ORIGINAL PAPER

EXPERIMENTAL STUDY OF THE PERFORMANCE AND PREDICTION OF CAVITATION PHENOMENON FOR A VARIABLE SPEED CENTRIFUGAL PUMP

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Abstract. *An experimental study of the performance of a variable speed pump and prediction of the cavitation behavior throughout a centrifugal pump has been carried out in this work. The cavitation phenomenon is one of the problems happened in the centrifugal pumps which leads to reduce the pressure supply and the flow rate further creating cracks and cavities at the impeller. In this work, a stander pump with eight forward vanes impeller where the inlet and outlet vanes angles are 20˚, 25˚ and 100 [mm], 200 [mm] inlet and outlet diameters, respectively were used. The results showed that the maximum head and flow rate are at 1400 [rot/min], while the maximum efficiency is at 1100 [rot/min]. The relation between the available and the required net positive suction head was studied to predict the cavitation phenomenon which is the most important aspect to be considered when choosing the pump system. It's found that the* $NPSH_R$ *increases and the* $NPSH_A$ *decrease when the flow rate increases for a certain speed, which leads to creating the cavitation.*

Keywords: *Absence of cavitation, Appearance of the cavitation, Cavitation phenomena in variable speeds.*

1. INTRODUCTION

There are many indications that researchers have used in their laboratory experiments to detect, namely the decrease in the performance of the pump, the sound indication of the emergence of abnormal sounds and vibrations and a visible sign of the emergence of bubbles in different parts of the pump [1].

Concerning the development of cavitation in a centrifugal pump, numerical simulations and experimental investigations have been carried out in a closed hydraulic test rig. The internal flow characteristics and pressure pulsation at pump inlet and outlet have been analyzed during the process of cavitation development. The results of the research reveal the occurrence and development of cavitation in the centrifugal pump which has been confirmed through experiments and numerical simulation. The degree of pump cavitation could be monitored through pump inlet and outlet pressure pulsation. Compared with pump outlet pressure pulsation, pump inlet pressure pulsation is more sensitive to the change of cavitation.

According to experimental research, the typical frequency of pump inlet pressure pulsation could be regarded as around 30 Hz in the severe cavitation conditions. Meanwhile,

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the pump head dropped by 0.77% from noncavitation conditions which could be regarded as a symbol of incipient cavitation [2].

An experimental study has been carried out in order to analyze the cavitation of a centrifugal pump and its effect on transient hydrodynamic performance during transient operation. The transient characteristics of the centrifugal pump were tested under various suction pressure and starting conditions. In transient operation of continuous starting and stopping process, instantaneous rotational speed, head, flow rate and suction pressure of the pump were measured. The effect of cavitation on transient performance of the centrifugal pump during transient operation was analyzed, and then the effects of starting acceleration rate and suction pressure of pump on cavitation were presented. Results showed that the cavitation would be delayed during rapid starting period. However, in the condition of low suction pressure and high rotational speed, pump cavitation is inescapable even if the starting period is less than a second. After the serious transient cavitation occurred, the transient performance of centrifugal pump would decline obviously, and the instantaneous head of pump would fluctuate [3].

Several techniques have been used to detect the cavitation and study its effect on the centrifugal pump parts using the above indications. For example determination of the net positive suction head (NPSH) at a constant speed and flow rate, visualization of the inlet flow at the impeller, paint erosion on impeller blades, static pressure measurement within the flow, vibration measurement of the pump structure and acoustic measurement in the pump. Those measurements show the difference between cavitation and non-cavitation condition about 10 dB in average, but also more than 15 dB of difference is possible [4].

The working conditions in centrifugal pumps also may lead to this phenomenon (cavitation), especially when the suction pressure is close to the vapor pressure of the liquid, low pressure causes the phenomenon of cavitation in the areas close to the eye of the impeller because of the increase in pressure drop value in the suction line [5].

Experimental and theoretical investigations of cavitation flow throughout a centrifugal pump at different flow rates were presented by Ye et al, visualization experiments were carried out to a deeper understanding of cavitation evaluation and provide a reference for numerical simulation [6]. Comparisons between experimental and numerical results were made towards the pump head, cavity length and vapor volume fraction. The other devices used are acoustic, by making use of the natural frequencies that occur in the pump during its operation and comparing them with the abnormal frequencies that occur in the presence of cavitation using a special frequency sensitive mechanism that is connected with the frequency drawing instruments [7].

Mcnulty investigated the initial cavitation in the centrifugal pumps based on the sound measurements for series of pumps with different sizes and run at various velocities. He used pressure sensors and a magnetic tape recorder connected to a system to measure the recorded information. Provided background levels are sufficiently low and can show the inception point of cavitation as well as where the performance is down. [8]

Dong et al. used the pressure and noise measurements to study the effect of modifications to tongue and impeller geometries on the flow structure and resulting noise in a centrifugal pump. It was demonstrated that the primary sources of noise are associated with the interactions of the nonuniform outflux from the impeller (jet/wake phenomenon) with the tongue. Consequently, a significant reduction of noise is achieved by increasing the gap between the tongue and the impeller up to about 20 percent of the impeller radius, a further increase in the gap affects the performance adversely with a minimal impact on the noise level [9].

Barrio et al. presented the behavior of unsteady flow near the tongue region of a single-suction volute-type centrifugal pump, which is available at the laboratory. Commercial

CFD software solved the *Navier-Stoke* equations for three-dimensional unsteady flow. The numerical model was used to evaluate the evolution of the leakage flow between the impeller tongue gap and the flow existing at the impeller through some specific angular intervals, during one single-blade passage [10].

Watanab et al. experimentally investigated the cavitation effects on the corrosion parts of the impeller in the centrifugal pumps by using a laboratory test device. The researchers concluded that the damage of corrosion caused by cavitation is minimal in the normal flow area (there is no inverse flow formed at the input of the impeller), but it suddenly increases to the highest value at the point of the reverse flow, and whenever the reverse flow is less and the damage is less, but the value remains larger than the normal flow areas [11].

Cavitation problem in the centrifugal pumps can cause serious damage and loss of function if it is not diagnosed [12]. The most common area of sensitivity to the appearance of the cavitation in the centrifugal pumps is the low pressure side of the impeller vanes near the front edge and front fascia where the curvature is very high. The axial flow with a high specific speed when it doesn't have a front cover (Open Impeller) will be more sensitive to cavitation on the low pressure side of the edges of the vane as well as the sensitive areas of the pump cover containing low pressure sides and low pressure sides of diffuser blades near the inner edge [13].

The effect of the number of vanes on the pump performance by using laboratory pump device with a six-vane pump with a closed circuit and another pump with six blades, but three of them are incomplete, was investigated by Sh. Yedidiah. It was realized that the cavitation starts at the highest value of the net positive suction head in the large impeller than the other impeller, which means the pressure head starts dropping for the large impeller at the highest value of the net positive suction head [14].

Kergourlay et al. studied the influence of adding splitter blades on the performance of a hydraulic centrifugal pump. They experimentally and concluded that adding splitters has negative and positive effects on the pump behavior. It increases the head rise compared to the original impeller, and it decreases the pressure fluctuations and reorganizes more conveniently the flow at the volute outlet. This option has proven to be ineffective, and the studied flow rates increase the interaction between the volute tongue and the flow [15].

For impeller crack diagnosis of centrifugal pumps, Sun et al. utilized MCSA technology (Motor Current Signature Analysis) in the operation of detection, which could provide a reference for the cavitation prediction, and proved the feasibility and effectiveness of MCSA technology adopted for monitoring centrifugal pump operation situation [16].

In experimental research, Adamkowski et al. found the reason of occurring fractures in pump shafts, namely, the fractures were caused mainly by the resonance between the pump shaft torsional natural vibrations and those following from the pressure fluctuations related to the frequency of the shaft rotational speed and the number of impeller blades [17].

Many effective cavitation models have been proposed with the evolution and innovation of computational fluid dynamics technologies [18]. Other researchers like Yue Hao and Lei Tan studied the cavitation flows of mixed flow pump-turbine experimentally and numerically. The radial force on the principal axis was recorded and compared between pumps with symmetrical and unsymmetrical tip clearance. The clearance has a great influence on the pump cavitation performance, in comparison with the symmetrical tip clearance; the unsymmetrical tip clearance makes the pump cavitation performance worse. Also, the unsymmetrical tip clearance simultaneously influences the magnitude and direction of the radial force, while the symmetrical tip clearance influences the magnitude of the radial force only [19].

In this manuscript, the performance of variable speed radial pump of the forward model impeller was determined experimentally in the laboratory. The performance of the pump was studied in the absence of cavitation using two variables. The first is used in this research which is the discharge rate by manually controlling the valve at the discharge point of the pump. The second is using a variable rotational speed of (900, 1000,1100,1200,1300 and 1400) [rot/mim] using an electric variable speed motor (DC motor) to study the performance at each speed and comparing between them. Study the performance of the pump was conducted by the appearance of the cavitation of the above speeds by controlling the amount of pressure inside the pump manually using a valve at the point of withdrawal while the drain valve remains fully open. The results and relationships that were obtained were compared in practice with the previous theoretical research and available sources, and the compatibility of the current practical study with the previous theoretical research in this fie was examined.

2. MATERIALS AND METHODS

The performance of the centrifugal pump was determined experimentally in the laboratory (figure 2) at the University of Technology, Mechanical Engineering Department, laboratory of fluid mechanics, where the study comprises the three axes, was summarized before.

The main parts of the system are as follows:

– Variable speed electric motor: 1.5 [kW], 1500 [rot/mim], field voltage 180 [V], armature voltage 150 [V].

– Variable speed centrifugal pump (Testing Pump): It is a 200 [mm] outer diameter, 100 [mm] inner diameter, with a twelve straight vanes diffuser controlling the outlet flow through the outlet pipe diameter 75 [mm]. The pump casing is made of cast iron and consists of two parts, the upper one contains a visible plastic plate (Perspex) to allow watching the flow rate and cavitation phenomenon, and the pump shaft is joined to the shaft of the electric motor by means of rigid coupling with rubber shock absorbers. The impeller used (test impeller) is a forward curved vane made of aluminum material that is fixed with four screws, it's geometry and schematic diagram (Fig. 1) are as follows:

–Main Pump: It is a horizontal, single impeller centrifugal pump, power 5.5 [kW], discharge 12 [l/s], with constant rotational speed 2900 [rot/mim], its job is to raise the water pressure when the system works.

Table 1. The characteristics of the pump

Number of vanes	8
Angle of inlet measured from radius of the vane	20°
The exit angle is measured from the radius of the vane	25°
The shape of the vane is a circle with radius	82.5 mm
Internal diameter	100 mm
External diameter	200 mm
Vane Height	15 mm
Vane thickness	3 mm

- Main Tank: The tank is 120 [l] capacities and comprises two valves, the upper one for filling the tank and the lower one for draining it from water; it also contains water level indicator and safety valve for excess pressure.

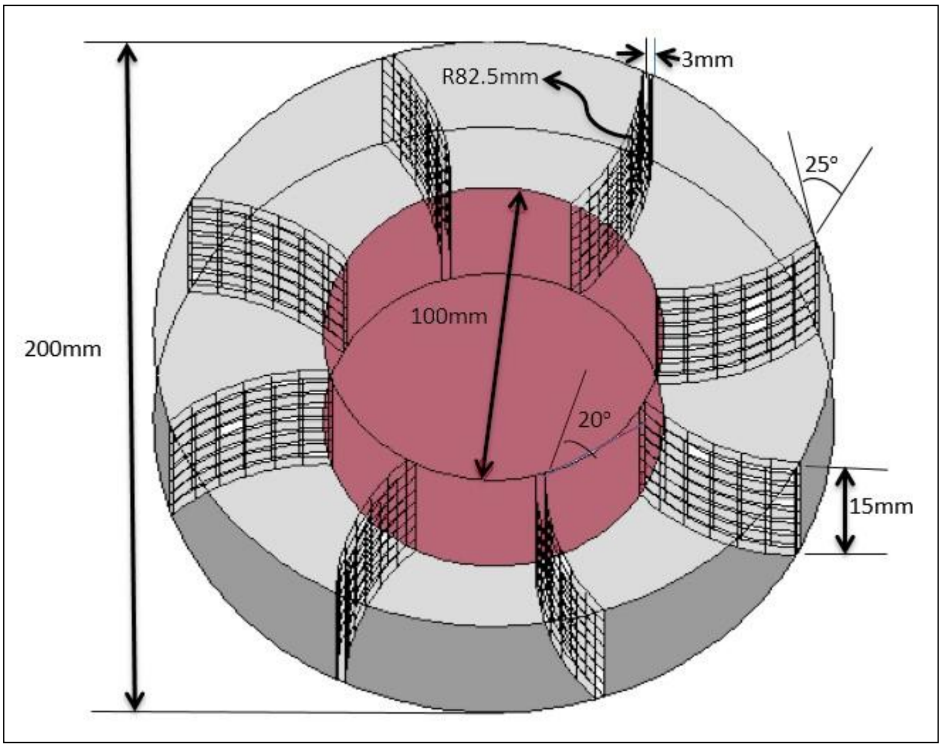

Figure 1. Impeller schematic diagram.

−Piping system: As shown in Fig. 2, they are 75 [mm] diameter and connect all the parts of the system. There are 3 gate valves to control the flow rate of the system; two at the discharge and one for suction.

Figure 2. A centrifugal pump testing system.

Measuring devices used in the laboratory to determine the experienced values: *torque meter*, it is used for measuring the torque of electric variable speed motor - maximum reading 19 [Nm]; *venture meter*, located between the main tank and suction valve to measure the flow rate by determining the pressure difference at the through and the outlet, it is connected to the pipe by a rubber connector, d_1=75 [mm], d_2=37.5 [mm]; *tachometer*, to measure the rotational speed of the variable speed electric motor; *ampere meter*, to measure the electric current at the variable speed electric motor; *pressure transducers*, eight pressure transducers, six of them are used to read the static pressure (as shown in Fig. 3), and two at the venture meter to read the differential pressure.

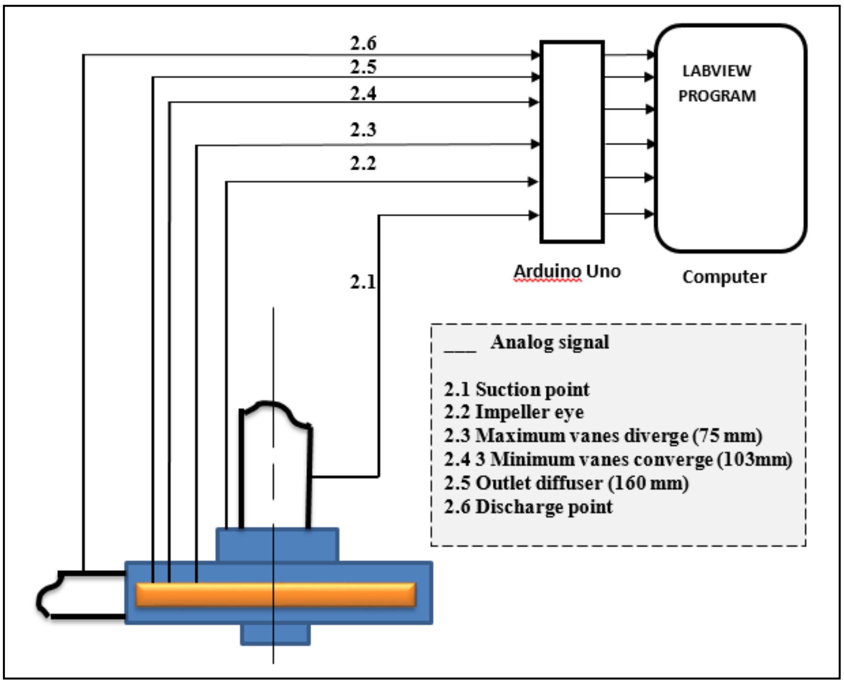

Figure 3. Schematic plan points of pressure measurement.

2.1. COMPUTER SOFTWARE

The microcontroller on the board is programmed by Arduino's programing language and Arduino's integrated development (Ardunio IDE) [20-22]. Ardunio is connected to its sensors and electronic parts only or it is connected to communications with program on the computer, such as processing and Max MSP and lab view, the specifications of Arduino are: Microcontroller; AT mega 328; Working voltage; 5 [V]; Input voltage limits; 6-20 [V] and preferably 7-12 [V]; I/Q outlet; 40 [Ma]; Electrical-voltage in pin 3.3 [V]; 50 [mA]; Memory size; 32 [KB]; Speed; 16 [MHz]; Eight Analogs and two digital signals.

Lab view program reads the variation in eight pressure sensors supported in the system, six of them read a static pressure directly in PC, while the others read the differential across ***Venturi*** meter and transfers this reading to the program, which converts it to flow rate.

2.2. EXPERIMENTAL PROCEDURE

Prepare the system by filling the main tank with water and opening the air pressure valve to discharge, the air and the valve should be closed after ensuring that the air is discharged.

The suction and discharge valves should be open, set the pressure regulator on 0.4 [bar] to fill the device with water and set the pressure gauge of the pump on [10 bar]. Operate the main pump for (3 minutes) to discharge the air trapped in the system. Stop the main centrifugal pump and start the initial measurements by turning on the computer and the sensors, and check the compatibility of sensors with the computer readings by pressing on the static pressure readings sin computer.

Start operating the variable speed motor (note: start from zero and then increase the speed gradually) to approach the required speed to avoid the direct electric loading, when attaining the required speed by Tachometer (used in the system) and its compatibility with manual Tachometer. Suction and delivery valves should be completely open, the pressures and flow rate readings are to be taken, and to get the cavitation phenomenon, close the delivery valves and gradually close the suction valve.

The limiting operating conditions:

- Maximum rotational speed 1400 [rot/min];
- Maximum current value 15 [A];
- Maximum water pressure in the system 2.5[bar];
- Maximum pressure difference through the pump 1.5 [bar];
- Maximum used water temperature 50 [^{o}C];

2.3. DATA REDUCTION

The pump characteristic curves can be defined as ‘the graphical representation of a particular pump’s behavior and performance under different operating conditions’. The operating properties of a pump are established by the geometry and dimensions of the pump’s impeller and casing [23]. Curves relating total head, efficiency, power, and net positive suction head required (NPSHR) to discharge or pump capacity ($\dot{V}$) are utilized to describe the operating properties (characteristics) of a pump [24]. The following equations are used to obtain the head, power, and efficiency from the data recorded from the experimental work.

The system flow rate is defined by the following equation:

$$\dot{V} = A \cdot v \tag{1}$$

where:

$\dot{V}$ = volumetric flow rate (*l/s*)
A = area of the pipe or channel (*m*^{2})
v = velocity of the liquid (*m/s*)

The theoretical power of the pump is defined by:

$$p_{tot} = p_s + p_d \tag{2}$$

or:

$$p_{tot} = p_d - p_s + \frac{1}{2}\rho(v_d^2 - v_s^2) \tag{3}$$

The two terms describe the static and dynamic pressure difference between the discharge (d) and the suction (s) of the pump. Then, the head of the pump could be calculated from:

$$H = \frac{\Delta p_{tot}}{\rho \cdot g} = \frac{p_{tot,\mathrm{out}} - p_{tot,in}}{\rho \cdot g} \tag{4}$$

The theoretical driving power is obtained with the relation:

$$P = \dot{V} \cdot \Delta p \tag{5}$$

$$\Delta p = \rho \cdot g \cdot H \tag{6}$$

Δp - pressure increase [N / m^2];
H - pumping height [m];
ρ - the density of the conveyed fluid [kg / m^3].

Replacing, the efficiency of the pump is:

$$\eta = \frac{p_{out}}{p_{in}} = \frac{\rho \cdot g \cdot \dot{V} \cdot H}{v \cdot I} \tag{7}$$

$$p_{out} = hydraulic\ \ power = \rho \cdot g \cdot \dot{V} \cdot H \tag{8}$$

The net positive suction head (NPSH) value is a concept for judging the suction behavior of a centrifugal pump [25]. NPSH is defined by the difference in total pressure at the inlet of the pump and the vapor pressure. A distinction is made between the available value $NPSH_A$ and the NPSH value required by the pump $NPSH_R$. The $NPSH_R$ is the minimum head required to prevent the pump from cavitation. The available NPSH is defined as:

$$NPSH_A = \frac{p_{tot,s} - p_v}{\rho g} \tag{9}$$

$P_{tot,s}$ is the total pressure in the pump suction branch. For cavitation-free operation, the available NPSH must be greater than the required NPSH.

$$NPSH_A \geq NPSH_R \tag{10}$$

If the suction branch is directly connected to a closed reservoir, the $NPSH_A$ equation becomes:

$$NPSH_A = \frac{p_{\tan k} - p_v}{\rho g} \tag{11}$$

For safe operation:

$$NPSH_A \geq NPSH_R + \text{margin} \tag{12}$$

The *NPSH margin* describes the safety factor by which NPSH-A must exceed NPSH-R to avoid cavitation (Fig. 4). It can be quoted in two ways: as a ratio of NPSH-A to NPSH-R. For example, an NPSH margin ratio of 1.1 indicates that NPSH-A is 10% greater than NPSH-R or as the difference between NPSH-A and NPSH-R. As a rule of thumb, it is necessary to ensure that the NPSH margin is 0.5 m or higher (that is: NPSH-A ³ NPSH-R + 0.5 m).

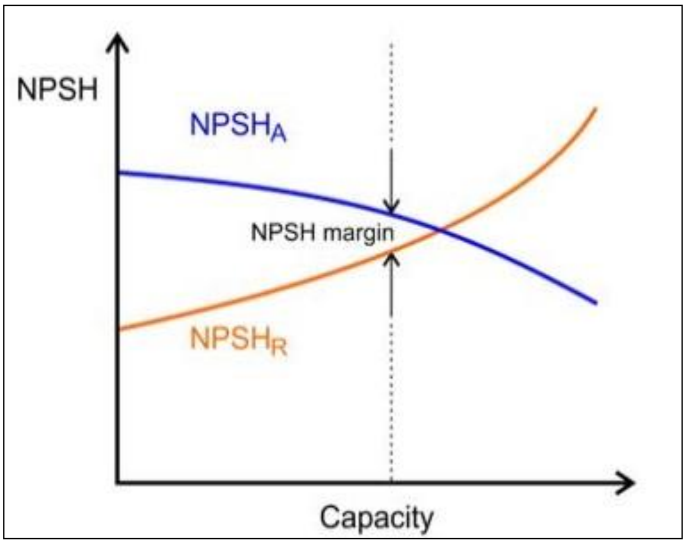

Figure 4. Variantion of PSH-R and NPSH-A with capacity (discharge flow).

A 50% NPSH margin is generally selected during the design process for pumps [26]. Cavitation appears when the pressure is decreased from this inception level; the region of cavitation enlarges, eventually starts to cause noise, performance change and possibly pump damage [27]. The latter result from the fact that beyond the inception, the pressure associated with the cavity collapse is high enough to cause the failure of (impeller) material. By the time, the inlet pressure is lowered as to cause one to three percent drop in the pump head, cavitation bubbles start to block the runner inlet and cavitation is fully established. Thus, the NPSH$_R$ (NPSH$_{3\%}$) characteristic (e.g. three-percent head drop) is defined as the NPSH, at which the total pump head decreases by three percent due to cavitation. The equation of the (NPSH$_{3\%}$) is:

$$NPSH_{3\%} = \frac{1}{g}\left(\frac{\frac{n}{60}\cdot\sqrt{\dot{V}}}{S_q}\right)^{\frac{4}{3}} \tag{13}$$

n - machine speed [rot/min];
S_q - the suction number (0.4 ÷0.5).

The NPSH$_R$ can be measured and plotted in a ($\dot{V}$, NPSH$_R$) diagram for centrifugal pumps [28]. To study the effect of cavitation on the performance of hydraulic machines, the common cavitation coefficient used for this purpose is the Thoma cavitation coefficient it is defined [29]:

$$\sigma_{th} = \frac{NPSH_A}{H} \tag{14}$$

A certain critical value is given by the following equation:

$$\sigma_{the} = \frac{NPSH_R}{H} \tag{15}$$

To avoid cavitation, it must be:

$$\sigma_{th} \geq \sigma_{the} \tag{16}$$

When the pumping bubbles are closing the pump entrance completely, resulting in the collapse of pump performance (head and efficiency), and the cavitation coefficient is called "Breakdown Cavitation Coefficient", which is defined by the following equation.

$$\sigma_{break} = \left[\frac{\sin\beta_1}{1+\cos\beta_1}\right]\Phi_1 \tag{17}$$

$$\Phi_1 = \frac{Q}{A_1 u_1} \tag{18}$$

where, β_1 is the angle of the blade at the inlet.

3. RESULTS AND DISCUSSION

Fig. 5 represents the performance of the pump used. It displays the (input power, head, and efficiency) changes according to the variation of flow rates at a constant speed. From Fig. 5 one can observe that the dependence P = f (n) is a linear function; the lowest power is at 900 [rot/min] and flow rate nearly zero end, the maximum power is at maximum speed 1400 [rot/min] and flow rate 8.6 [l/s]. Considering the experimental results obtained through laboratory tests and comparing them with those of the specialized literature, the results are acceptable [30].

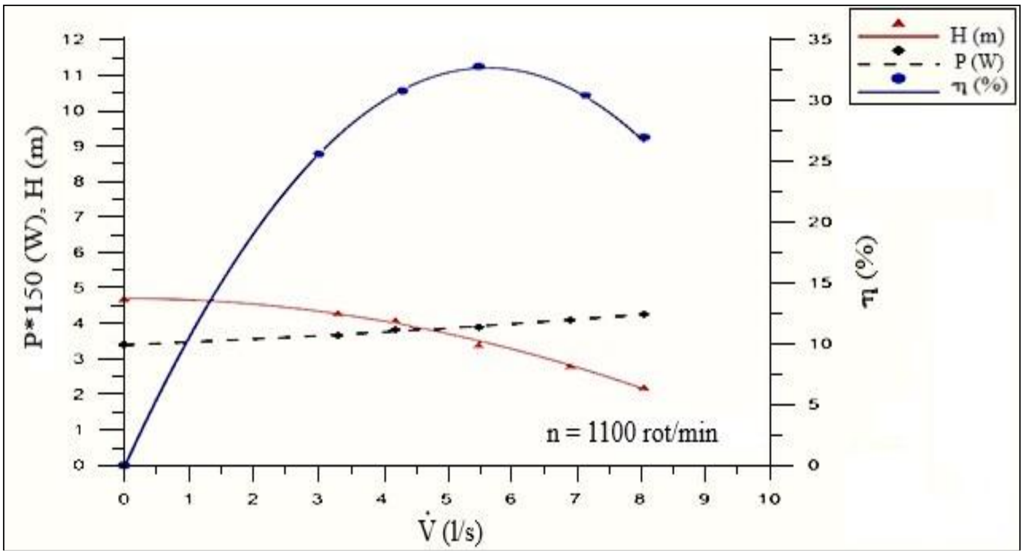

Figure 5. Pump performance (the relation between input power, head, and efficiency) and variation of flow rates at constant speed.

Fig. 6 manifests the relation between the $NPSH_R$ and the flow rate for a range of speeds 900 – 1400 [rot/min], it is found that $NPSH_R$ increases as the speed and flow rate increase. The relation between $NPSH_A$ and the flow rate for different speeds 900-1100 [rot/min], is illustrated in Fig. 7. It is obtained that $NPSH_A$ decreases when the speed and the flow rate increase because the losses increase, which lead to $NPSH_A$ decrease. It is known that cavitation appears when $NPSH_R > NPSH_A$; from Figs. 6-7 it was found that cavitation will occur in the used system at 1400 [rot/min] speed and 8.6 [l/s] flow rate.

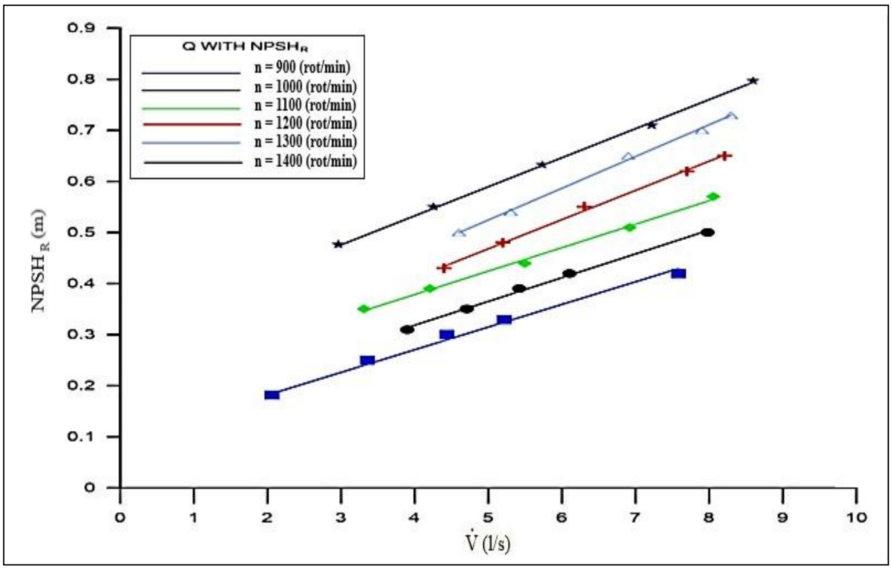

Figure 6. The relation between the $NPSH_R$ and the flow rate for a range of speeds (900 – 1400 rot/min).

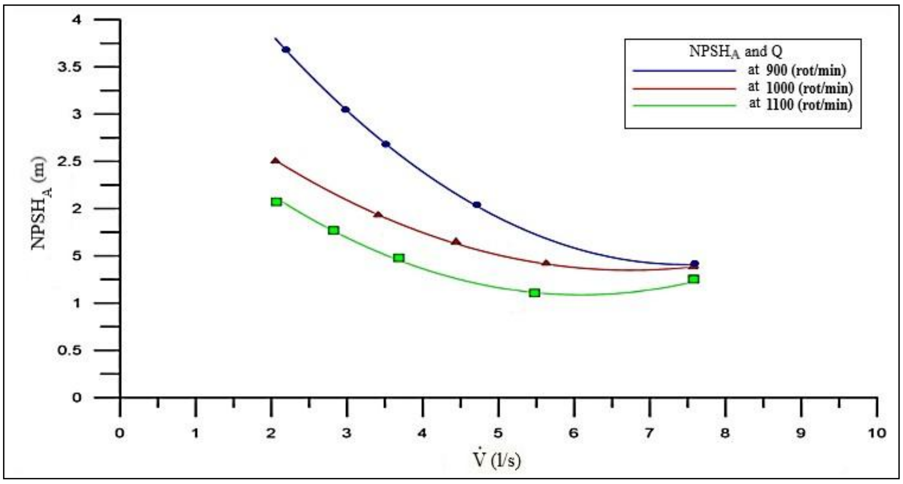

Figure 7. The relation between ($NPSH_A$) and the flow rate for different speeds (900 -1100 rot/min).

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Figure 8. Appearance of cavitation phenomenon in the centrifugal test pump.

The pressure reduces and the speed then increases which let the minimum suction head at the pump inlet 0.803 [m]. From Fig. 7, one can observe that the $NPSH_A$ is equal to 0.82 [m]. The cavitation will appear because $NPSH_R > NPSH_A$ as seen in Fig. 8, results that are similar to those in the literature [31].

Fig. 9 represents the relation between flow rate and $\boldsymbol{\sigma}_{thc}$ (Thoma critical cavitation coefficient), at which the pump head drops by (3%). Increasing the flow rate will increase the value of $\boldsymbol{\sigma}_{thc}$. Depending on the reference quantites chosen either the Thoma's cavitation factor σ, so called Thoma number or the net positive suction specific energy coefficient cψ can be chosen to define a dimension-less cavitation number. The lowest value (0.047) was found at 900 [rot/min] and 2.05 [l/s] flow rate while the highest value (0.28) was found at 1400 [rot/min] and 8.6 [l/s] flow rate.

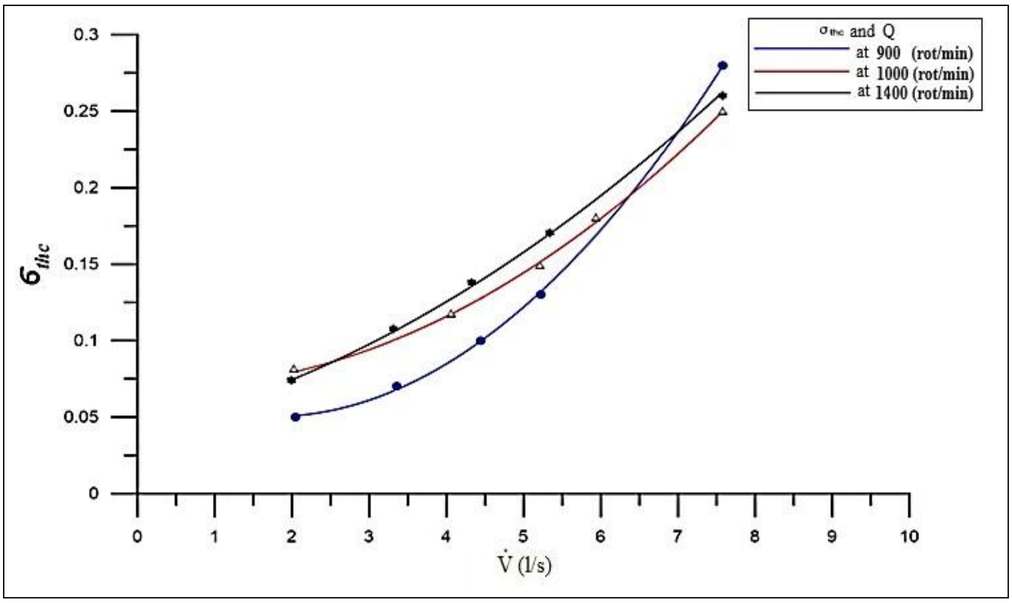

Figure 9. The relation between flow rate and σ_{the} **, at different values of flow rate.**

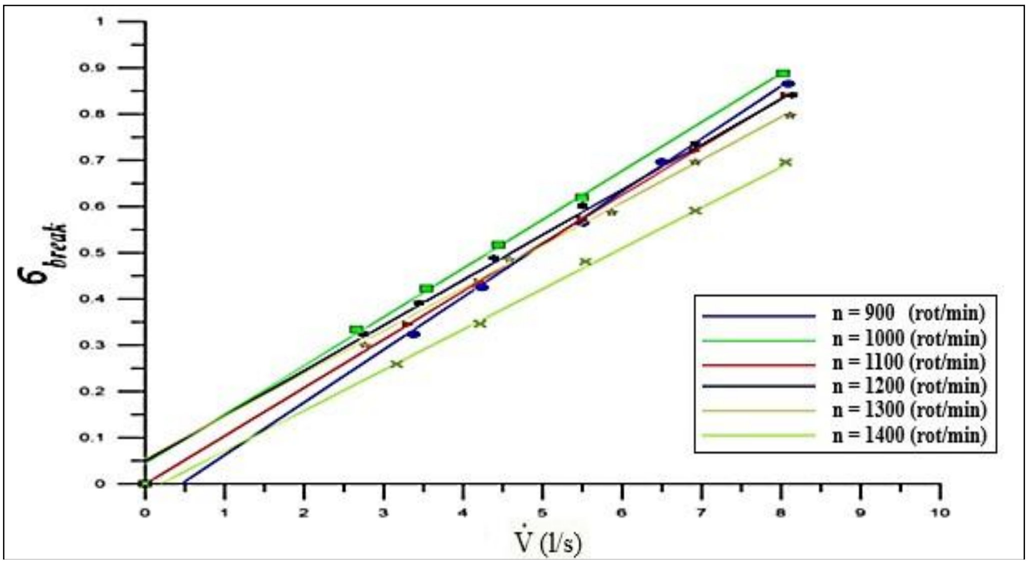

Figure 20. The relation between flow rate and σ_{break} **, at different values of flow rate.**

From Fig. 10, one can observe the relation of dependence between the flow rate and the brake down cavitation coefficient ($\boldsymbol{\sigma}_{break}$), (brake down of head and efficiency), resulting in a linear function. When the cavitation bubbles block the impeller inlet at different values of flow rate; the minimum value of $\boldsymbol{\sigma}_{\text{break}}$ at 900 [rot/min] is 0.26, and the maximum value at 1400 [rot/min] is 0.96.

4. CONCLUSIONS

Experimental work has been carried out to estimate the performance and Cavitation Phenomenon at variable speed pumps. The following conclusions have been obtained:

– When the speed is constant, and the flow rate increases, the delivery head decreases and the power will increase, while for variable speed and constant flow rate, the delivery head and the power will increase due to the reduction of the suction head.

– For variable speed pumps, the ($NPSH_A$) and ($NPSH_R$) will be different from one speed to another. Therefore, the estimation of the ($NPSH_A$) should be calculated properly according to the variable value at the suction head and the losses.

– From the results, the cavitation appears at a certain speed and suction head also at the leading edge at the vanes of the impeller. That means the critical cavitation coefficient is according to the increase in ($NPSH_R$).

– The break cavitation coefficient occurs at different speeds and suction heads; therefore, it should be noted during the design condition, i.e. the system should be shut-down to prevent the breakdown of the head and efficiency of the system.

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