Performance analysis of cavitation flow pump

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Abstract. In the current research, an experimental and theoretical investigation has been performed in non-cavitating and cavitating flow conditions, carried out in the centrifugal pump. The test used an open-loop drudge pump system to investigate the effects of Net positive suction head available (NPSHa) with a range (1 to 10 m), The centrifugal pump concert was coordinated by a flow rate with incremental values (80 to 450 m³/h), water temperature with values (12.5, 25.5, 35.5 and 45.5 °C) and pump speed with values (1000, 1200, 1400, 1600, 1800 and 2000 rpm). Thus, the theoretical study is included a two-phase flow 2-D simulation (ANSYS FLUENT software) within the centrifugal pump. It was noted that the percentage drop in the flow rate less than (2%) at a head drop of 3%, whereas the percentage drop in performance greater than (3%). This also showed that NPSHa decreases by increasing the flow rate and temperature and increases by reducing the velocity. This study showed that the noise level of the pump was increased with the decreasing NPSHa and the flow rate. Furthermore, the difference in the noise level between non-cavitating and cavitating flows was around (8 dB) (3 percent head fall). Thus, it was noted that Net positive suction head required (NPSHr) was increases by increasing the flow rate and decreases with higher temperature and lower velocity. The initiation of cavitation at NPSHa was approximately equal (4.3 m) to the flow rate (310 m³/h).

1. Introduction
The compressible and wobbly turbulent flow induced by the transfer of mass between gas and liquid phases is cavitation. Tan, Lei et al. [1], the word cavity from which the word "cavitation" originates is derived from the Latin term Gall caves, whose revenue is hollow J. Gall and KenDiani [2]. In the technique and approach of countless liquid treatment turbomachines, cavitation physics plays an important role. X. Luo [3]. In particular, vibration, erosion damage, noise, and deteriorating hydraulic efficiency can be exacerbated by cavitation J. Černetić [4] and S. Hattori, et al [5]. Therefore, cavitation refers to the creation of vapor bubbles in counties within the fluid flow region when the absolute liquid pressure decreases to the vapor pressure equal to the operating temperature of the liquid. S. Zhang, et al. [6]. An enhanced cavitation presentation for centrifugal pumps has lower blade inlet consignment and higher sideways blade suction intensity. If the collapse happens on a solid 's surface, the liquid roar can damage small areas with immense contained pressures, pitting the surface and eroding it to fill the blank space left by the bubbles. V. Minster and J. Proost [7]. Their research focuses on the effect of the cavitation phenomenon on the efficiency of the centrifugal pump (the type of volute casing), the pressures and temperatures expected to occur during the spherical collapse in the gas inside the bubble.
Pitting triggers blades. H. Liu, et al [8]. Solid surface damage is characterized as the high viscosity of the specific model of Navier Stokes reduces the variability of cavitation and therefore makes it difficult to determine the impartiality of the bubbles. A. Steinmann et al [9]. The cavitation effect of the impeller passing frequency on the pump torque is very similar to the flow range of 36 to 0 m³/h observed in the other variables. As for suction pressure and vibration, the main difference occurs with flow rates greater than 42 m³/h of the impeller frequency portion and torque. Despite this, whenever greater sensitivity is needed, changes in the root-mean-square value M. M. Stopa, et al [10]. Subsequently, the straight variables associated with the pump's hydraulic presentation, bubbles enter an exclusive pump area wherever the pressure is greater than the Chahine fluid vapor pressure, G. L. Chahine, et al [11]. In addition to the excitation pressure stage driving the subtleties of the bubble, the comparative proportions between the bubble and its maximum volume then the dimension expands the material wall. W. M. Konč, et al. [12]. The lowest NPSH caused by a centrifugal pump cavitation corrosion container only after vapor bubbles collapse near the anxious boundary presented in a bolted circuit in a centrifugal impeller where N=1150 r/min the national cavitation indicates a rather industrialized piece of cavitation at NPSH=-3 percent the critical amount of NPSH erosion increases first, thus increases designed to decrease erosion J. Pei, et al [13]. A centrifugal pump impeller achieves improved cavitation and hydraulic recital for enhanced cavitation presentation to the enhanced impeller and the NPSHR decreases by 0.63 m relative to the ground-breaking model. The best mixture, D1=196 mm, β=5.5 °, π=130 °, was obtained for pump cavitation. The lowest NPSH that a centrifugal pump cavitation corrosion container is induced only after vapor bubbles collapse near the boundary anxious presented in a bolted circuit in a centrifugal impeller that N=1150 r/min the national cavitation indicates a very industrialized cavitation piece at NPSH=-3% the critical NPSH the first supplies erosion number, Thus, the lowest NPSH is built for a decreasing NPSH that a centrifugal pump cavitation corrosion container is induced only after vapor bubbles collapse near the boundary anxiously presented in a bolted circuit in a centrifugal impeller that N=1150 r/min the national cavitation indicates a rather industrialized cavitation piece at NPSH=-3 percent the critical NPSH erosion increases first, thus increases designed to decrease erosion C. Nwaoha [14] Cavitation remains a recognized problem popular centrifugal pumps, with the latent loss of function in addition to inflicting thoughtful harm, elucidating the different causes of cavitation. G.Ziervogel et al [15]. In the cavitation of the model, the hydrodynamic study of the centrifugal pump is applied to the viscid Reynolds-averaged Navier-Stokes solver, a numerical confirmation of the current cavitation even in the case of low flow speeds. J. Choi and S. L. Ceccio [16]. Noise release was tested for the collapse of a cavitation bubble starting in the middle of a streak vortex and the lengthened cavitation bubble stands experimentally prejudiced by the fundamental non-validating vortex holdings cavitation prediction method based on impeller pressure on is a surface at single-phase media. A new cavitation performance prediction method is proposed, and the feasibility of this method is demonstrated in combination with experiments, which will greatly accelerate the pump hydraulic optimization design. A. Stuparu [17] proposed a new model to recognize the cavitation intensity of axial piston pumps. To improve the recognition accuracy under a noisy environment. In this work the inserted pressure concentration when using the centrifugal-pumped drudge pump was experimentally calculated by reducing the suction pressure of the pump below the critical pressure and then evaluating the critical point for each curve after measuring the 3 % head drop value. Using packet ANSYS-FLUENT 2020R1, numerical simulation was performed. In comparison with experimental and numerical conditions, steady-state, Newtonian flow, incompressible, and three-dimensional.

2. Experimental Device and Procedure.

2.1 Test Section Regime.
1. The inner pump inner flow and caviting of the centrifugal pump under various circumstances. One test between the experiments with values of flow rate. Water temperature and pump speed (170-450 m³/h), (12.5-45.5), and (1000-2000 rpm), respectively. Underway to explain the cavitation flow within
the domain by adjusting NPSHa values, Temperature, and pump speed, according to which many contours were planned. Inlet pressure at the inlet zone (pump inlet port) was compulsory.

2. The outer pump threaded pipe (250 mm) to connect the centrifugal pump discharge pipe is made of several lengths of steel pipe with flanged ends. A cylindrical vacuum pipe was manufactured in the local market by using carbon steel plates with a thickness (20 mm), the net height for this pipe was (300 mm) with an internal diameter and (250 mm) outlet. A pressure gauge is often used to calculate the centrifugal pump output pressure at the inlet.

2.2 Description of Experimental Test.

The schematic diagram of the experimental test is shown in Figure (1) the drudge-pumped excavator was carried out on the experimental test platform, as the following procedure:

1- The vacuum suction-pipe was filled until the water reaches the water level indicated.
2- The centrifugal pump was turned on to circulate the water inside the system.

In this case, as a first reading was attaching on inlet/outlet, all the following parameters were recorded: water temperature, water flow rate, pump suction pressure, pump discharge pressure, pipe pressure, and pump noise level.

The effect of changing flow rate, pump speed, and water temperature the effect of NPSHa decreasing [20] (which occurs as a result of vacuum process) were investigated in the experimental work. Flow Rate values was (160, 220, 310, 375, 390, 400 and 450) m³/h, water temperature values were (12.5, 25.5, 35.5 and 45.5) °C, whereas Density of water 998.2 kg and Viscosity 1.003 \times 10^{-3} \text{kg/m.s}, Figure (2).

2.3 Measuring Devices.

1. Flow Measurement; To calculate the volumetric flow rate of the system a water variable-area flow meter (Digital rotameter) with a range of (80 to 450) m³/h was installed. In leading space, the transparent flow meter screen is the subject matter.
2. Pressure measurement; Before and after the centrifugal pump, two pressure gages with a range of (–1 to 3) bars were installed to measure the static pressure at these points.
3. Temperature measurement; A dial Digital thermocouple with a range of (0 to 120) °C has been used to quantify the water temperature of the pipe. This electronic thermocouple was set for centrifugal pumps respectively at the inside and outside of the pipe between the suction and discharge ports.
4. Pump Speed Measurement; A tachometer revolution-counter, tachometer, rev-counter, RPM gauge is a device measuring a shaft's rotation speed range of (0 to 30) 100 rpm was used for the centrifugal shaft's velocity portion.
5. Pump Noise Measurement; To measure the noise level that is often increased in pumps when cavitating flow occurs, a digital sound level meter (decibel meter) was used. Measuring Ranges = low (30-70) db, medium (60-100) db, and high (90-130) db. The medium-range was designated for all tests. The source of noise (pump casing here) with a small distance between them to avoid the vibration that occurs in the pump [17]. the display reads within “94±0.2” db.
Figure 1: Test drudge pump

Figure 2: Model of the Centrifugal-pump [volute, Impeller].

| Parameter              | Value         |
|------------------------|---------------|
| Rated discharge $Q_d$ ($\text{m}^3/\text{h}$) | 170-450       |
| Rated head $H$ (m)     | 12-40         |
| Rotational speed $n$ (r/min) | 1000-2000   |
| Aver. Hydraulic Power (kW) | 15.95      |
| Number of blade $Z$    | 4             |
| Impeller diameter $D_2$ (mm) | 760        |
| Blade width at exit $b_2$ (mm) | 60            |
2.4 Experimental Work.

The following operating conditions were used to estimate the performance of the cavitation in the centrifugal pump: the experimental test through flow rate values, water temperature and pump speed (400 m³/h, 25.5 °C, and 1200 rpm) respectively were employed in this boundary condition, and foretold curves were plotted conferring to that. In the first boundary condition, cavitation can be attained by reducing both the inlet and outlet pressures gradually till the inlet pressure reaches the cavitation conditions. Whereas in the second boundary condition, only the outlet pressure will be reduced slowly, and in turn, the predicted inlet pressure will decrease automatically to spread the cavitation conditions, while the pump inlet velocity remained constant.

2.5 Performance Variables of Experimental Work.

The experimental work investigation is constructed upon the accuracy of the measurement device, the uncertainty for present work are calculated for NPSHa and efficiency. The entire head of the pump (H) is defined as the alteration of liquid energy, stationary pressure heads, velocity heads, and promotions in meters of liquid between all three discharge and suction mechanisms, and can be located by the following equation [3]:

\[
H = \frac{P_d - P_s}{\rho g} + \frac{v_d^2 - v_s^2}{2g} + (Z_d - Z_s)
\]  

If the discharge and suction diameters of the pump ports are the same as in the test pump, then \(v_d = v_s\), therefore, equation (1) can be rewritten as follows [1]:

\[
H = \frac{P_a - P_s}{\rho g} + \text{Height difference between measuring points}
\]

where the height difference between the measuring points \((Z_d - Z_s)\). The whole pump head (H) is defined as the alteration between all three mechanisms of the discharge and suction liquid energy, stationary pressure heads, velocity heads, and promotions, which is in meters of liquid, and it can be located signified by the following equation [3]:

\[
H = \frac{P_a - P_s}{\rho g} + \text{Height difference between measuring points}
\]

There is a significant parameter called the net positive suction head (NPSHa), which is premeditated as follows [20]:

\[
NPSHa = \frac{P_s + \frac{v_s^2}{2g} - \frac{P_v}{\rho g}}{\rho g}
\]

Another parameter called (suction specific speed \(N_{sS}\)) is a dimensionless measure that describes the suction physiognomies of a pumping system. This parameter can be computed by the following equation [10]:

\[
N_{sS} = \frac{\sqrt{\frac{\rho g Q^2}{\pi NPSHa^2}}}{}
\]

Cavitating flow is commonly described either by Thoma cavitation number \(\sigma_{TH}\) or by cavitation number \(\sigma\) (in addition to NPSHa), and these two dimensionless numbers are defined as follows [7]:

\[
\sigma_{TH} = \frac{NPSHa}{H}
\]

\[
\sigma = \frac{P_s - P_v}{\frac{1}{2} \rho U^2}
\]

where \(U\) represents the inlet-blade tip speed = \(U_{eye} = \Omega R_{IT}\) and \(R_{IT}\) is the inlet-blade tip radius. Also, there are two important coefficients usually used in the non-cavitating and cavitating flow fields, one of them relating to the pump head and the other to the flow rate [20].

\[
\psi = \frac{gH}{R_{2T}^2 \phi^2}
\]

where \(R_{2T}\) represents the outlet-blade tip radius [14].

\[
\phi = \frac{Q}{A_{2T} R_{2T}^2}
\]
where $R_2$ represents the outlet radius of the impeller and $A_2$ represents the impeller outlet area ($A_2 = 2πR_2b_2$). Where the width-blade $b_2$ is at the outlet of the impeller [3]. In the test pump $R_2 T$ is $R_2$ because its impeller has shrouded blades as shown in figure (2), where the values of $R_2$ and $b_2$ are (760 mm) and (250 mm) respectively.

To calculate the pump efficiency, pump hydraulic power, and shaft power.

The water horsepower and can be calculated as follows [8]:

$$P_H = \rho gQH$$  \hspace{1cm} (9)

The engine power ($P_M$) is the power required to drive the pump shaft, which is usually called the brake horsepower. The power of the engine can be calculated with the following equation [12]:

$$p_{sh} = T_\omega = T_\omega \frac{2\pi N}{60} = m(U_2C_2 - U_2C_2) = mU_2C_2$$  \hspace{1cm} (10)

$$P_{sh} = P_H$$  \hspace{1cm} (11)

$$H = \frac{U_2^2}{g} - \frac{U_2Q_{imp}}{\pi D_2 b_2 \tan \beta}$$  \hspace{1cm} (12)

The engine efficiency at this value of the power factor was estimated to be (80%) according to the typical variations curve of the power factor and engine efficiency with load [10].

Now, the ratio of pump hydraulic power to shaft power provides capacity for the pump ($\eta_p$) which represents the hydraulic efficiency and its equation as follows [15]:

$$\eta_p = \frac{P_H}{P_M} = \frac{\rho gQH}{\eta_M}$$  \hspace{1cm} (13)

3. Numerical Analysis and Boundary Condition.

The pressure, flow rate, velocity, and temperature drop distribution for straight pump were predicted by using the ANSYS FLUENT 2020R1 package. The following assumption is applied on cavitation flow for the water of centrifugal-pump, to get a proper solution, the following steps were followed in ANSYS FLUENT software to simulate the stream inside the computational domain:

The fluid flow can be depicted by the continuity, Naviers stokes, and energy equations in differential form. The continuity equation used for the mixture [13] can shows as follows;

$$\frac{\partial}{\partial t} (\rho_m) + \nabla \cdot (\rho_m \vec{v}_m) = 0$$  \hspace{1cm} (14)

$$\vec{v}_m = \frac{\alpha_L \rho_L + \alpha_V \rho_V}{\rho_m} \vec{v}$$  \hspace{1cm} (15)

$$\rho_m = \alpha_L \rho_L + \alpha_V \rho_V$$  \hspace{1cm} (16)

$$\alpha_L + \alpha_V = 1$$  \hspace{1cm} (17)

The momentum equation for the mixture container can be obtained by summing up the discrete momentum equations for entire phrases and articulating them as:

$$\frac{\partial}{\partial t} (\rho_m \vec{v}_m) + \nabla \cdot (\rho_m \vec{v}_m \vec{v}_m) = -\nabla P_m + \nabla \cdot \left[ \mu_m (\nabla \vec{v}_m) + \nabla \vec{v}_m^T \right] + \rho_m g + F + \nabla \cdot \left[ \sum_{k=1}^{2} \alpha_k \rho_k \vec{v}_{dr,k} \vec{v}_{dr,k} \right]$$  \hspace{1cm} (18)

$$\mu_m = \alpha_L \mu_L + \alpha_V \mu_V$$  \hspace{1cm} (19)

$$\vec{v}_{dr,V} = \vec{v}_V - \vec{v}_m$$  \hspace{1cm} (20)

The relative velocity also called the slip velocity is defined in place of the vapor phase velocity relative to the liquid phase velocity:

$$\vec{v}_{VL} = \vec{v}_V - \vec{v}_L$$  \hspace{1cm} (21)

The mass fraction for the two phases is defined as,
\[ f_L = \frac{a_L \rho_L}{\rho_m} \]  
\[ f_V = \frac{a_V \rho_V}{\rho_m} \]  

(22)  
(23)

The drift speed and the relative velocity are related by the following expression:

\[ \vec{v}_{dr.V} = \vec{v}_{VL} - (f_L \vec{v}_{VL} + f_V \vec{v}_{LV}) \]  

(24)

The mixture model applied in FLUENT uses numerical slip formulation; the relative velocity form is specified by:

\[ \vec{v}_{VL} = \frac{\tau_v}{c_{drag}} \left( \frac{\rho_V - \rho_m}{\rho_V} \right) \vec{a} \]  

(25)

where \( \tau_v \) is the bubble relaxation time.

\[ \tau_v = \frac{\rho V d_b^2}{18 L} \]  

(26)

the default drag function \( c_{drag} \) is calculated as follows from Adrian Stuparu [17].

\[ c_{drag} = \begin{cases} 
1 + 0.15R_e^{0.687} & R_e \leq 1000 \\
0.0183 & R_e > 1000 
\end{cases} \]  

(27)

And acceleration is a type of:

\[ \ddot{a} = \ddot{g} - (\vec{v}_m \cdot \nabla) \vec{v}_m - \frac{\partial \vec{v}_m}{\partial t} \]  

(28)

3.1 Mesh Generation

The explanation of the physical process by using the numerical solution was used by the grid generation as the first step. The result of the solution depends upon the excellence of the grid. In the current study, tetrahedral cells are used for three dimensions of design for a good mesh with higher correctness solution as shown in figure (3). For multifaceted model and mesh generation, the number of cells increases with the time resolution equally the accuracy increases that depends on the aptitude of computer, memory, and process to solve and proposal good mesh in this study the number of nodes (106679) and the number of the element (562773). The steady-state pressure-based solver with standard K-epsilon model was selected because of low speed incompressible turbulent flow model \( (k-\epsilon) \) is used in the present analysis to place fluid flow and cavitation in the centrifugal pump.

![Figure 3. Mesh Generations.](image-url)
4. Experimental Results.
Different experiments on the experimental test were conducted to study the effect of changing the pump head (H), pump flow rate (Q), pump efficiency (ηp), net positive suction head (NPSHa), net positive suction head required (NPSHr), cavitation number (σth), cavitation number (σ), suction specific speed (Ne), head coefficient (ηp), flow coefficient (ηp) and pump noise level. In previous studies to describe the cavitation progress stages is NPSHa, (σth) and (σ), where some parameters have been plotted versus the NPSHa and the remaining have been plotted versus the (σth), then determining the critical point for each curve after calculating the value of 3% head drop.

4.1 (Head ~NPSHa) Curves.
Figure (4) shows the effect of NPSHa on the pump head at various flow speeds, which means that when the NPSHa value decreases as a result of the vacuum cycle, the head values initially remain approximately constant (there is no cavitation at this point). A further reduction in the NPSHa will decrease the head, where this process continues until the 3% drop in the head has been reached (i.e., NPSHa becomes equal to NPSHr). Reducing NPSHa means that decreasing the inlet pressure (suction pressure) of the pump according to Eq. (3). A large number of vapor bubbles will block the impeller channels, which results in a sudden decrease in the pump head [1] and [6]. the inception of cavitation occurs at NPSHa (which called Incipient net positive suction head (NPSHi)) is approximately equal to (4.3 m) for flow rate (310 m³/h), where the flow with NPSHa less than (4.3 m) represents a cavitating flow for all flow rates as indicated in this figure.

![Image](image_url)

**Figure (4):** Influence of NPSHa on the head for different flow rates

Figure (5) Indicates the effect of NPSHa on the pump head at different temperatures, which indicates that the NPSHa and NPSHr values decrease with increased temperature. Figure (6) indicates the influence of NPSHa on the pump head for different pump speeds, which shows that decreasing the speed leads to an increase in the NPSHa and decreases the NPSHr; where the decrease in speed leads to an increase in the pump inlet pressure and decrease the inlet critical pressure.
4.2 (*Flow Rate-NPSHa*) Curves.

Figure (7) indicates the influence of NPSHa on the pump flow rate for different flow rates, which illustrates that the flow rate curves have the same behavior as the head curves concerning the effect of decreasing NPSHa which leads to a sudden drop in the pump flow rate due to blocking the impeller channels with vapor bubbles. Figure (8) indicates the effect of NPSHa on the pump flow rate at dissimilar pump speeds, which indicates that the flow rate decreases as the flow rate is directly proportional to the velocity of the pump.
Figure (8): Influence of NPSHa on the flow rate for different speeds

4.3 (Efficiency NPSHa) Curves and (Thoma Cavitation Number-NPSHa) Curves.

Figure (9) Indicates the impact of NPSHa on pump efficiency at different flow levels, the effect of NPSHa decreasing, where the percentage of efficiency drop at 3% head drop is more than 3% because the drop in both the head and flow rate has been considered in calculating the efficiency according to Eq. (13). Figure (10) indicates the influence of NPSHa on the Thoma cavitation number for different flow rates, which illustrates that the Thoma cavitation number decreases linearly as NPSHa decreases because it is directly proportional to the NPSHa according to Eq. (5). Thoma cavitation number (at 3% head drop) increases with increasing the flow rate.

Figure (9): Influence of NPSHa on the efficiency for different flow rates.

Figure (10): Influence of NPSHa on the Thoma cavitation number for different flow rates.

4.4 (Noise Level -NPSHa) Curves.

Figure (11) Indicates the effect of NPSHa for various flow levels on the pump noise level, which shows that when the NPSHa decreases the noise level increases. This figure also shows that the difference in the pump noise level between non-cavitating and cavitating flow (at 3% head drop) is about 8 db, where the pump noise level at non-cavitating flow conditions is about 72 db whereas the noise
level in the surroundings is approximately equal to 51.8 db before turning on the pump. Also, it is seen in this figure that the noise level decreases slightly with increasing the flow rate.

4.5 (Suction Specific Speed –Thoma Cavitation Number) Curves and (Head Coefficient –Thoma Cavitation Number) Curves.

Figure (12) indicates the influence of Thoma cavitation number on the suction specific speed for different flow rates, which shows that the common velocity of suction increases nonlinearly as the Thoma cavitation number decreases, according to Eq. (4) the suction specific speed (at 3% head drop). Figure (13) specifies the effect of the Thoma cavitation number on the head coefficient for various flow rates, the head coefficient is directly proportional to the head according to Eq. (7).

Figure (11): Influence of NPSHa on the pump noise level for different flow rates.

Figure (12): Inspiration of Thoma cavitation number on the suction specific speed

Figure (13): Inspiration of Thoma cavitation number on the head coefficient for different flow rate
5. Numerical Results.

CFD simulation has been performed by using ANSYS FLUENT software to analyze the cavitating behavior inside the centrifugal pump.

**Case 1- pressure inlet and pressure outlet**

Figures 14, 17, 20, 23, 26, and 29 show one test among the experiments with values of flow rate, water temperature, and pump speed has been employed to illustrate the computational domain contours of the cavitation regions concerning the experimental results by changing the following:

1. NPSHa with values (1.45, 1.9, 2.52, 2.77, 3.22, and 4.31 m).
2. Water temperature with values (12.5, 25.5, 35.5, and 45.5 °C).
3. Pump speed with values (1000, 1200, 1400, 1600, 1800, and 2000 rpm).

**Case 2- velocity inlet and pressure outlet**

Figures 15, 16, 18, 19, 21, 22, 24, 25, 27, 28, 30, and 31 show the second test experimental with values of flow rate, temperature and pump speed, cavitation condition has been adjusted by decreasing the outlet pressure while the inlet velocity has been remained constant the head remains approximately constant where this process continues until the 3% drop in the head has been reached after the critical point.

![Figure 14: Cavitation regions (NPSHa= 4.31 m, 35.5 °C and 2000 rpm).](image)

![Figure 15: Static pressure distribution (NPSHa= 4.31 m, 35.5 °C, and 2000 rpm)](image)

![Figure 16: Velocity distribution (NPSHa= 4.31 m, 35.5 °C and 2000 rpm)](image)

![Figure 17: Cavitation regions (NPSHa= 3.22 m, 35.5 °C and 1800 rpm).](image)
Figure (18): Static pressure distribution (NPSHa= 3.22 m, 35.5 °C and 1800 rpm)

Figure (19): Velocity distribution (NPSHa= 3.22 m, 35.5 °C and 1800 rpm)

Figure (20): Cavitation regions (NPSHa= 2.77 m, 35.5 °C and 1600 rpm)

Figure (21): Static pressure distribution (NPSHa= 2.77 m, 35.5 °C and 1600 rpm)

Figure (22): Velocity (NPSHa= 2.77 m, 35.5 °C and 1600 rpm)

Figure (23): Cavitation regions (NPSHa= 2.52 m, 35.5 °C and 1400 rpm)
Figure (24): Static pressure distribution (NPSHa= 2.52 m, 35.5 °C and 1400 rpm)

Figure (25): Velocity distribution (NPSHa= 2.52 m, 35.5 °C and 1400 rpm)

Figure (26): Cavitation regions (NPSHa=1.9 m, 35.5 °C and 1200 rpm).

Figure (27): Static pressure distribution (NPSHa= 1.9 m, 35.5 °C and 1200 rpm)

Figure (28): Velocity (NPSHa= 1.9 m, 35.5 °C and 1200 rpm)

Figure (29): Cavitation regions (NPSHa=1.45 m, 35.5 °C and 1000 rpm).
6. Conclusions

The usage of the centrifugal pump with non-cavitating and cavitating flow of the head must decrease to meet the critical point (3% head drop at concentration) at and the volume flow rate of 310 m$^3$/h must result in an NPSHa increase, these cavitation regions expand with decreasing the NPSHa and increasing the temperature whereas they reduce with decreasing the speed. Also, it was found that the inception of cavitation occurs at NPSHa is approximately equal to (4.3 m) for flow rate (310 m$^3$/h). To order to avoid cavitation, the NPSHa must be consistently greater than the NPSHr with an adequate safety margin of at least 0.6 m [6] to prevent any major pump head adjustments. The simulation results by using ANSYS FLUENT 2020R1 package gives in agreement with experimental work the correlation is predicted and a time step of the calculations should be the amount of case was (5-6) hour.
Nomenclature

| Symbol | Meaning                  | Unit |
|--------|--------------------------|------|
| $c_{drag}$ | Drag function          | –    |
| D      | Impeller diameter       | m    |
| $d_V$  | Bubble diameter         | m    |
| f      | Friction factor         | –    |
| g      | Gravitational acceleration | m/s² |
| $G_k$  | Turbulent kinetic energy generation term | m²/s² |
| H      | Pump head               | m    |
| $h_f$  | Frictional head loss    | m    |
| T      | Torque                  | N m  |
| NPSHa  | Net positive suction head available | m |
| NPSHi  | Incipient net positive suction head | m |
| NPSHr  | The net positive suction head required | m |
| $N_{ss}$ | Suction specific speed | –    |
| P      | Static pressure         | Pa   |
| $P_H$  | Pump hydraulic power    | Watt |
| $\rho_{sh}$ | shaft power         | Watt |
| Q      | Volumetric flow rate    | m³/s |
| $R_c$  | Condensation rate term  | –    |
| Re     | Reynolds number         | –    |
| Temp   | Temperature             | °C   |
| U      | Inlet-blade tip speed   | m/s  |
| $v$    | The average velocity of flow | m/s |
| $\Omega$ | rotational speed       | rad/s |

Greek Symbols

| Symbol | Meaning                  | Unit |
|--------|--------------------------|------|
| $\gamma$ | Effective exchange coefficient | –    |
| $\varepsilon$ | Turbulent dissipation rate | m²/s² |
| $\eta_p$  | Pump efficiency          | –    |
| $\beta$  | Relative flow angle at the impeller inlet, outlet | – |
| $f$      | Mass fraction            | –    |
| $\rho$   | Density                  | kg/m³ |
| $\sigma$ | Cavitation number        | –    |
| $\sigma_D$ | Prandtl dispersion coefficient | – |
| $\sigma_{TH}$ | Thoma cavitation number | – |
| $\phi$   | Flow coefficient         | –    |
| $\psi$   | Head coefficient         | –    |
| $\Omega$ | Angular speed = $\pi N/30$ rad/s |

Abbreviations

| Abbreviation | Meaning                  |
|--------------|--------------------------|
| 2-D          | Two-Dimension            |
| 3-D          | Three-Dimension          |
| BEP          | Best Efficiency Point    |
| CFD          | Computational Fluid Dynamics |
| db           | Decibel                  |

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7. References

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