9,74	8,79	9,42	8,33	7,51
2000 (rpm)	Number	1	2	3	4	5
	Flow rate [$l.min^{-1}$]	12,02	11,68	11,42	10,65	8,59
	Head [m]	4,08	4,18	4,08	4,18	4,18
	Power output [W]	8,01	7,98	7,61	7,27	5,86
	Power input [W]	104,71	98,43	111,002	98,43	104,71
	Efficiency	7,65	8,10	6,85	7,39	5,60

14 CURVES CALCULATION

The plot of pump characteristic curves $H = f(Q)$, $P_w = f(Q)$, $P_f = f(Q)$, $\eta = f(Q)$ are as follows, as we mentioned that H is function of Q , the beginning flow rate was equal 0 and the head has its highest value, when the flow rate is increasing the head decreases. But the efficiency has direct proportional relationship with flow rate, when Q is increasing gradually efficiency η is increasing. The same functional relation exists between flow rate, power input and power output. The characteristic curves are various, in different types of pumps and calculated experimental data. A pump curve indicates the head as a function of the flow. The flow Q is the rate of fluid going through the pump. The flow is generally stated in cubic meter per hour [$m^3.h^{-1}$] but used in formulas cubic meter per second [$m^3.s^{-1}$] is used, in our experiment showed [$l.min^{-1}$] see Tab. 1. Performance curves are used by the customer to select pump matching his requirements for a given application. The information about the head H at different flows Q , shows Fig. 13. The requirements for head and flow determine the overall dimensions of the pump [Shahram 2008].

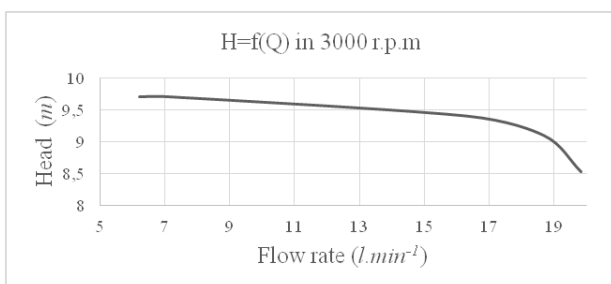


Figure 13. Flow rate and head curve

Power given to the fluid or output power with flow rate in the given speed is shown by increasing the hydraulic power Fig. 14, flow rate increases as well.

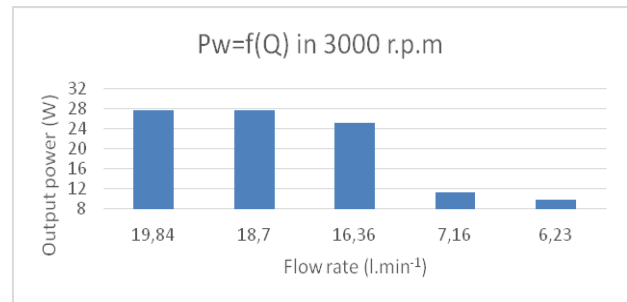


Figure 14. Flow rate and power output curve

The flow where the pump has the highest efficiency Fig. 15, is called the optimum point or the best efficiency point (BSE). The efficiency curve is used for selecting the most efficient pump in the specified operating range.

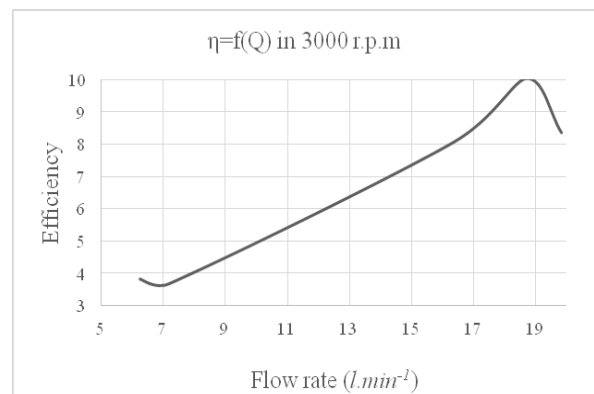


Figure 15. Flow rate with efficiency curve

In addition to head, the power consumption P_f is also calculated. The power consumption or machine power is used for dimensioning the installations which must supply the pump with energy. The power consumption is as the head shown as a function of the flow [Ramos 2002].

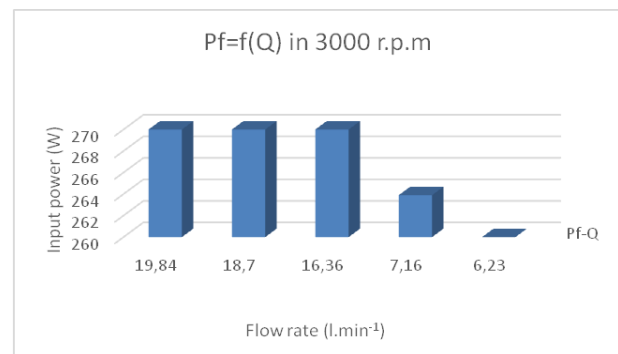


Figure 16. Flow rate with power input curve

From measured data in Tab. 3, we obtain the characteristic curves of a pump, Curve with different rotational speed and flow rate Fig. 17, it shows proportionally increasing flow rate with increasing the speed revolution.

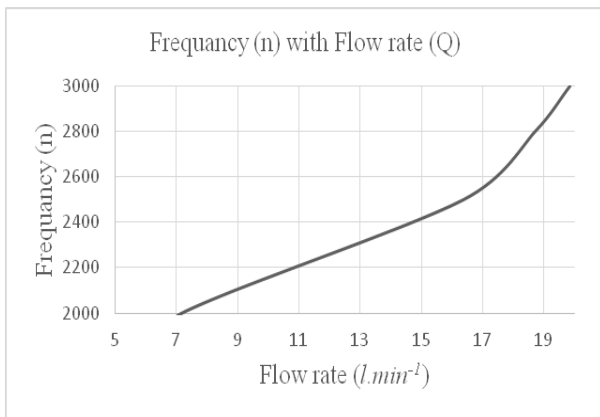


Figure 17. Different rotation speed with flow rate curve

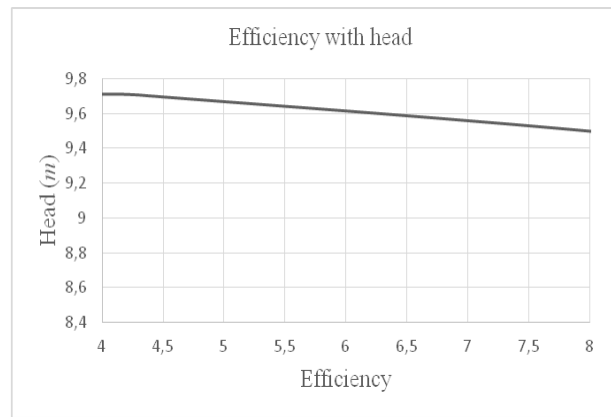


Figure 20. Efficiency with head curve

The head H , and power output P_w , relationship in Fig. 18, indicate little change of head during augment of liquid flow rate.

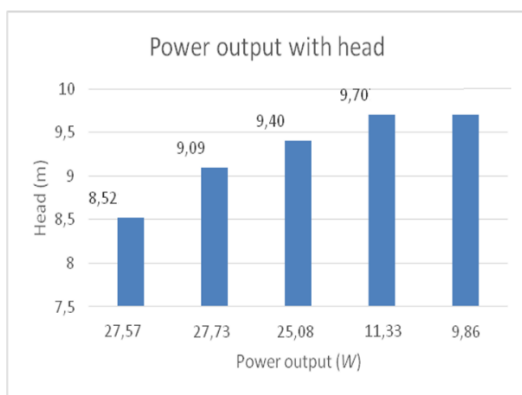


Figure 18. Hydraulic power with head curve

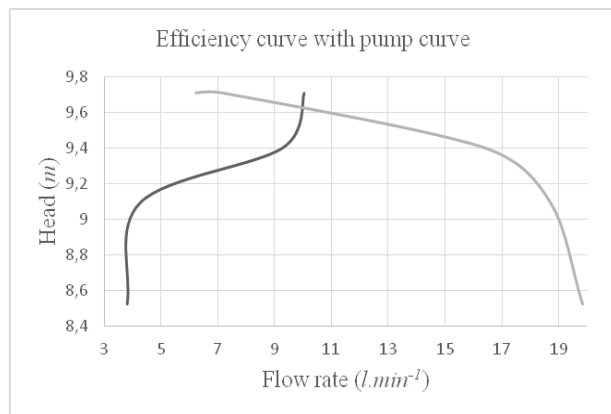


Figure 21. Efficiency intersection with pump curve

Power input P_f , columns in Fig. 19, demonstrates the hydraulic power is transferred from the pump to the fluid. An independent column for the hydraulic power is usually not shown in data but is a part of the calculation of the pump efficiency.

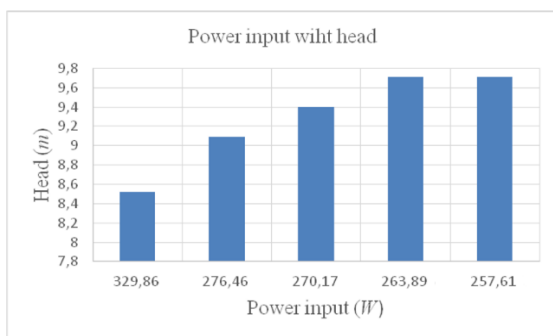


Figure 19: Mechanical power with head curve

System curve with pump curve see Fig. 22, indicates the intersection point of pump curve and system curve of pump. At constant speed 3000 revolutions are plotted. The intersection of pump curve with system curve shows our operating point which head value is 9,4 [m] and Q is approximately 16 [l.min⁻¹], but when are pumps connected parallel Q is 33,46 [l.min⁻¹] see Tab. 2, connection of parallel pumps.

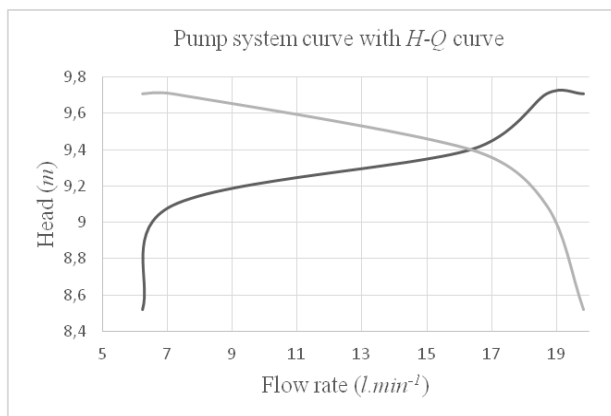


Figure 22. Pump system curve with H-Q curve

Efficiency curve with head Fig. 20, represents small changing of head values and efficiency of pump. By passing the time the efficiency of pump changing is negligible.

Intersection curves of efficiency with pump curve Fig. 21, the efficiency is always below 100% while the supplied power is always larger than the hydraulic power due to losses in controller, motor and pump components. The total efficiency for the entire pump is the product of the individual efficiencies [Ramos 2002].

Relationship between hydraulic power with head and mechanical power with head shows that by increasing the powers the head is decreasing. The pump performance curve also gives information on pump efficiency. The efficiency curves intersect with the head curve are labeled with percentages. The pump's efficiency varies throughout its operating range. Each pump will have its own maximum efficiency point. The best efficiency point (BEP) is the point of highest efficiency of the pump. All points to the right or left of the BEP have lower efficiency [Ramos 2001].

15 RESULTS

The experimental data showed that centrifugal pump can operate different rotational speeds, various heads and flow rates without any mechanical problem. The results are shown in Tab. 1. In this tables, the flow rate, pressure inlet, pressure outlet, and torque are measured different speeds of rotation from (3000-2000 rpm) and the result is calculated in Tab. 3, from equations (2-4-6-7 and 11) for $n=3000$ rpm. The H , P_w , P_f and η are calculated the first step 8,52 [m], 27, 57 [W], 329,86 [W], and 8, 36 respectively. The experiment was continued until five steps, and according to these data we have Fig. 13, Fig. 14, Fig. 15, Fig. 16, for net head, power output, efficiency and power input under consideration of flow rate, respectively. In Fig. 13, the pump curve represented that in the value of H , is 9,6 [m] the flow rate Q is 11 [l.min⁻¹], decreasing of H to 8,5 [m] caused to increase the flow rate to 19,84 [l.min⁻¹]. According to Fig. 14, for flow rate of 13 [l.min⁻¹], consumption hydraulic power is 20 [W], by increasing hydraulic power to 26 [W], the flow rate also increase to 17 [l.min⁻¹]. In Fig. 15, the efficiency and flow rate have direct proportionally, where efficiency in point of 10,03 curve is adopt to 18,7 [l.min⁻¹] flow rate. The curve of input with flow rate in Fig. 16, shows correct results as well. The NPSHR calculated 0,693 [m]. Pumps parallel and serial connection results shows in Tab. 2. All our estimated Figures adopt to previous studies, for example reference [Haidary 2013]. However, the shape of impeller, pump's structure has effect on characteristics of pumps.

16 DISCUSSION

Demonstration of centrifugal pumps connection in parallel and in series showed, parallel connection used to transfer high quantity of liquid to specific height, but in series connection our aim is to transfer liquid to net head required. According to experimental data we obtain for 2800 (rpm) see Tab. 3, that shows the highest value of head started from 8,8 [m] that adopt to minimum flow rate 4, 5 [l.min⁻¹]. By decreasing of head values, the flow rate is increasing, which shows the same proportionally relation with pump revolution in 3000 (rpm). The intersection point in Fig. 22, indicate H is 9,4 [m] and flow rate Q is 16,36 [l.min⁻¹] which represent our best efficiency point (BEP). Therefore, estimating the complete characteristic curves of centrifugal pump based on its BEP which is very remarkable [Razhivina 2015]. Experimental data showed that the dimensionless characteristic curves of pumps based on their BEP were approximately the same. Some researchers reported that pumps with nearly the same specific speeds may have very different head ratios h and flow rate ratios q . For example, if two pumps have the same specific speeds, the more efficient pump operates as a turbine in greater h and q [Bozek 2014]. Based on relationship of hydraulic power curve Fig. 18, and mechanical power curve Fig. 19, with head showed approximately the same shapes in [Haidary 2013].

17 CONCLUSION

The multi pump testing bench, provided by Edibon Spain, allows measuring the operation characteristics of four pumps. In the equipment, do not exceed 1400 (rpm) for this type of pump due to high pressure reached. An important advantage of this type of pump is that they do not need valves either in the aspiration nor in the drive, that is to say, they are able to pump air, gas or liquid without any negative effect, also they do not require a feed. High pressures are possible, although the flow speeds are limited. The main disadvantage of this type of pump is that very close interstices between the end of rotors and the

carcass are needed. Any waste or corrosion in this region caused by the material pumped reduces the efficiency of the pump. Experiments showed that a centrifugal pump can appropriately operate in various rotational speeds, heads and flow rates. A centrifugal pump works in higher head and flow rate at the same rotational speed. Pumps with higher specific speeds have lower ratios of h and q . In future for maximum efficiency on a selected curves $\eta = f(Q)$ of graphs we considered to calculate the annual electricity consumption of the pump with considering of pumps working hours per year. In further research, it is possible to focus on visualization and analysis describes the process using holographic interferometry [Cernecky 2013].

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