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Numerical Study of Reducer Modification with Adjuster Opening Variations on Turbine Lube Oil Pipe

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Abstract

The success criteria of the power plant are represented by EAF (Equivalent Availability Factor) value. In 2021, one of the causes of the Forced Outage (FO) at Adipala Steam Power Plant was the failure of the main lube oil turbine pipe. The sudden contraction of the reducer resulted in the previous failure. Modifications were made to the reducer using a conical shape to reduce the number of welding processes and minimize the probability of failure. Therefore, a study on reducer modification needs to be carried out to study the impact of changing the shape of the reducer and changing the opening check valve on process parameters and flow in the main lube oil turbine pipe. The modification of the reducer was the main focus in this study. This research aims to analyze the pressure of turbine lube oil in the existing adjuster and the modified adjuster, using the Computational Fluid Dynamics technique with check valve adjuster variations (distance between the tip of the check valve and the inner wall of the reducer). The inlet boundary condition was defined by a pressure inlet of 285803.4 Pa. Outlet boundary condition was set by a mass flow of 65.72 kg/s. The curve surface was set as wall boundary conditions with a stationary wall, no-slip, and standard roughness model. The oil pressure in the modified adjuster increased compared to the oil pressure in the existing adjuster. The pressure on the oil flow was required to distribute oil to the turbine bearings. With the small increase in pressure after the modification, the turbine lube oil transfer improved. The pressure drop (ΔP) value that occurred in the conical-shaped modified adjuster was smaller than the pressure drop (ΔP) in the existing adjuster. This is because the resistance on the modified adjuster was less than the existing adjuster. The flow streamlines that formed backflow and vortex on the conical modified adjuster were less compared to the existing adjuster due to fewer obstacles in the conical modified adjuster.

Keywords: Power-plant, adjuster, sudden contraction, conical

1. Introduction

The development of a long-term electric power system is encouraged to follow the increasing electricity demand. The reliability and efficiency of existing power plants must be maintained to ensure electricity supply availability for consumers. The criteria for the success of a power plant can be seen from how large the unit's reliability value is equivalently (Equivalent Availability Factor, EAF). Another factor that affects the performance or readiness of the unit is an Outage. The Outage occurs when a unit is out of the electric network and is not in the Reserve Shutdown state.

Forced Outage is undesirable in the operation of power plants because it can interfere with the power plant's performance in an equivalent manner (Equivalent Forced Outage Rated, EFOR) [1]. In 2021, One of the causes of the Forced Outage at Adipala Power Plant was the failure in the main lube oil turbine pipe. Modifications were made to the existing reducer to prevent repeated failures. The previous failure occurred in the reducer using a sudden contraction shape in the main lube oil turbine pipe. Modifications were made to the reducer using a conical shape to reduce the number of welding, which had the potential to cause failure. It could reduce the impact of repeated failures. Therefore, a study on reducer modification needs to be carried out to study the impact of changing the shape of the reducer and changing the opening check valve on process parameters and flow in the main lube oil turbine pipe.

Previous research carried out by Kaushik et al. studied the effect of flow on pipe reducers [2]. Examined pipes experiencing sudden contraction and expansion, resulting in analysis of Computational Fluid Dynamics (CFD) models used to predict hydrodynamic characteristics in contraction and expansion flows to be used in practical designs. This study also produced steps to reduce fouling and guidelines for designing sudden contraction and expansion pipes.

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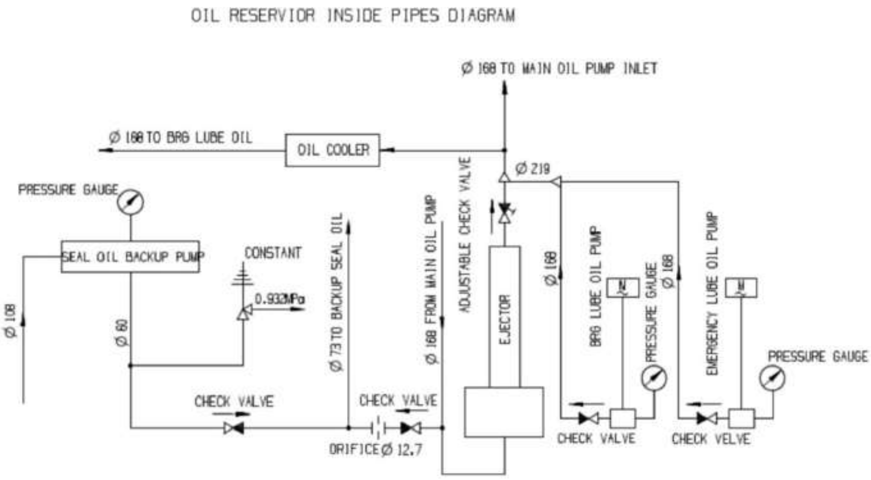


Figure 1. Oil reservoir inside pipe diagram.

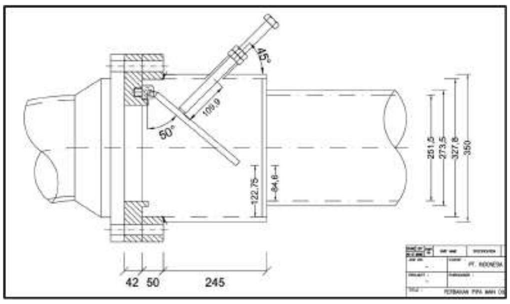
Agarwal and Mthembu [3] investigated fluid flow parameters such as velocity, pressure, and mass flow in conical diffusers with angle analysis of 10°, 20°, 30°. The research resulted in an analysis of the CFD turbulence model. The diffuser angle significantly influenced the fluid velocity in the inlet and outlet zones of the diffuser. The angle of the diffuser cone also affected the pressure drop across the diffuser. The maximum pressure drop in the diffuser was at an angle of 20°. The minimum pressure was at a diffuser angle of 10°. The highest kinetic energy turbulence was near the wall and the nozzle.

Dehkordi et al. [4] researched the hydrodynamic behavior of high viscous oil-water flow through a horizontal pipe that experienced sudden expansion using CFD studies and experimental validation. This study showed that continuous low oil dispersion in water could not be captured because the length scale was not proportional to the pipe diameter. Therefore, a finer mesh resolution was required to capture the scattered stream with a numerical scheme. During core-annular flow, CFD simulation could predict the eccentricity of the oil core that was not in contact with the pipe wall, consistent with the experimentally observed flow visualization.

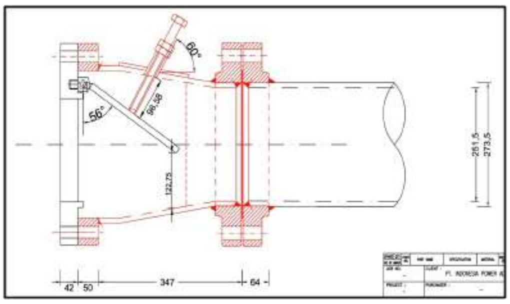
Saleh et al. [5]investigated the flow characteristics in a horizontal pipe conveying a non-Newtonian power-law fluid under laminar conditions. The crude oil entered the pipe with a uniform velocity where the velocity was zero at the wall. As the fluid proceeded along the pipe, the fluid in the region adjacent to the wall decelerated, resulting in acceleration of the fluid in regions near the centerline of the pipe due to continuity. Velocity profile showed clearly more steepened parabolic velocity profile over cross-section compared to the parabolic profile for Newtonian fluids. The CFD simulated axial velocity profile at (z/D = 42) also showed an excellent agreement with the velocity profile calculated using the equation at the fully developed flow.

Wu et al. [6] studied the effect of geometrical contraction on vortex breakdown of swirling turbulent flow in a model combustor. It was found that outlet contraction could significantly impact the vortex breakdown structure and precessing vortex core. Around the axis of the main chamber, there was a vortex core in which the fluid was undergoing helical motion toward the upstream regions of the main chamber.

This study conducts a numerical analysis of the effect of modification of the existing reducer (sudden contraction) into a conical reducer on the main lube oil turbine pipe.



(a) Existing adjuster.



(b) Modified adjuster.

Figure 2. Dimension of adjuster.

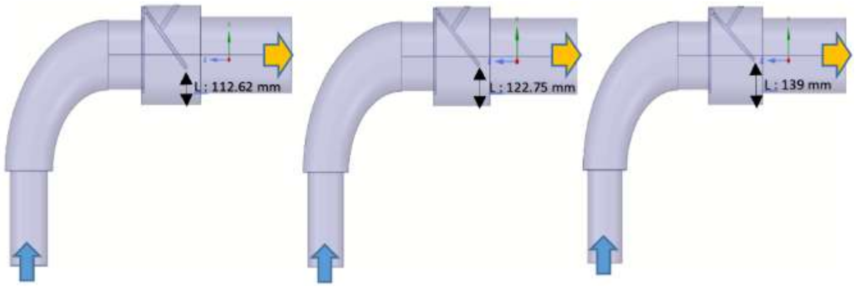


Figure 3. Variation of adjuster opening (existing/sudden contraction).

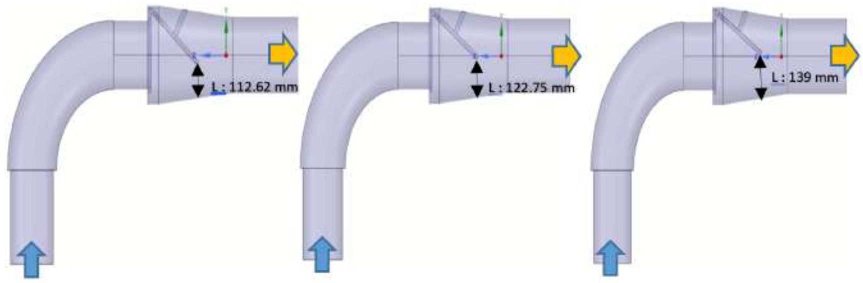


Figure 4. Variation of adjuster opening (modification/conical).

2. Numerical method

The main lube oil supply to the main bearing turbine was pumped by Main Oil Pump (MOP), which ran directly by the turbine shaft. As a backup when the turbine operation was low, the oil supply was pumped by Bearing Oil Pump (BOP) [7]. The check valve had a function to prevent black flow from the BOP pump to the suction MOP pump. The check valve had an adjuster function to regulate flow pressure from the MOP pump [8].

Based on Figure 1, when the turbine normally operated at around 3000 rpm, the lubricant of the turbine bearing pumped by Main Oil Pump (MOP) passed through the ejector in the Main Lube Oil Tank (MOT) tank to add the volume of debit supply lube oil. Then, the lubricant passed through the adjuster, which arranged the pressure required by the turbine bearing. The adjuster separated the lubricant output into two parts. Part of the lubricant went to the turbine bearing and the other part returned to the suction side of the Main Oil Pump (MOP) [9].

As shown in Figure 2, the change from the existing adjuster, which shaped sudden contraction, to the modified adjuster, which shaped sudden conical. The pressure of turbine lube oil in an adjuster existing and modified was analyzed using the Computational Fluid Dynamics Technique. The check valve adjuster variations (distance between the tip of the check valve and the inner wall of the reducer) in the existing condition were 122.75 mm. The maximum aperture of the check valve adjuster was 139 mm and the minimum aperture was 112.62 mm, as shown in Figures 3 and 4.

The CAD model of the adjuster had been developed and imported into the ANSYS design modeler, as shown in Figure 5. The meshing followed the straight or curve profile of the reducer and adjuster, producing the natural fluid flow pattern [10]. Neat and tight meshing was needed for simulation with the $k-\omega$ standard model. The meshing used was combined hex and polyhedral meshing. The combination of hex and polyhedral mesh produced good density with a total node of 904383 nodes.

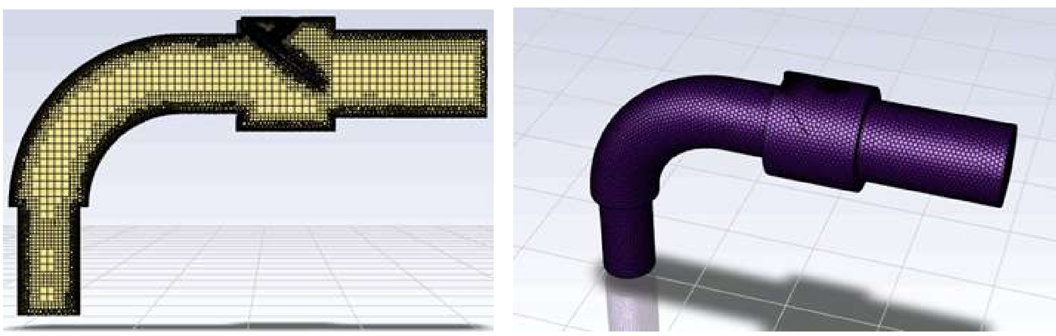


Figure 5. Meshing of computational domain.

The inlet boundary condition was defined by a pressure inlet of 285803.4 Pa. Outlet boundary condition was set by a mass flow of 65.72 kg/s. The curve surface was set as wall boundary conditions with a stationary wall, no-slip, and standard roughness model. The solution methods used in this research can be seen in Table 1.

Table 1. Solution methods fluent 2020.

Parameter	Type
Pressure-Velocity Coupling Scheme	SIMPLE
Spatial Discretization	-
Gradient	Least Squares Cell Based
Pressure	Second Order
Momentum	Second Order Upwind
Turbulent Kinetic Energy	Second Order Upwind
Turbulent Dissipation Rate	Second Order Upwind

3. Results and Discussion

The CFD Analysis was done on the adjuster existing and modification using ANSYS 2020 RA. The adjuster varied in different adjuster openings, which were 112.62 mm, 122.75 mm, and 139.00 mm. The adjuster variation was the distance between the wall and the endpoint of the disc check valve. This analysis used the $k-\omega$ model, consisting of pressure contour, velocity contour, and velocity streamline.

3.1. Validation

The result of the numerical simulation for existing adjusters with standard adjuster openings of 122.75 mm had been validated before the simulation continued for other variations in adjuster openings. Validation was carried out by comparing the existing adjuster simulation outlet pressure with the pressure indicator validation data, as shown in Figure 6.

As shown in Figure 7, the pressure indicator on the downstream side had a pressure of 280 kPa and the existing adjuster simulation results had an outlet pressure of 279.9 kPa. So, simulation results were close to the actual condition with a deviation value of 0.0357%. The error limit was less than $\pm$ 5%. Therefore, the adjuster simulation could be continued.

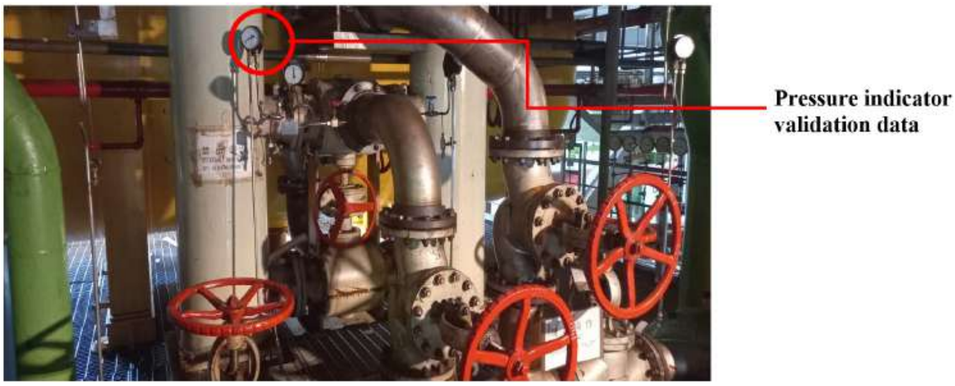


Figure 6. Position of data collection for outlet pressure adjuster (before filter).

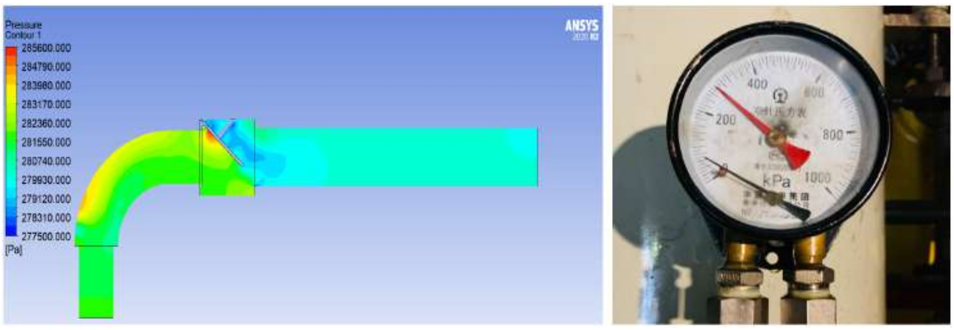


Figure 7. Simulation results (pressure contour) compared to the pressure indicator on adjuster-opening 122.75 mm.

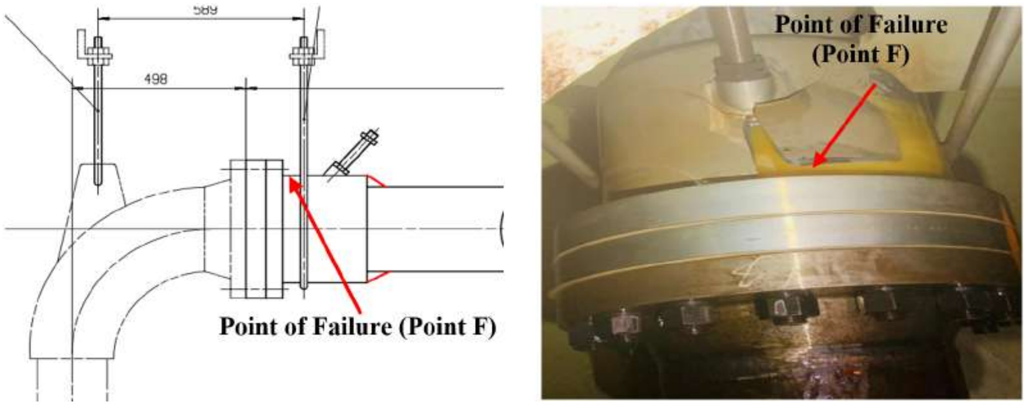
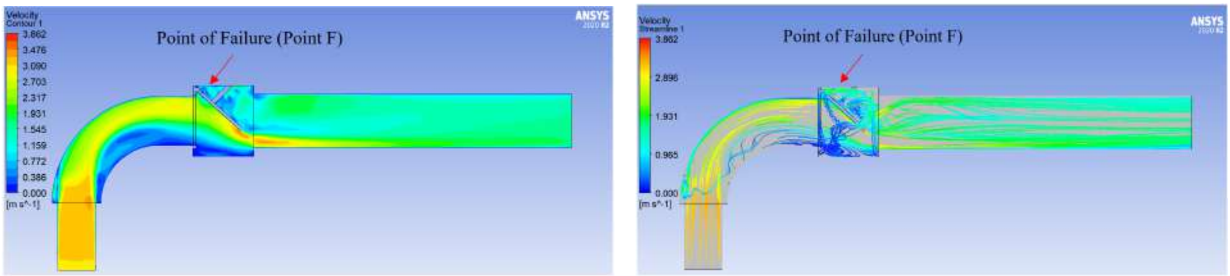


Figure 8. The position of failure (point F).



(a) Velocity contour.

(b) Velocity streamline.

Figure 9. Simulation result of existing adjuster.

3.2. Flow Characteristic on Point F

This research was based on the failure history of the existing adjuster, as shown in Figure 8. The methodology used had been developed using Computational Fluid Dynamics (CFD).

The failure point (point F) could be analyzed according to the flow pattern on the existing adjuster. Figure 9 showed the simulation result with a standard adjuster opening on normal lube oil pressure, which was 0.2858 Mpa.

Figure 9(a) showed a significantly increased lubricating oil velocity at the end of the adjuster disc. The oil velocity on the existing adjuster with a standard opening condition of 112.75 mm at point F was 0.2615 m/s. A void occurred on the back side of the check valve disc in the point F area, which created high pressure in the area of the void. This was due to the re-circulation of flow, causing a vortex of flow in the void area. The velocity in the void was low and the pressure increased. Figure 9(b) showed that the disc check valve blocked the flow from the inlet. Then the flow was blocked at the corners of the sudden contraction reducer, causing a vortex of flow in that area until the flow reached point F. The continuous vortex of flow caused a continuous impact in the area. So, this could cause failure in the void area due to the impact pressure of the flow that occurred repeatedly.

As shown in Figure 10, the pressure contour was also relevant to the velocity contour, which increased significantly in the check valve disc area caused by flow resistance. The dimension of the sudden contraction reducer also caused flow resistance, so the flow at the corners of the reducer also increased. This resistance has caused backflow, so the circulation occurred as a vortex, as seen in Figure 9(b). The pressure that occurred at point F was increased caused by a vortex. If the flow pressure occurred continuously, the flow pressure put pressure on the area and caused a failure.

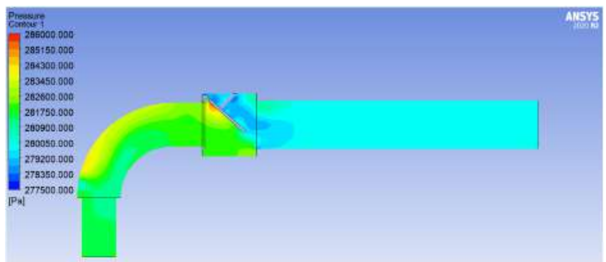
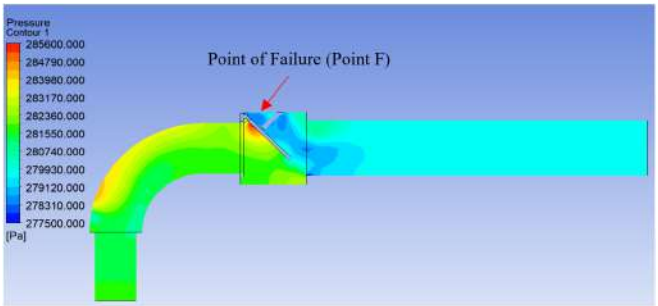
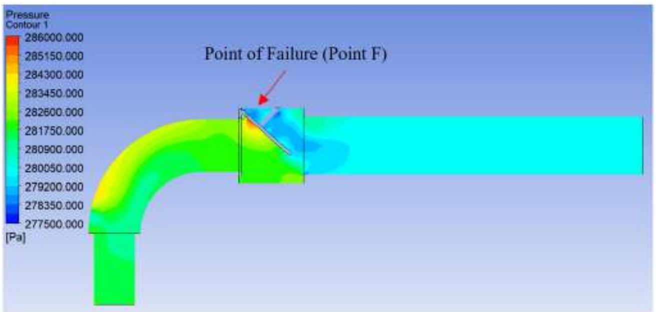


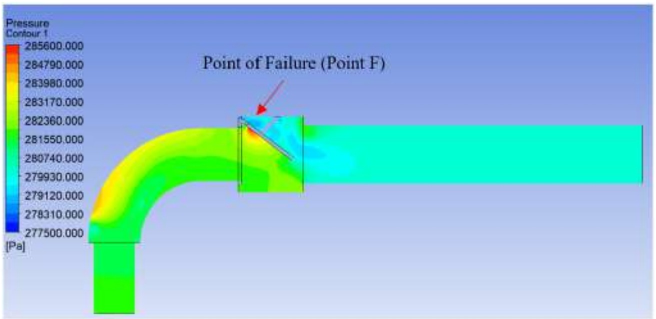
Figure 10. Pressure contour of existing adjuster (sudden contraction).



(a) Adjuster-Opening 112.62 mm

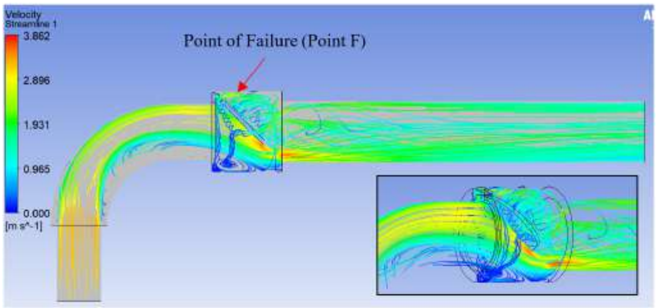


(b) Adjuster-Opening 122.75 mm

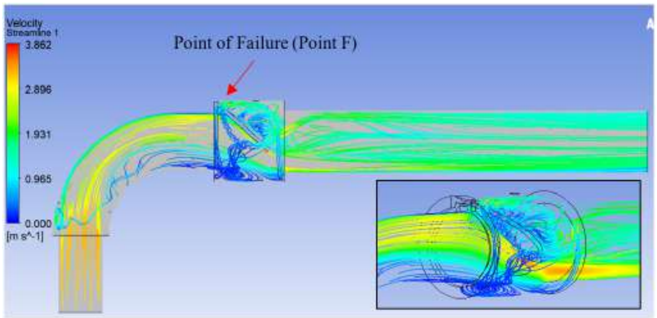


(c) Adjuster-Opening 139.00 mm

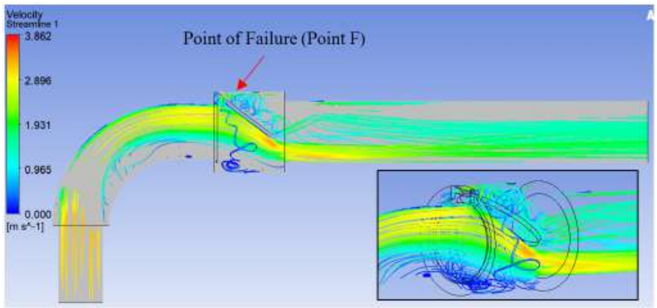
Figure 11. Pressure contour of existing adjuster (sudden contraction)



(a) Adjuster-Opening 112.62 mm



(b) Adjuster-Opening 122.75 mm



(c) Adjuster-Opening 139.00 mm

Figure 12. Streamline of existing adjuster (sudden contraction)

3.3. Simulation of Existing Adjuster

Simulation of the existing adjuster with variations of adjuster opening was carried out to observe the flow pattern with an inlet pressure of 0.2858 MPa and mass flow of 65.72 kg/s. As shown in Figure 11, pressure contours for all models showed an increase in oil pressure which approached and passed through the check-valve disc, showed by red and yellow colors. The oil pressure reduced after it passed through the adjuster, moving toward the outlet. An increase in velocity caused pressure reduction in the cross-sectional area. The pressure was reduced to the minimum value point when the fluid passed through the cross-sectional area. After that, the pressure increased slightly above the minimum value. All models' pressure at point F showed an increase in oil pressure as the adjuster opening increased, as shown by the color change of point F in Figure 11. The reduced flow in the cross-sectional area caused pressure reduction downstream. The maximum and minimum pressure at point F was 0.2795 MPa and 0.2789 MPa, respectively. Both occurred in an adjuster opening of 139 mm.

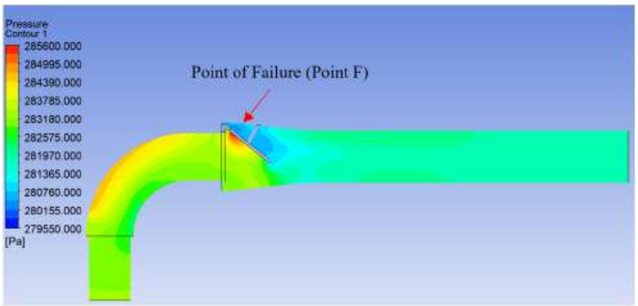
Figure 12 showed the flow streamline pattern. The flow streamline pattern showed that the oil flow was taking up a vortex on the front of the adjuster. The resistance because of the reducer dimension shape (sudden contraction) created backflow and vortex that passed through the wall toward point F as a space blocked by a check-valve disc. The flow of vortex (recirculation) toward point F added the potential for repeated failure caused by flow pressure toward the area of point F. Adjuster opening affected the vortex flow. The adjuster opening standard of 122.75 mm produced a larger vortex than the adjuster opening of 112.62 mm and 139.00 mm. The smallest vortex resulted from an adjuster-opening 139 mm. Table 2 compared the pressure and velocity at point F for variations of adjuster openings. The increasing pressure at point F was aligned with the adjuster opening and inversely proportional to the velocity.

Table 2. Data of existing adjuster simulation.

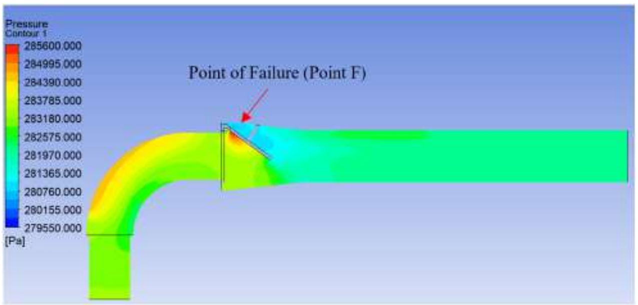
Adjuster-Opening (mm)	Pressure-Point F (MPa)	Velocity-Point F (m/s)
112.62	0.2789	0.7608
122.75	0.2791	0.2615
139.00	0.2795	0.1459

Table 3. Data of modified adjuster simulation.

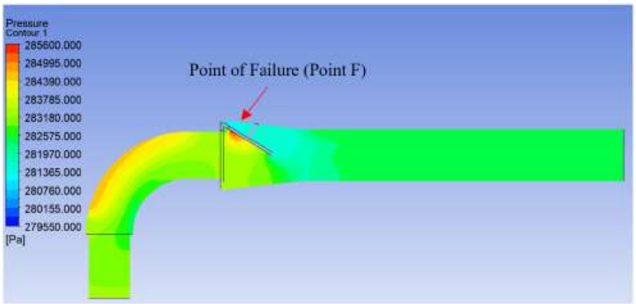
Adjuster-Opening (mm)	Pressure-Point F (MPa)	Velocity-Point F (m/s)
112.62	0.2804	0.1132
122.75	0.2810	0.0908
139.00	0.2813	0.0737



(a) Adjuster-Opening 112.62 mm

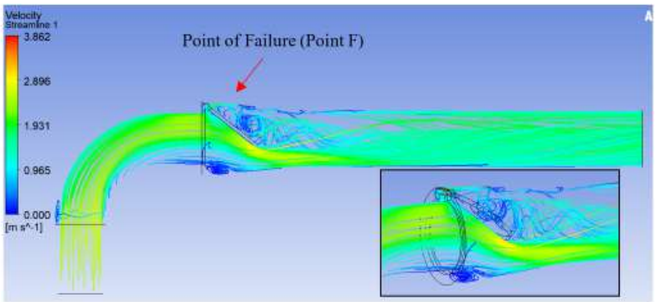


(b) Adjuster-Opening 122.75 mm

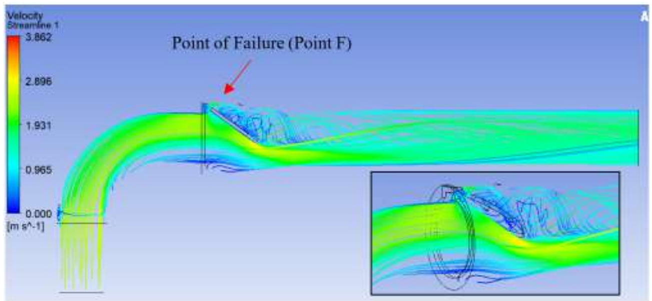


(c) Adjuster-Opening 139.00 mm

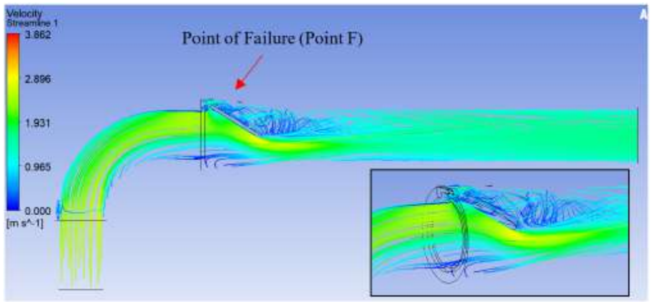
Figure 13. Pressure contour of modified adjuster (conical)



(a) Adjuster-Opening 112.62 mm



(b) Adjuster-Opening 122.75 mm



(c) Adjuster-Opening 139.00 mm

Figure 14. Streamline of modified adjuster (conical)

3.4. Simulation of Modified Adjuster

A simulation of the modified adjuster was carried out to find the flow pattern. This research was to know and analyze the flow characteristics of the pipe adjuster main lube oil turbine, which focused on the point of failure (point F). Figures 13 and 14 showed the simulation result of the flow characteristic with a modified design, inlet pressure 0.2858 MPa and mass flow 65.72 kg/s.

There was an increase of pressure in the area of point F after modification, compared to the pressure in the existing adjuster simulation, as shown by the pressure contour in Figure 13. A decrease in velocity caused pressure to increase due to the enlargement in the cross-sectional area compared to the existing adjuster. The shape of the existing adjuster, which was a sudden contraction, had a smaller cross-sectional area causing increased velocity value and decreased pressure value. The highest pressure of point F was at the adjuster opening of 139 mm. The pressure of the oil flow was required to distribute oil to the turbine bearings. With the small increase in pressure after the modification, the turbine lube oil transfer improved.

Table 4. Data of pressure and velocity simulation in point F.

	Adjuster-Opening (mm)		
	112.62	122.75	139
Pressure (P_F)			
Modification Adjuster	0.2804 MPa	0.2810 MPa	0.2813 MPa
Existing Adjuster	0.2789 MPa	0.2791 MPa	0.2795 MPa
Velocity (V_F)			
Modification Adjuster	0.1132 m/s	0.0908 m/s	0.0737 m/s
Existing Adjuster	0.7608 m/s	0.2615 m/s	0.1459 m/s

Figure 14 showed the decrease in the number of vortex compared to the existing adjuster. The modified adjuster had less resistance than the existing adjuster. Less resistance caused less backflow and circulation. It caused a vortex and increased pressure in the area. This flow caused a repeated impact on the adjuster wall. With a smaller resistance after modification, the flow was perfect, and the impact pressure of the oil flow was getting smaller.

The modified adjuster with an opening of 122.75 mm produced less vortex than the adjusters with an opening of 112.62 mm and 139.00 mm, as shown in Table 3. So, with a smaller value of vortex, the modified adjuster is better than the existing adjuster. The simulation result of three models in the existing adjuster and modified adjuster was showed in Table 4.

3.5. Comparison Between Existing and Modified Adjusters

The comparison of existing and modification adjusters for pressure and velocity was showed in Figures 15 and 16. The modified adjuster had increased pressure in point F compared with the existing adjuster, with a value of 0.2813 MPa with the adjuster opening of 139.00 mm. In the existing adjuster, the highest pressure value occurred in opening 139.00 mm with a value of 0.2795 MPa. This value was aligned with the reduction of velocity in point F in both adjuster modified and existing adjuster. Although the modification adjuster (conical) produced high pressure, the flow pattern was better because of less number of the vortex. Based on the explanation above, with a smaller vortex number, the potential failure from repeated flow impact pressures was smaller.

The modified adjuster had a lower pressure drop value than the existing adjuster by calculating the inlet pressure and the outlet pressure difference, as shown in Table 5. The highest pressure drop occurred with the adjuster opening of 112.62 mm.

The lower pressure drop was in a modified adjuster with an adjuster opening of 139.00 mm. The flow of the existing adjuster has gotten more resistance which caused the pressure drop to increase compared with the modified adjuster. So, the modified adjuster with a lower pressure drop was better than the existing adjuster.

Table 5. Data of pressure drop

Opening Adjuster (mm)	ΔP Modified-Adjuster (MPa)	ΔP Existing-Adjuster (MPa)
112.62	0.0041	0.0062
122.75	0.0039	0.0059
139.00	0.0035	0.0057

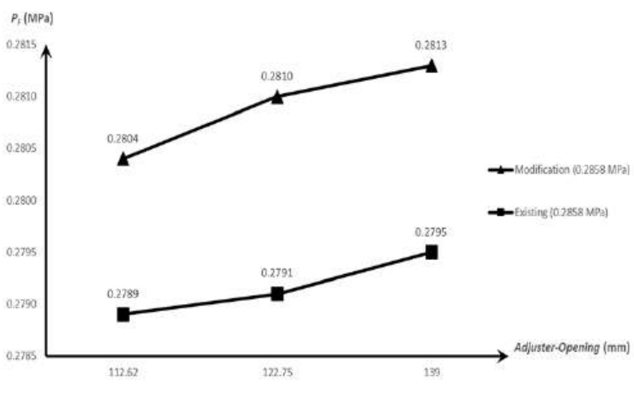


Figure 15. Graphic of pressure on point F with adjuster opening variation.

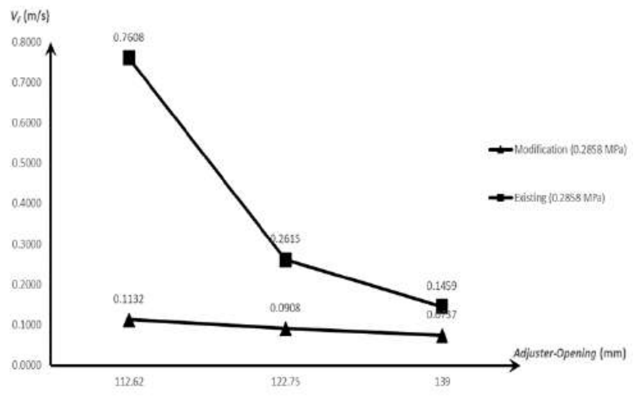


Figure 16. Graphic velocity on point F with adjuster opening variation.

4. Conclusion

The simulation results showed that the change in pressure was proportional to the change in velocity. The modified adjuster had larger pressure in point F than the existing adjuster, with a value of 0.2813 MPa with an adjuster opening of 139 mm. In the existing adjuster, the highest pressure value occurred in opening 139.00 mm with a value of 0.2795 MPa. This value was aligned with the reduction of velocity in point F. The result conformed to the analytical representation of energy conservation from the Bernoulli equation.

The pressure at point F in the existing adjuster had increased in oil pressure aligned with the adjuster opening caused by the reduction of flow in the cross-sectional area. The pressure value after adjuster modification increased compared to the pressure value in the existing adjuster. The pressure on the oil flow was required to distribute oil to the turbine bearings. With the small increase in pressure after the modification, the turbine lube oil transfer improved.

The pressure drop (ΔP) in the conical-shaped modified adjuster was smaller than the pressure drop (ΔP) in the existing adjuster. This is because the resistance on the modified adjuster was less than the existing adjuster. The flow streamlines that formed backflow and vortex on the conical modified adjuster were less compared to the existing adjuster due to fewer obstacles in the conical modified adjuster.

Based on the analysis of pressure, streamline, wall shear, pressure drop and operating parameters, the recommendation was to use a conical modified adjuster with an opening of 122.75 mm.

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