



Investigation of Parameters Affecting Axial Load in an End Suction Centrifugal Pump by Numerical Analysis

H. Pehlivan and Z. Parlak[†]

Department of Mechanical Engineering, Sakarya University, Serdivan, Sakarya, 54187, Turkey

[†]Corresponding Author Email: zparlak@sakarya.edu.tr

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ABSTRACT

The total force produced in the axial direction on a pump is called axial load and is caused by the pressure difference between the front and rear of the impeller and the hydrostatic force in the suction direction. In a centrifugal pump, 3D computer-aided analysis programs are used to design and reduce R&D and manufacturing costs. In this study, parameters affecting axial load of the centrifugal pump with a single suction and closed impeller were investigated by using the Computational Fluid Dynamics (CFD) method. In this context, the flow rate and the some physical properties such as the back gap of the impeller, wear ring and balancing holes, of the centrifugal pump were investigated to determine how much affected the axial load. The results showed that the wear ring and the balancing holes give rise to effective results on the axial load, while the back gap of the impeller does not affect the large extent. With the design changes made with these parameterizations, the axial force was reduced by up to 60%, whereas the efficiency was decreased by 5%. The loss of efficiency due to this decrease in axial force is negligible. However, higher efficiency values were also found at a different point from the working point where the axial load is lowest.

Keywords: Centrifugal pump; Thrust load; Axial load; Balancing hole; Wear ring; Computational fluid dynamics.

NOMENCLATURE

AL	axial load	ψ	head coefficient
d	diameter	η	efficiency
F	force	ρ	density
g	acceleration of gravity	ψ	head coefficient
H	head		
HP	hydraulic power		
\bar{k}	average rotation factor	subscript	
MFR	mass flow rate	Ds=sp	on front shroud
\dot{n}	revolutions per minute	h	hydraulic
p	static pressure	Hy	axial
r	radius	La	impeller
SP	shaft power	Ts=D	on rear shroud
u	velocity	2	outlet of impeller

1. INTRODUCTION

Pumps are a kind of general machinery with varied applications. According to statistics, energy consumption of the pumps accounts for nearly 22% of the world's energy used by electric motors (Wang *et al.* 2017), so pumps have huge energy consumption and great energy saving potential. Also, axial piston pumps are widely used to provide

hydraulic power for the hydraulic circuit because of their high pressure, high efficiency, high power density and convenient flow regulation. They have a wide range of applications in areas such as petroleum, chemical, and aerospace industries. In reality, the centrifugal pumps with a variable-pitch inducer usually work under various operational conditions, especially at various rotational speeds. In the case of high rotational speeds, the use of the

impeller with a variable-pitch inducer upstream is an effective method to improve the weakened anti-cavitation performance of the pumps. The axial piston pump comprises several pistons mounted within the cylinder bores at equal angular intervals around the centerline of the cylinder. The cylinder is pushed towards the fixed valve plate by a compressed cylinder spring. On the other hand, the compressed spring force is transferred to the spherical cup through several pins so that a hold-down force can be applied on the retainer. By the hold down mechanism, the reasonable contact between the slipper and swash plate can be realized. The shaft and cylinder are coupled by means of a spline mechanism and they are driven around the axis of the shaft. During rotation, each piston periodically reciprocates within its corresponding cylinder bore due to the inclined swash plate and the retainer. There are three main lubricating interfaces in the axial piston pump: slipper/swash plate interface, piston/cylinder interface and cylinder/valve plate interface (Zhang *et al.* 2017a, Guo *et al.* 2017, Zhang *et al.* 2017b). In the past, several experimental and numerical studies have been performed, to investigate the operation behavior of centrifugal pumps under various operation conditions. Operation of centrifugal pumps under single-phase flow conditions has been investigated, for example, regarding effects like flow induced pressure pulsations and resulting vibration and noise.

A review of the literature shows that few studies have been performed on the experimentally. Dick *et al.* (2001) compared the results between the experimental results and the different CFD solution methods. Non-temporal solutions were risking to come out of real results. Chalhoun *et al.* (2016) investigated the transient characteristics of a centrifugal pump in the initial period. They noted that the pump characteristic curve shows good agreement with the numerical and experimental results. Vasant Godbole *et al.* (2012) conducted a study of balancing fins in their work and commented on experimental measurements. They determined the most effective parameters as the rear blade height, the number of blades and the distance between the blade and the body. It revealed that the axial thrust value changes according to the rear blade height and the change of the gap distance. Zhao *et al.* (2012) studied the method of balancing the axial force with balance hole based on the combination of CFD and experimental research. The relationship between the radial position of balance hole and the effect of axial force and external characteristics of the pump is in-depth studied. Chen *et al.* (2017) are put forward the design idea and method for the spatial geometry of twisted gap drainage blade, which develop the design method of gap drainage blade from 2-D to 3-D twisted blade and will achieve an advancement for introducing gap drainage technology into practice. Yan *et al.* (2017) presented a systematic comparison of the fluid flow and heat transfer characteristics between standard and cross-drilled ventilated brake discs incorporating pin-fins. Also, experiments and numerical simulations were conducted under steady-state conditions. Schäfer *et al.* (2017) performed an

experimental study on the effects of additional gas entrainment in centrifugal pumps designed for conveying liquid phases only. The pump performance has been evaluated for several gas entrainment conditions, and for various operational settings of the pump, such as its alignment and the rotational speed of the impeller. Kalinichenko and Suprun (2012) proposed the development of a nodal point for axial balancing of a centrifugal pump rotor. They noted that the change in the pump rotor reduced volume and mechanical energy loss and increased efficiency. Zhou *et al.* (2013) used the experimental and numerical methods to analyze the oblique impeller trimming method and studied the impact of different impeller rear shroud radii. The axial force and pump performance captured by two methods were compared and discussed. They employed the numerical simulations and experiments to study the axial force and pump efficiency in a deep-well centrifugal pump. Five impellers' rear shroud Radius were trimmed to different values, and then assembled with the same diffuser to carry out the test and simulations. Zhou *et al.* (2014) also investigated the axial force in new type DCP, and explore the accurate method for axial force calculation. Consequently, with the integration of theoretical analysis, numerical simulation and experimental results, three different calculation formulas of axial force were compared and analyzed, and a detailed numerical simulation and an axial force experiment were carried out to verify these results. At rated flow point, the deviation of the axial force obtained by numerical simulation with the experimental value is approximately 3.8%. This is a unique advantage of numerical simulation, in which one could visualize the flow field and estimate the pump performance (including the axial force) during the design stage without building the physical prototype

With the development of computational fluid dynamics (CFD) models, more studies are using numerical simulations to study the flow fields in centrifugal pumps to understand the which parameters affect design parameters. Grapsas *et al.* (2008) performed a CFD analysis and compared experimental results of the turbulent flow through the test impeller, and found a adaptation in a laboratory pump propeller with the relevant measurements. They also examined the effects of some blade design parameters such as blade length, inlet height and front edge slope on the performance and efficiency of the propeller. Baun and Flack (1999) worked on calibrating the load cell on a bearing in order to measure the hydraulic forces. On this research pump, axial and radial force measurements were made to demonstrate the potential of the magnetic bed load cells and these measurements are compared with the values in the literature. Budea (2015) emphasized that the calculation of axial and radial hydraulic forces is important for the design of the impeller in their work. In multi-stage pumps, the axial force was increased in proportion to the number of stages. The front and rear ring diameters have to be of the same diameter and the balancing blades and rear opening are important. Wang and Qian (2017) investigated the effects of the silt concentration and silt size on pump performance by devising a new experiment for

a double-suction centrifugal pump. The experimental results show that the pump head and the shaft power are lower compared with that of pure water condition due to the presence of silt, and decrease as the silt concentration and silt particle size increase. [Zhao et al. \(2013\)](#) studied the effects of axial and radial forces on the wear ring. According to the results obtained, the wear ring effect on the axial force is great.

[Salvadori et al. \(2007\)](#) used a CFD method, to design a bearing by working on a horizontal multi-stage pump. They argued that the beds of a multi-stage pump must be designed appropriately. The change of the axial force is expressed by the mass flow rate. The wear ring determined the operating conditions of the pump by emphasizing the importance of the bleed channels. [Cao Lei et al. \(2015\)](#) worked on the gap between the rotating impeller and the fixed body cover in the centrifugal pump. They studied the effect of the flow on the main flow and how the hydraulic, volumetric and mechanical activities of the axial gap changes. Axial clearance distances of the closed impeller are effective in pump performance. Effectiveness is reduced in the front opening and the shaft power has remained the same in the front opening. The speed distribution changed with axial gap. As the front opening increased, the flow interfered more with the main flow and the back flow at the inlet of the impeller was increasing. The volumetric effect is sensitive to the axial gap, and the mechanical effect is slightly altered by both the flow rate and the axial gap. [Ibrahim \(2016\)](#) focused on balancing holes, one of the parameters affecting axial load. He has developed a formula to predict the most appropriate location of the hole sites, making it the cheapest and easiest way of balancing. The optimal hole diameter causes a efficiency reduction of 3%. [Spence and Amaral-Teixeria \(2009\)](#) worked a CFD analysis to investigate the vibrations in a double-inlet pump. When investigating the effect of geometry on vibrations, according to the order of importance, they pointed out that the blade plane, the inlet gaps, the side gap and the rubber gap are in order of importance. [Della Gatta et al. \(2006\)](#) have conducted a multi-stage centrifugal pump with the CFD method. Correct design and dimensioning of axial bearings has been the main objective in order to balance the hydraulic force. CFD analyzes have shown that the side chambers have a large effect on the axial thrust and that the pressure distribution is highly influenced by the leaky mass flow and local rotation speed in the cavity. [Han et al. \(2012\)](#) studied the effect of axial flow on CFD analysis of solid-liquid flow with two shells in a screwed centrifugal pump.

Purpose of this study is to monitored how some design parameters affect the axial load of a centrifugal pump. Methodology in the study is that the effects of some design parameters on the axial load of the centrifugal pump are examined by Computational Fluid Dynamics (CFD) verified experimentally while the general design of the centrifugal pump including impeller is kept constant. A limited number of publications have been found in the literature about axial load of the centrifugal

pumps to be experimentally and CFD analysis. Originality of the study is that the effects of six parameters, which are the distance between the back of impeller and the body cover, length of external wear ring, balancing hole, flow rate, wear ring cavity, internal wear ring, are examined affecting the axial load in a single study. In addition, the effect of the internal wear ring was examined for the first time in the literature. It was also observed how the change of the discharge head for the same pump while examining the axial load variation.

2. NUMERICAL MODEL AND METHODOLOGY

During operation and working of centrifugal pumps, the kinetic energy of liquid is converted into pressure energy. This high pressure liquid is continuously all over the circumference of the impeller and also gets entrapped inside the clearances between impeller and casing. The high pressured liquid trapped between impeller and casing which exerts pressure on the outlet passages and shrouds of the impeller resulting in generation of forces; radial force which generated in lateral direction is due to dissimilar pressure generation in volute and axial force which generated in longitudinal direction is on account of different areas of impeller exposed to trapped pressurized liquid.

Two different analytical methods are used in the calculation of axial force, Gülich and HI. [Gülich \(2001\)](#) discussed the prevention of cavitation damage and concludes by presenting hydraulic criteria that can be used for the selection of suction impellers. It is a study to summarize some hydraulic criteria in pump selection to prevent cavitation damage. Gülich method, the value of the static pressures at the impeller outlet must be known in order to calculate the axial force. The following equations are used to calculate the axial forces.

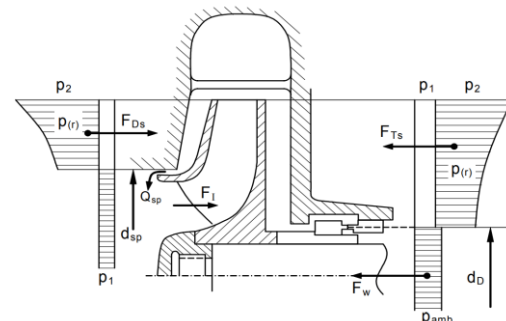


Fig. 1. Pressure distribution and axial forces in pump (Gülich, 2008).

$$\Delta p_{La} = \rho g H \left(1 - \frac{\Psi}{4\eta_h} \right) \frac{\eta_h \cdot La}{\eta_h} \quad (1)$$

$$\Psi = \frac{2 g H}{u_2^2}; u_2 = \frac{\pi d_2 \dot{n}}{60} \quad (2)$$

$$F_{Hy} = 2\pi \int p \cdot r \cdot dr \quad (3)$$

η_h hydraulic efficiency of the pump, $\eta_{h, La}$ hydraulic efficiency of impeller, ΔP_{La} static pressure rise in impeller.

The integration is done from r_{sp} ($d_{sp}/2$ in Fig. 1) or r_D ($d_D/2$ in Fig. 1) to r_2 (radius of impeller outlet); in the general case it is performed separately for rear and front shroud.

Axial force on impeller is;

$$F_{Hy} = F_{Ts} - F_{Ds} \quad (4)$$

$$F_{Hy} = \frac{\pi}{4} (d_{sp}^2 - d_D^2) \left\{ \Delta p_{La} - \frac{\rho}{2} \bar{k}^2 u_2^2 \left(1 - \frac{d_{sp}^2 + d_D^2}{2d_2^2} \right) \right\} \quad (5)$$

where F_{Ds} is axial force on front shroud, F_{Ts} is axial force on rear shroud and \bar{k} is calculation of average rotation factor from the measurement of a pressure difference in the impeller sidewall gaps.

$$\bar{k} = \sqrt{\frac{2(p_2 - p)}{\rho u_2^2 \left(1 - \frac{r^2}{r_2^2} \right)}} \quad (6)$$

where p_2 is pressure in outlet of impeller.

Force acting on front shroud due to liquid of delivery pressure entrapped between pump casing and front shroud. Force acting on back hub is due to liquid of delivery pressure entrapped between casing cover and back hub. Force acting in the direction of the liquid flow is due to its momentum change.

Engineers primarily need flow rate, head and revolutions per minute (rpm) to design the centrifugal pump. The pump type is selected according to the specific speed determined accordingly. Other parameters (impeller diameter, number of blades, angle of blade, etc.) are determined by empirical methods. Some choices are fixed with experimental experiences and are selected in this frame (Kalinichenko and Suprun, 2012).

Pump designers cannot see how the stream in the pump flows a path. However, with the help of the Computational fluid dynamics (CFD) method, it can improve its design by seeing and interpreting even most detail points of the flow. Thus, it is able to design a more advanced pump by increasing efficiency in the product to be produced. In terms of R & D costs, the contribution of computer-assisted numerical solutions to production is also undeniable.

There are many methods of balancing axial thrust in the pumps like balancing hole, back vanes, wear ring, double suction impeller and even number of impellers of multistage pump. CFD can also provide great convenience to the designers about balancing axial load.

2.1. CFD Models for Calculating Axial Load

CFD analysis is being increasingly applied in the

design of centrifugal pumps. With the aid of the CFD approach, the complex internal flows in water pump impellers, which are not fully understood yet, can be well predicted, to speed up the pump design procedure. Thus, CFD is any important tool for pump designers. The use of CFD tools in turbo machinery industry is quite common today. Recent advances in computing power, together with powerful graphics and interactive 3D manipulation of models have made the process of creating a CFD model and analyzing results much less labour intensive, reducing time and, hence, cost. Advanced solvers contain algorithms which enable robust solutions of the flow field in a reasonable time. As a result of these factors, Computational Fluid Dynamics is now an established industrial design tool, helping to reduce design time scales and improve processes throughout the engineering world.

ANSYS Fluent tool was used to examine the axial load in centrifugal pumps. The pump was designed for 2900 rpm, 30 meters of head, 36 tons/hour of flow rate, 174 mm of impeller diameter. The impellers and volute models obtained were modeled in Vista CPD tool in ANSYS 16.2. Some improvement works were carried out for the impeller and volute after that assembly were provided.

In the CFD analyzes, the flow rate of the design point and 0 Pa effective pressure were defined as boundary conditions at the inlet and outlet of the pump, respectively. Also, two different domains were defined, namely the volume swept by the rotating impeller and the remaining stationary volume. Interfaces boundary conditions were defined in the surfaces which are contact each other of the domains. Also, all surfaces except the inlet and outlet were defined as the wall boundary condition. The pressure in outlet and efficiency of pump were calculated in CFD-Post as two output parameters. The assembled state of the pump in the study is shown in Fig. 2.



Fig. 2. Assembly view of the centrifugal pump.

Computation Grid

The quality of the mesh plays a significant role in the

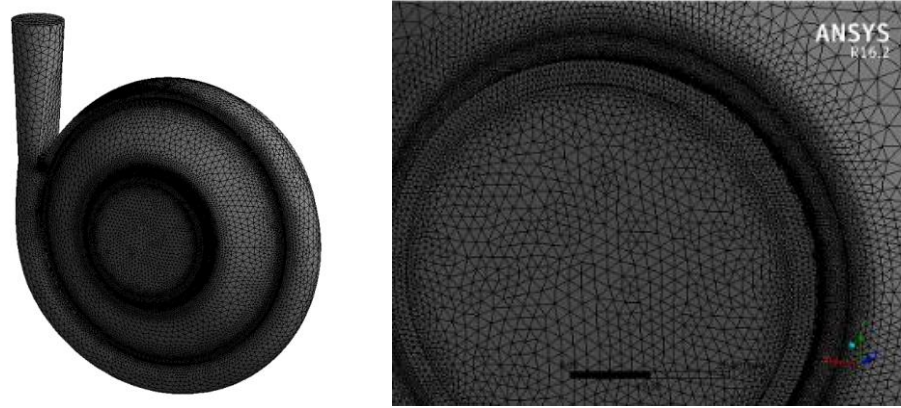


Fig. 3. The mesh structure in flow domain.

accuracy and stability of the CFD analyses. Regardless of the type of mesh used in a flow domain, checking the quality of the mesh is essential. In this study, ANSYS TurboGrid was used to generate grid for the flow volume of the impeller domain and ANSYS Meshing was used for volute. These two grids were merged and analyzes were carried out. A fine mesh is required near the wall to resolve the laminar sub-layer. Therefore, a rectangular mesh with grid adaption for $y^+ < 2$ at an adjacent wall region is used to resolve laminar sub-layer. In the study, y^+ was specified as 1 and first layer thicknesses of the grid near the wall was calculated about 0.00286 mm. The mesh independence was examined to see the adequacy of the mesh used. The mesh structure in flow domain is presented in Fig. 3.

The mesh independence in the flow domain must be determined before starting the CFD studies. In Fig. 4, the number of elements on the mesh is increased and the number of elements where the stabilization result is obtained is determined by looking at the change over the axial force.

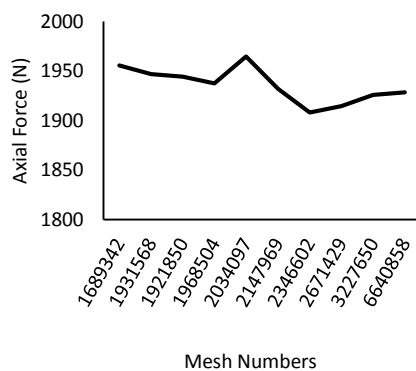


Fig. 4. Mesh independency.

In this study, the number of expected mesh elements, which are sufficient to start from the low solution mesh numbers, were determined. In subsequent works, approximately 3250000 elements were used in mesh structure. It is determined that this value is sufficient for our analysis processes.

Solution Method

The geometry and mesh structure were transferred to the Fluent tool. In this step, $k-\omega$ turbulence model was selected due to faster convergence and good agreement with the experimental results. Two domains which are rotating and stationary, zero input pressure in inlet and the value of flow rate in outlet were determined and SIMPLER solution method was used in this study. The discretization scheme for the convection terms of each governing equation were specified as second order for pressure, second order upwind for momentum and turbulent kinetic energy.

Continuity, momentum and turbulence equations are solved for all the elements in the mesh. The solution process ends when 10^{-3} of convergence criterion is reached, because the comparison with experimental results showed that this convergence was sufficient. In addition, since the convergence of the continuity equation was relatively slow compared to the others, when the continuity equation reached 10^{-3} , the momentum and turbulence equations reached a much lower convergence value. Thus, a significant saving was achieved from the calculation times considering the all of analysis.

2.2. Pump Manufacture and Test Setup

The investment casting method was used for the manufacturing of the prototype impeller and the volute was machined in the CNC machining center (Fig. 5).

In the design of the test setup, the tank volume, pipe diameters, valves and measuring devices were specified by taking into account the reference flow rate and head, in order to avoid any problem in flow. An electronic drive and data acquisition were selected depending on the calculation result of the estimated power consumption of the pump. Experiments were carried out by the pump and test setup, and the results obtained from these experiments and analyzes were compared. Figure 5 shows the pictures of the impeller and volute used with the test setup. Thanks to the interface design developed, the data was saved automatically and the curves were obtained easily (Fig. 6).



Fig. 5. Impeller and volute manufactured and tested.



Fig. 6. Pump test setup.

Experiments were carried out by the pump and test setup, and the results obtained from these experiments and analyzes were compared.

3. RESULTS AND DISCUSSIONS

It is important to balance the axial load, which is one of the hydraulic forces that result from the operation of the end suction centrifugal pump. The parameters affecting the axial load are important when designing in terms of not damaging the shaft and bearings. In this study, the parameters affecting the axial load were investigated step by step. While balancing holes are used in closed impellers to compensate the axial load, balancing blades are usually used in semi-open impellers. The design parameters affecting the axial load and their upper and lower design limits determined for the study are given in Table 1.

Table 1 Design parameters and upper and lower limits

Parameters	Lower-Upper Limits
The distance between the back of impeller and the body cover (mm)	5 – 40
Length of external wear ring (mm)	20 – 30
Balancing hole (mm)	3 – 10
Flow rate (kg/s)	6.67 – 10.83
Wear ring cavity (mm)	0.25 – 2.5
Internal Wear Ring (mm)	20 – 40

The comparison of the results of the experiment and analysis for the same pump is given in Fig. 7. A good agreement between 2.3% and 11.1% was observed between the analysis and the experiment, however it is 3.6% at the 10 kg/s of flow rate at which the analyzes are performed mostly. Analyzes the following with this validated model were carried out.

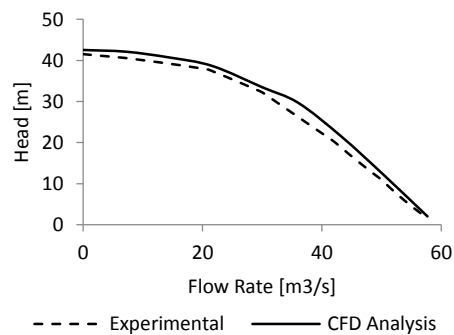


Fig. 7. Comparison of the experiment and analysis.

3.1. Analysis According to the Distance Between the Back of Impeller and the Body Cover

The distance between the back of the impeller and the the body cover, as can be seen Fig. 8, was defined as the back length. As this distance increases, the

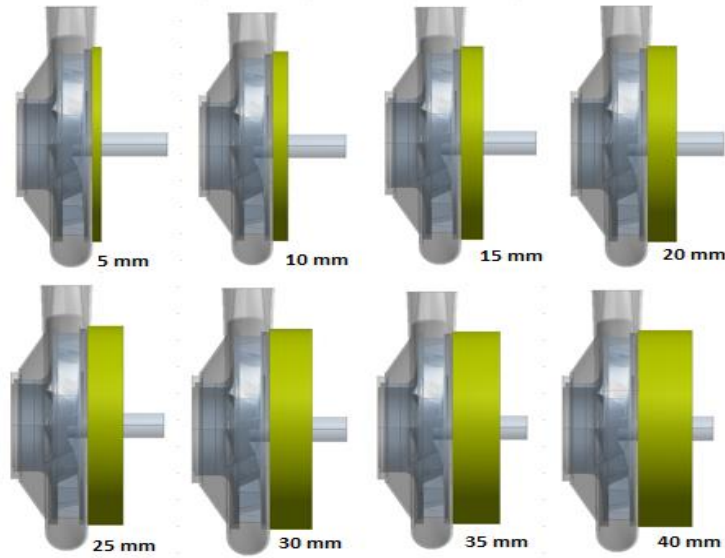


Fig. 8. Distance between the back of impeller and the body cover.

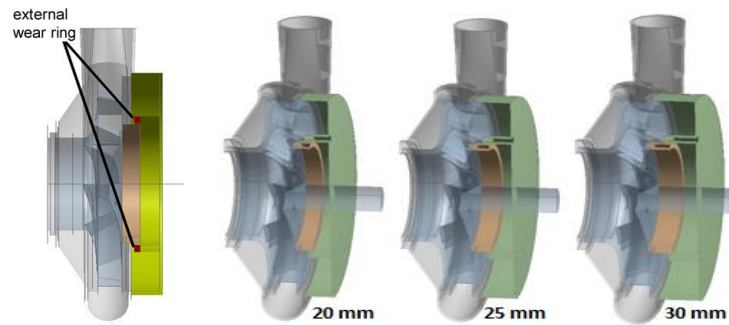


Fig. 9. Different length of external wear ring.

axial load, the head and efficiency values were obtained in the analysis results.

In this analysis, the external wear ring, balancing hole or balancing impeller were not used behind the impeller. The analysis results are given Table 2.

Table 2 Effect of distance between the back of impeller and body cover

Dis. (mm)	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
5	10	1932.1	297141	2985.8	5820.4	51.3
10	10	1946.2	297321	2978.6	5782.1	51.5
15	10	1980.1	296469	2935.7	5728.4	51.2
20	10	2013.3	297304	2950	5742.1	51.4
25	10	1996.7	298310	2958.9	5701.7	51.9
30	10	2041.1	298040	2978.9	5756	51.8
35	10	2010.4	298305	2988.4	5756.2	51.9
40	10	2020.2	297273	2971.1	5766	51.5

According to the results; small differences arise of the values of the efficiency and the pressure at outlet. At the axial force value, there is a slight increase as the distance increases. It can be said from these results that the distance has not a serious effect on the axial load.

The pump design should be set a distance to examine the next step in the use of the wear ring. According to the selected distance, the wear ring was designed and analyzed.

3.2. Analysis According to Length of External Wear Ring

The use of wear rings is very important in order to protect seep flows. This part, which is placed between the ring on the body cover and the impeller back bracelet, separates the high pressure and low pressure areas of the pump and must be replaced when it is depreciation. If the wear ring had not been used, the impeller and body would have worked very close to each other, and later replacement would be costly. Therefore, the use of the wear ring is also important in terms of cost.

In the analyses carried out, the wear ring which wraps from outside the impeller bracelet was used

(Fig. 9) and was placed between the body ring and the impeller back bracelet and examined under the name of external wear ring.

In this analysis, three different length of external wear ring were researched and balancing hole or balancing blade was not used firstly. The cavity of wear ring is set 1.5 mm. The distance between the impeller and the body cover, which is considered suitable for making inspections, is accepted as 20 mm. In Table 3, the effect of different length of external wear ring without balancing hole and 112.6 mm of inner diameter is summarized.

Table 3. Effect of length of external wear ring

Length (mm)	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
20	10	2016.7	299208	2997.5	5799.6	51.7
25	10	2041.7	290172	2986.9	5736.3	50.3
30	10	2086.1	292545	2930.7	5765.9	50.8

According to the obtained results, there was no significant change in the efficiency and the pressure values, while the change in axial force showed a slight increase depending on the length of external wear ring. The use of the wear ring in this way has no significant effect on the axial load.

The analyses were made at various diameters of the external wear ring. The cavity of wear ring and length of external wear ring were set 1.5 mm and 20 mm, respectively. The results of effect of the diameter can be seen in Table 4.

Table 4 Effect of inner diameter of the external wear ring

Inner diameter of the wear ring	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
without the wear ring	10	2013.3	297304	2950.0	5742.1	51.4
92.6 mm	10	1232.7	283081	2835.9	5831.6	48.6
102.6 mm	10	1597.8	308686	2992.4	5760.5	51.9
112.6 mm	10	2016.7	299208	2997.5	5799.6	51.7
126 mm	10	2478.0	300564	3011.1	5789.8	51.3
132.6 mm	10	2969.5	309475	2968.3	5747.5	51.7

The numerical results showed that axial force appears to be highly influenced by the diameter. Therefore, the diameter of the wear ring should be carefully selected to compensate for the axial load on the impeller.

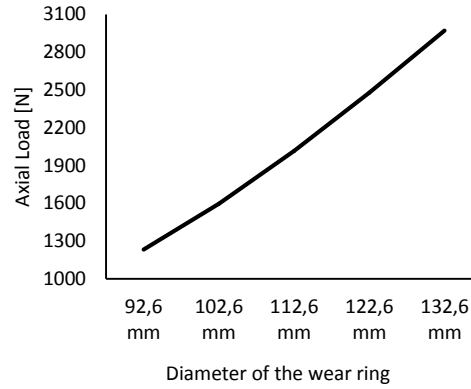


Fig. 10. Effect of change in diameter of the external wear ring on axial load.

In the absence of the balancing holes, the phenomenon is only related to the amount of fluid rotated with impeller in the backside of the hub, since there is no a fluid transition from the rear of the impeller to front of the impeller. As the amount of the fluid increases rotated by impeller, the static pressure rises and the momentum force, which is generated by the rotation effect, increases depending on proportion to the square of the radius, thus it affects the axial load (Fig. 10).

3.3. Analysis According to Effect of Balancing Holes

The influence of the use of balancing holes (Fig. 11) in closed impellers on the axial load was investigated. As a result of using the wear ring, it is necessary to use the holes to compensate the axial load. The number, diameter and position of the balancing holes are important. Six balancing holes, one for each blade passage, were placed in the blade entrances within the suction mouth of the pump. The effects 3 mm to 10 mm of hole diameter on the axial loads were investigated and the results, which are obtained for 20 mm length, 102.6 mm diameter and 1.5 mm cavity of the external wear ring, are given Table 5.

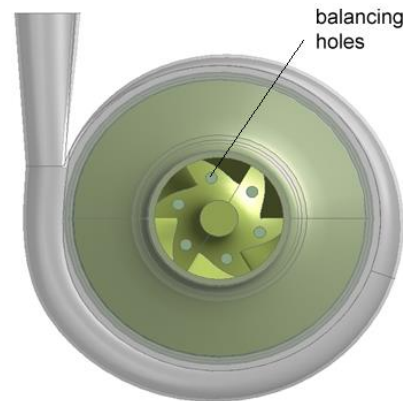


Fig. 11. Balancing holes.

The increase of the hole diameter has decreased the axial load notably. However, a significant decrease in the efficiency has been observed.

Table 5 Effect of balancing holes

Hole diameter (mm)	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
without holes and wear ring	10	2013.3	296604	2950.0	5742.1	51.37
3	10	1492.1	292321	2930.1	5950.7	49.24
4	10	1364.4	283169	2836.8	6083.2	46.63
5	10	1186.4	276270	2767.7	6221.7	44.48
6	10	867.1	272641	2713.7	6262.2	43.33
7	10	636.3	269899	2663.9	6298.9	42.29
7.5	10	701.1	263475	2639.5	6310.8	41.82
8	10	688.4	237507	2379.4	6430.4	37.00
9	10	522.8	237774	2382.1	6443.2	36.97
10	10	407.9	231692	2321.1	6485.6	35.79

Table 6. Effect of length of external wear ring with balancing hole

Length (mm)	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
15	10	1159.9	272173	2726.6	6154.8	44.3
20	10	1186.4	276270	2767.7	6221.7	44.48
25	10	1235.9	279096	2796.1	6214.2	45
30	10	1144.1	266292	2768.3	6261,8	44.21
35	10	1265.7	278154	2796.6	6259.1	44.68
40	10	1317.2	278427	2799.3	6253.4	44.76

Table 7 Effect of flow rate

MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η . (%)
6.667	1175.93	354999	2371.05	5188.54	45.70
7.5	1162.15	341057	2562.54	5424.14	47.24
8.333	1152.77	325749	2719.36	5665.82	48.00
9.167	1161.28	304582	2797.14	5939.07	47.10
10	1186.4	276270	2767.7	6221.7	44.48
10.833	1126.26	263773	2861.81	6349.56	45.07

this aspect of the result, an optimum balancing hole diameter with high efficiency and low axial load should be selected. Because, after a certain hole diameter, there is a significant decrease in efficiency. For these reasons, a hole diameter of 7 mm was determined for our pump. In the light of the result, an optimum hole diameter with high efficiency and low axial load should be specified.

The next step was to investigate the effect of length of external wear ring with 5 mm balancing hole, 1.5 mm wear ring cavity and 102.6 mm wear ring diameter and the results are shown Table 6.

There is no significant change in the axial load due to the length, as in the analysis without wear ring and balancing hole. It can be said that the length does not have too much effect, however the wear ring diameter and the balancing holes have a more significant effect on the axial load.

3.4. Analysis According to Effect of flow Rate

After all of these studies, the changes of the flow rate of the pump were examined. Analyzes were carried out for flow rate between 24 to 39 tons/hour and under the conditions, which are 20 mm length, 102.6 mm diameter and 1.5 mm cavity of the external wear ring and 5 mm balancing hole. The results are displayed in Table 7.

There is no big difference between the results aspect of axial load for mass flow rate. However, in the pump designs, the design point of the pump was specified at 36 tons/h (10 kg/s) and the discharge head at 30 m, due to the modifications made during assembly and used the wear ring and holes, the working point of the pump changed as can be seen Table 7. When look at the highest efficiency value that the operating flow rate of the pump is 30 tons/h

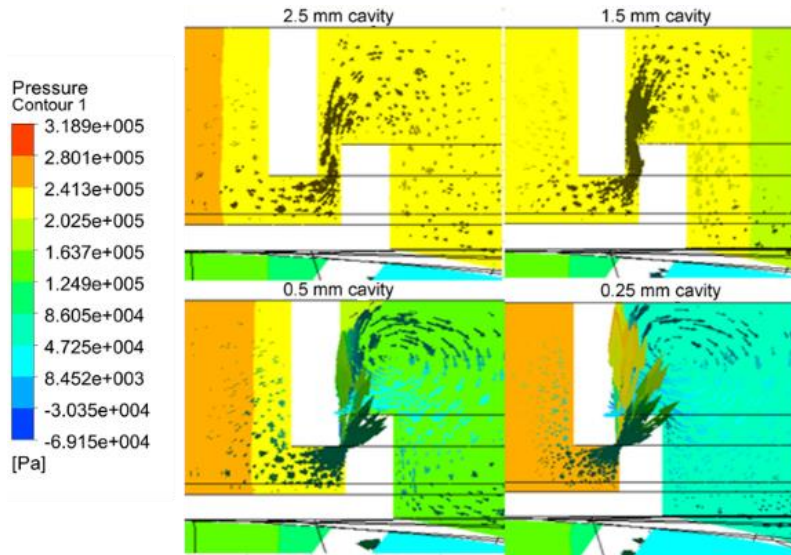


Fig. 12. Cavity between the wear ring and the impeller bracelet.

(8.333 kg/s) and the pump head increases according to the beginning of the design. The pump should be operated with the most suitable flow rate at the radial loads on the pump.

3.5. Analysis According to Effect of Wear Ring Cavity

In studies performed so far, the wear ring cavity was determined to be 1.5 mm. Generally, the cavity between the wear ring and the impeller bracelet ranges from 0.25 mm to 25 mm in the literature. If this cavity is less than 0.5 mm, the pump elements may be damaged. If it is more than 2.5 mm, leakage flow may increase and volumetric efficiency may decrease. In Fig. 12, the effect of wear ring cavity is presented.

CFD analyzes were performed to investigate the effect of the wear ring cavity. The effect of wear ring cavity for 8.33 kg/s mass flow rate, 5 mm balancing hole, 20 mm length and 102.6 mm diameter of external wear ring were shown in the Table 8.

Table 8 Effect of wear ring cavity

Cavity (mm)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
2.5	1192.12	323674	2702.04	5662.62	47.72
1.5	1152.77	325749	2719.36	5665.82	48.00
0.5	842.023	330835	2761.82	5582.92	49.47
0.25	418.345	338288	2824.04	5492.13	51.42

When the results are examined, there is a decrease exponentially in the axial load value as the wear ring cavity decreases (Fig. 13). It can be concluded that the reduction of the wear ring cavity is one of the most effective parameters to reduce the axial force. Also, smaller cavity values have a positive effect on

efficiency. However, the wear ring cavity should not be too small, because it may lead to damage caused by friction. The wear ring cavity was preferred 0.7 mm taking into the considerations. As can be seen from pressure distribution at these four different value of the cavity in Figure 14, the pressure value at the inlet of the impeller increases at smaller cavity values and the ΔP_{La} value decreases according to Eq. 1 and consequently the axial force decreases according to Eq. 5.

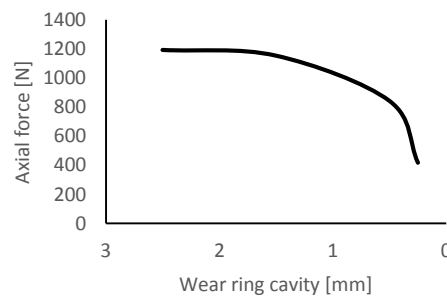


Fig. 13. Effect of the wear ring cavity on axial force.

3.6. Analysis According to Internal Wear Ring

The inner diameter of the impeller bracelet has been 89.6 mm and the wall thickness has been 5 mm in the analyses carried out so far. The internal wear ring was designed to be inside of the bracelet on the back surface of the impeller (Fig. 15). The effect of length of internal wear ring, which is range of 20-40 mm, on the axial load was investigated. The results of the CFD analysis on the 5 mm balancing hole diameter, 0.7 mm wear ring cavity and 30 tons/hour are shown in the Table 9.

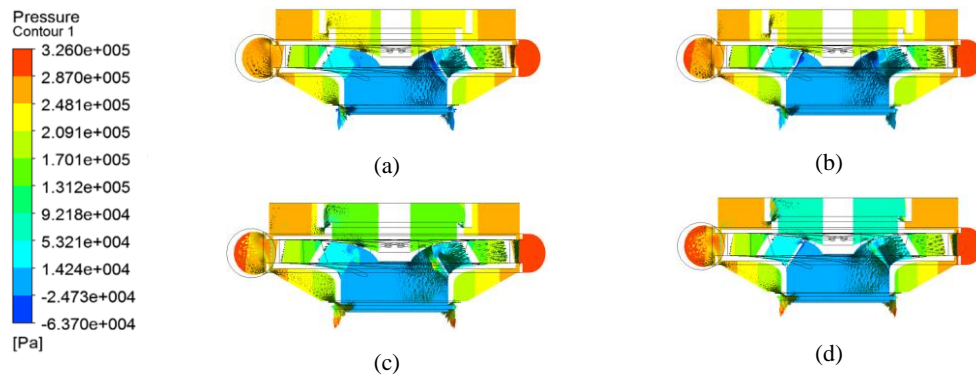


Fig. 14. The pressure contours in the wear ring cavity with different gap (a) 2.5 mm (b) 1.5 mm (c) 0.5 mm (d) 0.25 mm

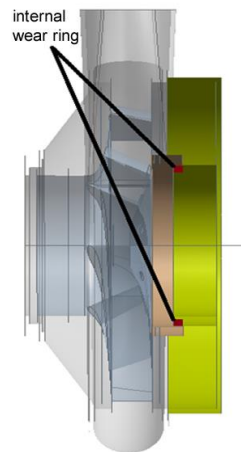


Fig. 15. Using internal wear ring.

Table 9 Effect of Internal Wear Ring

Length (mm)	MFR (kg/s)	AL (N)	p (Pa)	HP (W)	SP (W)	η (%)
20	8.333	729.9	335262	2798.7	5626.5	49.74
30	8.333	759.8	337173	2814.7	5642.9	49.88
35	8.333	761.6	339999	2838.3	5629.5	50.41
40	8.333	766.1	338411	2825.1	5638.6	50.12

The axial load value was lower than 900 N for 0.7 mm cavity. It is seen from the results that the effect of this parameter is low on the axial force. Thus, a small increase in efficiency was mentioned as length increases. It is given a comparison of pressure distribution and velocity vectors for external and internal wear ring in the Fig. 16.

4. CONCLUSION

In this study, the effects of some geometrical parameters and the flow rate affecting the axial load of an end suction centrifugal pump were investigated. Firstly, the influence of the distance between the back of impeller and the body cover on

performance of the pump, which has a closed impeller, was investigated. It was shown that while the distance increases the axial load increases negligible. After determining the distance it was considered the use of the wear ring. It was seen that the wear ring as internal use is more attractive than the external use due to reduce the axial load. According to the analysis results of the diameter of the wear ring, it was found that the diameter is close to the inlet diameter of the pump. Since the distance between the back of impeller and the body cover is not very important on the axial load, the design of the wear ring should be made appropriately and the material should be kept in the foreground.

There is no effect on the use of the balancing holes without wear ring. But, the wear ring should not be used without balancing holes. The balancing hole must be used to reduce the high pressure in the equilibrium chamber. The wear ring must be selected at a suitable cavity and the pressure in this chamber must be reduced by the fluid from the holes. In order to compensate the axial load in the outcomes; while reducing the magnitude of the axial load, the loss of efficiency should be considered. The hole diameter of 5 mm was preferred for the working capacity of the pump. Improvement studies of low efficiency should be continued at this value.

In addition, the axial load decreases as the flow rate increases. However, It is very important to operate at the optimum flow rate. This increased flow rate, a decrease in efficiency was also observed. This showed that the pump should be optimally operated at 30 tons/hour of the flow rate rather than the designed value of 36 tons/hour of the flow rate. It does not cause a serious change in axial loads.

The effect of the distance between the external wear ring and the bracelet of the rotating impeller on the axial load is important. As the distance decreases, it causes more difficult to pass, the axial load accumulated inside the ring reduces. Also, as the distance decreases, the efficiency increases.

The wear ring was analyzed from the inside by the determined the diameter and the cavity. As a result, a decrease in the axial load was observed by

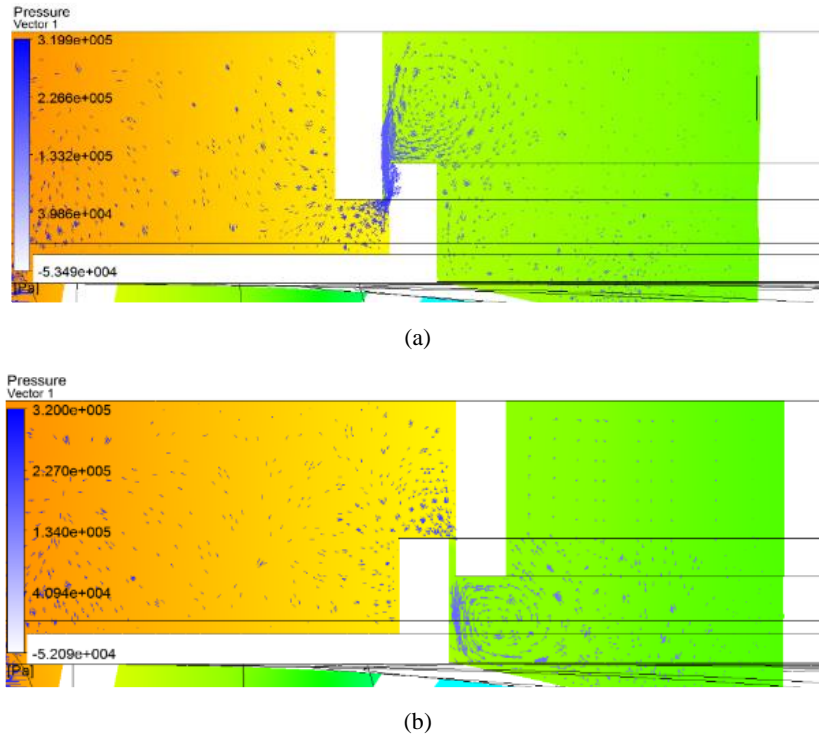


Fig. 16. Pressure distribution and velocity vectors for using (a) the external wear ring (b) the internal wear ring.

determining that the leakage flow was better directed to the balancing holes. This reduction was a value to be considered.

Within the scope of the work done, it was suggested a pump of 30 tons/hour of flow rate and 33 meters of head. It was advisable to select the use of the internal wear ring, the choice of the wear ring diameter near the inlet of the pump, and the wear ring cavity to be 0.7 mm. The distance between the back of impeller and the body cover was selected as 20 mm. The wear ring working distance can be up to 8 mm. The balancing holes were determined to be around 5 mm. The axial force can be reduced by paying attention to all these points.

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