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# Improvement of the Centrifugal Pump Performance by Restricting the Cavitation Phenomenon

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The phenomenon of cavitation and suction lusts are among the most important factors in the installation and installation of centrifugal pumps, which can damage the pump blades, so they should be prevented from occurring. Therefore, in this paper, a new layout before the centrifugal pump was presented and investigated in order to prevent the two phenomena mentioned; all of these results indicated that the pump performance was improved. Also, in reviewing this new method, we are trying to optimize and consume excess energy by restricting the cavitation phenomenon in the industrial sector.

## 1. Introduction

In pumps, cavitation and suction loose happen, while air bubbles and vapour flow into the high-pressure operation of the pump and then both are disintegrated by the dynamic operation of the impeller, diminish the efficiency and causes corrosion in the structure of the pump and propeller (Johann Friedrich gulich et al., 2014; Wang 2017; Tian et al., 2018; Xu et al., 2018; Sahebi et al., 2016). Cavitation causes vibrations in unspecified frequencies that makes the biased hydraulic forces on the impeller and the shaft, plus the premature failure of wells and bearings of the pumps (Gu et al., 2015; Bala et al., 2018; Hoseinzadeh et al., 2014; Jouybari et al 2012; Hoseinzadeh et al., 2014). As we know that fluid's flow inside a tubular or enclosed structure can exert centrifugal and Coriolis force to the structure (Guo et al., 2017; Hoseinzadeh et al., 2018; Yari et al., 2015; Sahebi et al., 2017; Ghasemi et al., 2012; Ghasemi et al., 2015; Ghasemiasl et al., 2018); so probing the places of the pump, in which the fluid's flow is occurring is crucial for improving an accurate model by Fluent or MATLAB software (Zhang et al., 2017; Cannistraro et al., 2018; Du et al., 2018; Fabiano et al., 2015; Fabiano et al., 2017; Hoseinzadeh et al., 2014; Bahrami et al 2015; Habibi et al., 2016; Yousef Nezhad et al., 2017). Rotating fluid flows can occur in the following states as Figure 1 (Askew, 2011): (1) Rotating flows on the ends of the blades, (2) Ere the liquid entrance region inside the impeller (inlet of the pump), (3) throughout the area within the assembly of the pump and impeller. Rotating fluid flows appearance modifies the flow and pump pressure in the field of occurrence, and it generates vibrations in the frequency of passing the blade, too. It should be noted that the optimization of these classifications, simultaneously including the use of loss in the manufacturing as an operating system of pumps performance, has not been attempted till now.

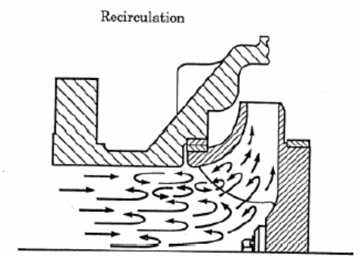



*Figure 1: rotating flows (Askew 2011)*

Also, to restrict the conditions for cavitation of pumps, as can be seen in Figure 2, in which cavitation was happening is perceived by continually reducing the pressure inside the reservoir (in a vacuum pump), and vibration and operating conditions of the pump are controlled (Shojaeifard et al., 1999). For Preventing cavitation and suction loss, the following methods can be used: Increasing the pressure of the suction tank. Reducing the vapour pressure of the liquid by cooling. Raising liquid level in suction source. Taking down the pump, the results of this action will be as before. Reduce energy loss in the suction pipe. Using the booster pump in suction part.

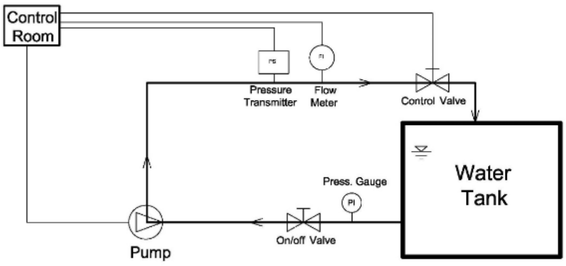
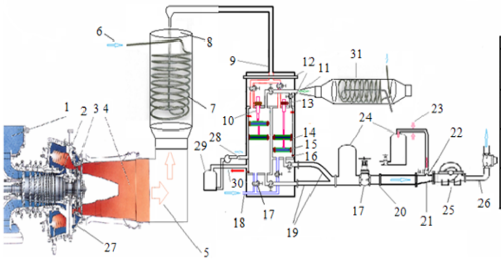



*Figure 2: Planning the cavitation measuring (Askew 2011)*

## 2. Designing and analysing of operational system

As introduced, one of the most critical parameters for restricting the phenomenon of cavitation in the centrifugal pumps is to enhance the fluid pressure in the suction unit. To perform this essential in the design of pre-commissioned pumps, using the surplus energy of the power plant sector and implement the necessary power to provide the pressure required before the pumping is suggested. In this process, the hot gases from a gas turbine are entered toward an aluminium fin exhaust pipe, which has the proper thermal exchange, and minimal corrosive effect. Then, by inserting sea water, the heat exchanger can be a vapour pressure outlet, which depends on the pressure of the water quantity of the heat exchanger. First, the piston in each compartment is embedded with a hydraulic motor, suction fluid and the vapour pressure to the rear of the cylinder, and the other chamber is ready for suction. This water vapour cooled by extra heat exchanger.

Additionally, during the cooling toward the heat exchanger, the temperature of fluid has been increased. By depositing the liquid under pressure in another small reservoir, it fulfils the inlet of the desired pump. Hence, due to the continuation of the pressure, the cylinder tanks in the planned pipelines has been applied to keep the pump's inlet in a high pressure, which is the intention sought to prevent cavitation and suction loss. In performing the method, it should be noted that the tank before pumping level is due to the appearance of such a source, to block flow fluctuations from the piston-cylinder chamber. After setting the pressure and flow in the tank, the single valve is automatically opened and admitted to entering the fluid. Moreover, by connecting a tube to that, it has been detected by the bubble germination at the inlet to eliminate the condensed air and prevent reheating of the pump. In this study, fluid temperature increases slightly, therefore it can be removed by cooling the piston-cylinder enclosure wall using a swirling water system. Figure 3 shows this suggested model in full based on the design of fin tube heat exchangers (Kohzadi et al., 2018). In before-mentioned design, by designing a boiler with thermal energy from the exhaust gases of one or more gas turbines, water can be pressurized with steam. By transferring power to gears, another piston moves in a double-sided cylinder to increase the fluid pressure for pumping with a centrifugal pump, and implement the previous conditions as the sedimentation tank obtained.



| | | | 1 | 2 | 3 | 4 |
|---|---|---|---|---|---|---|
| Typhoon (ISO) 4.35MW(e) | PRESSURE | PSIA | 14.7 | 188.4 | 183.6 | 14.7 |
| | | kPa | 101.3 | 1299 | 1266 | 101.3 |
| | TEMPERATURE | °C | 15 | 371 | 1070 | 546 |
| | | °F | 59 | 698 | 1958 | 1015 |
| Typhoon (ISO) 4.70MW(e) | PRESSURE | PSIA | 14.7 | 200.4 | 195.3 | 14.7 |
| | | kPa | 101.3 | 1382 | 1347 | 101.3 |
| | TEMPERATURE | °C | 15 | 387 | 1070 | 540 |
| | | °F | 59 | 728 | 1958 | 1004 |
| Typhoon (ISO) 5.05MW(e) | PRESSURE | PSIA | 14.7 | 209.5 | 204.2 | 14.7 |
| | | kPa | 101.3 | 1445 | 1408 | 101.3 |
| | TEMPERATURE | °C | 15 | 400 | 1110 | 550 |
| | | °F | 59 | 751 | 2030 | 1022 |
| Typhoon (ISO) 5.25MW(e) | PRESSURE | PSIA | 14.7 | 217.5 | 214 | 14.7 |
| | | kPa | 101.3 | 1500 | 1476 | 101.3 |
| | TEMPERATURE | °C | 15 | 405 | 1110 | 552 |
| | | °F | 59 | 761 | 2030 | 1026 |

*Figure 3: Overview of the project and described items [Exhaust(5), Cold water inlet(6) Blade tubes(7) Gas outlet(8) Steam Inlet(9) Hydraulic motor(10) Steam Outlet(11) Opening and closing the tank(12) Washbasin(13) Piston sealing(14) Insulation(15) Gauge level sensor(16) One way automatic valve(17) Tank inlet(18) Tank outlet(19) Flange(20) Reduced conversion(21) Air intake Tube(22) Measuring 1 inch(23)*

*Tank(24) Pump(25) Outlet Pump(26) Gas Turbine(27) Cold water inlet(28) Cooling water tank(29) Hot water outlet(30) Heat exchanger(31)] (Kohzadi et al., 2018)*

Analysing of two essential elements of the design, the heat exchanger and the centrifugal pump are critical. In this method, the heat exchanger technical and centrifugal pump specifications are shown in Table 1 and 2, which the entrance of seawater was at 293 k, 2 kg/s, and 4 bar pressure. Plus, inlet gas and outlet gas with 35 kg/s, the temperature of 825.15 k and atmospheric pressure were reported.

*Table 1: Technical specifications for tubular heat exchanger*

| The length of the fin (m) | Fin height (m) | Fin width (m) | The length of the heat exchanger shell (m) | The diameter of the heat exchanger shell (m) |
|---|---|---|---|---|
| 50 | 0.01 | 0.01 | 1.01 | 0.8 |
| Tube thickness(m) | Inner tube diameter (m) | Spiral diameter (m) | Spiral step (m) | Number of spiral rounds |
| 0.005 | 0.06 | 0.5 | 0.15 | 5 |

*Table 2: Technical Specifications of centrifugal pump*

| Pump speed (rpm) | Pump outlet pressure(bar) | Pump inlet pressure(bar) | Output width(mm) | Suction diameter(mm) | Diameter of the blade pump(mm) |
|---|---|---|---|---|---|
| 2900 | 4 | 1 | 14.63 | 94.9 | 212 |

As can be seen in Figure 4, the mesh sections in the pipe heat exchanger have four-sided meshes, and areas near the walls, fins, and inside the mesh tube are smaller, as well as the areas around the tubes and those parts, where the exhaust gases pass through the gas turbine exhaust; in these areas the mesh is also shrunk. In the heat exchanger part, there are generally three fluid and mesh regions, the first of them is the fluid region inside the tubes; the second is the tubes itself, which is solid; and the last part is the space around the tubes, or the space where the exhaust gas passes through the turbine exhaust. The steam output temperature is the variable and required parameter in this problem. As a result, the number of cells in the network is 602585.

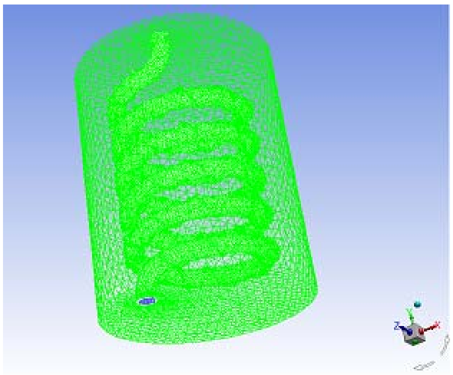


*Figure 4: Meshing of the heat exchanger*

## 3. Results and discussion

According to the Figure 5, temperature variations in different parts of pipe lengths are shown. These tangible temperature changes in the advancement of the proposed design are useful and important; because we only allow the drainage temperature of the exhaust gas turbine to be reduced to 423 K, since at lower temperatures, it approaches the dew point and the strangulation and the gas turbine exhaust wall will then be sucked down. The results of these temperature changes along the pipelines and transducer paths are given with Table in figure 5. As shown in Figures 6a, in this model, we will not be concerned about the issue of fluid doping and the unpleasant effects of corrosion and sedimentation, because the steam quality is equal to 0.999. According to Figure 6b, it has been determined that the discharge rate of 2 kg/s is removed from the converter and moves towards the piston cylinder chamber.

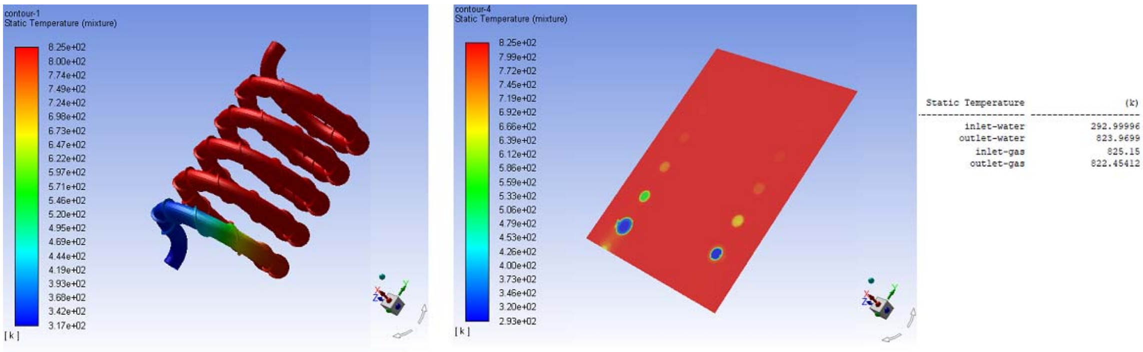



*Figure 5: Temperature changes in the heat exchanger*

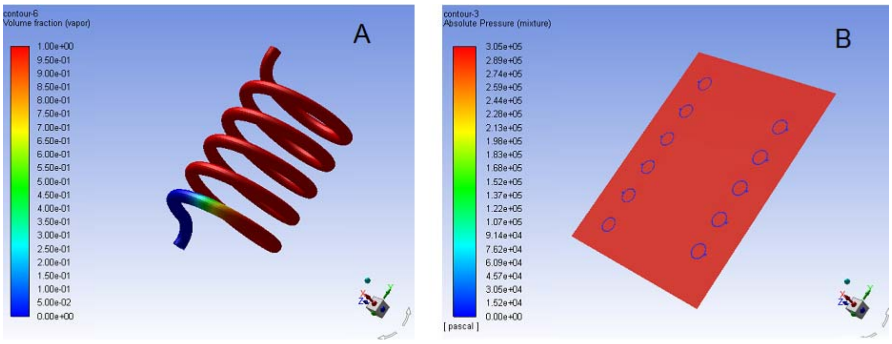



*Figure 6: Variation of the quality of steam output (A) and pressure (B)*

Regarding the mesh section of the centrifugal pump, the meshes in this simulation are rectangular cubes that are in fact organized. In such a way that is laminated and tiled near the walls of the blade of the pump, and is considered coarse in other parts of the blade. Figures 7A and 7B are illustrates the buckling scheme of the pump's blade. Pump output pressure as a variable and the answer to the problem. As a result, the number of cells in the network is 16500.

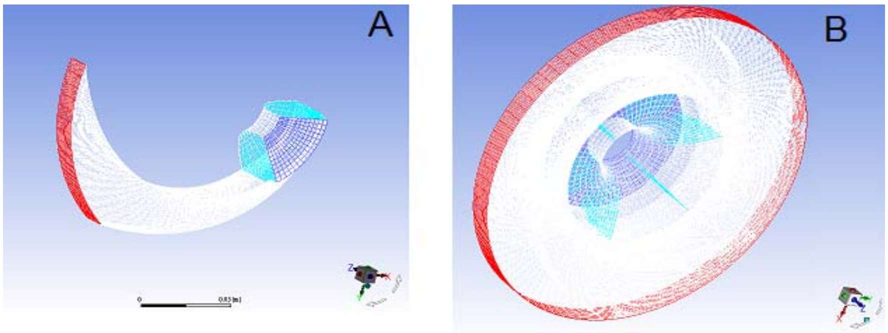



*Figure 7: The layer of the pump blade (A) and Assembled of the pump blade (B)*

According to the information obtained from the results of this arrangement in Figures 8 and 9, it can be found that by increasing the pressure in the suction centred of the centrifugal pump and without changing its physical structure, it can be increased by a pressure of about 3 times in the pump outlet required. This is same as improving the performance of a pump. To calculate the cost of water desalination. As we have now discussed, several methods have been suggested to prevent cavitation. Therefore, in the current design, prevention of this phenomenon with a high percentage of execution and warning is probable and will play a perfect performance in the industry. Accordingly, the advantages of the advanced model could be summarized as follows: energy saving occurred, production of water vapour pressure under single phase and multiple phase conditions released [19, 20], the pump performance without the need for physical changes increased, the probability of cavitation occurrence and suction loss reduced.

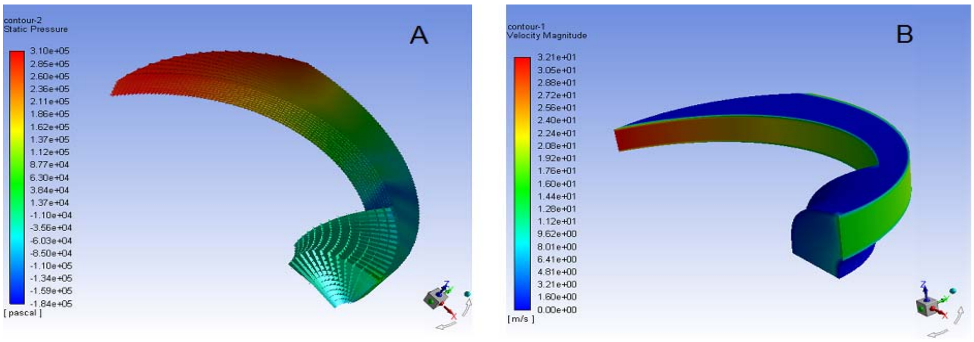



*Figure 8: Primary model; pressure changes in the pump wheel (A) and velocity of the centrifugal pump (B)*

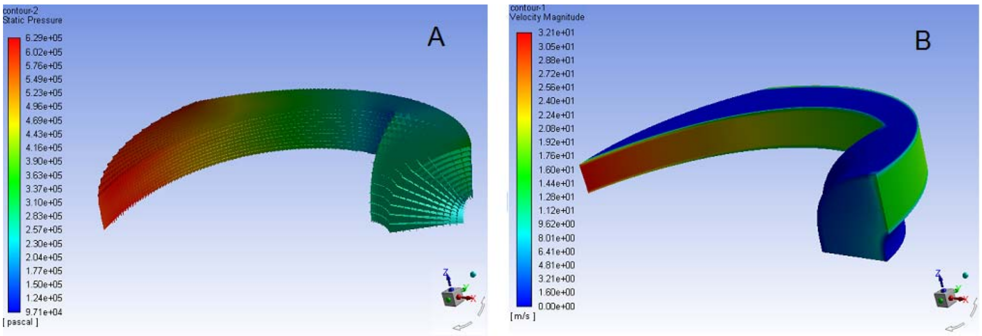



*Figure 9: Proposed model; pressure changing in the pump wheel (A) and velocity of the centrifugal pump (B)*

## 4. Conclusions

In this research, as the results have shown, it is essential to note that with increasing the pressure of the suction pump, a pressure equivalent to 3 bar, then in the drift of the pump, it had accrued the same increase in pressure without changing the pump's physical structure and efficiency. By increasing the pressure in the thrust, cavitation phenomena restricted and, this increase in pressure also improves the pump performance and prevents cavitation from occurring. It is noteworthy that energy losses in the industrial sector provide all the energy needed for the heat source of the heat exchanger used in this design. In the end, due to the efficiency and capability requirement for implementation of this study, it should be consistent and appropriate to the world of today's industry.

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