Design & Analysis of Orifice Bypass Arrangement System

Dr. CHITRIKI THOTAPPA [1], Dr.G JAGANNATHA REDDY [2] and SANTOSH KUMAR K [3]

Abstract- A good design of the systemshould always ensureproper functioning and its safety. Due to advances in computer technology & numerical methods, simulation is feasible to check and improve the design of the system. In this project an existing Automatic Recirculation Control (ARC) valve has been replaced with a newly designed **orifice bypass arrangement system.** As the existing ARC system is complicated and it develops high pressure and velocity that leads to damage of the bearings, shafts etc. and it also involves high cost valve, to control the bypass of fluid flow for the safety of Multi stage centrifugal pump,a new orifice bypass system is designed to overcome the disadvantage of existing system.The newly designed orifice bypass system comprises 6 stages, which resist flow& pressure drop of the fluid so, as to the safe guard the centrifugal pump**. This paper**deals with design and analysis of various stresses like Radial, Hoop and Vonmises stress of thedesigned model (orifice bypass arrangement system) using solid works software. The newly design Orifice bypass system will reduce the flow pressure from 210 bar to negligible outlet pressure and also reduce Velocity from 130m/sec to negligible velocity, keeping the inlet pressure constant for further process. Thus, the newly designed orifice bypass arrangement system functions effectively and also safeguard the centrifugal pump under no load conditions.

Index terms- Orifice bypass system, Automatic Recirculation Control (ARC), Pressure, velocity, Radial, Hoop, Vonmisesstresses.

1. INTRODUCTION [1]

Since the dawn of civilization, mankind has always had a fascination with fluids; whether it is the flow of water in rivers, the wind and weather in our atmosphere, the smelting of metals, powerful ocean currents or the flow of blood around our bodies.

In antiquity, great Greek thinkers like Heraclitus postulated that "Everything flows" but he was thinking of this in a philosophical sense rather than in a recognizably scientific way. However, Archimedes initiated the fields of static mechanics, hydrostatics, and determined how to measure densities and volumes of objects. The focus at the time was on waterworks: aqueducts, canals, harbors, and bathhouses, which the ancient Romans perfected to a science.

Leonardo was followed in the late 17th Century by Isaac Newton in England. Newton tried to quantify and predict fluid flow phenomena through his elementary Newtonian physical equations. His contributions to fluid mechanics included his second law: F=m.a or m.a the concept of Newtonian viscosity in which stress and the rate of strain vary linearly, the reciprocity principle: the force applied upon a stationary object by a moving fluid is equal to the change in momentum of the fluid as it deflects around the front of the object, and the relationship between the speed of waves at a liquid surface and their wavelength.

In the early 20th Century, much work was done on refining theories of boundary layers and turbulence in fluid flow. Ludwig Prandtl (1875-1953) proposed a boundary layer theory, the mixing length concept, the Prandtl number, andmuch more that we take for granted.

It is debatable as to who did the earliest CFD calculations (in a modern sense) although Lewis Fry Richardson in England (1881-1953) developed the first numerical weather prediction system when he divided physical space into grid cells and used the finite difference approximations of Bjerknes's "primitive differential equations". His own attempt to calculate weather for a single eight-hour period took six weeks of real time and ended in failure! His model's enormous calculation requirements led Richardson to propose a solution he called the "forecast-factory". The "factory" would have involved filling a vast stadium with 64,000 people. Each one, armed with a mechanical calculator, would perform part of the flow calculation. A leader in the center, using colored signal lights and telegraph communication, would coordinate the forecast. What he was proposing would have been a very rudimentary CFD calculation. The earliest numerical solution for flow past a cylinder was carried out in 1933 by Thom and reported in England.

Fluid dynamics deals with the dynamic behavior of fluids and its mathematical interpretation is called as Computational Fluid Dynamics. Fluid dynamics is governed by sets of partial differential equations, which in most cases are difficult or rather impossible to obtain analytical solution. CFD is a computational technology that enables the study of dynamics of things that flow.
The procedure for the CFD analysis in FLOTRAN follows the simple steps below:

- ✓ The model used for the analysis is drawn, meshed and the boundary layers are determined. This is done using the Solid works software.
- ✓ The model is defined for the type of solver and boundary conditions to be used.
- ✓ The model is defined according to the type of analysis required.
- ✓ The model is solved by setting the required parameters in the solution panel and then iterated for convergence.
- ✓ Results can be obtained from the graphic.
- ✓ Finally, the results and all the data can be saved for future references by writing the files.

2. LITERATURE REVIEW

2.1 Existing System [2]

Initially,Turbo-Cascade High Pressure Control Valves were designed to combat erosive forces caused by high velocity flow and to prevent pressure drop which can cause cavitations damage due to consequence of the collapse of vapour bubbles in the fluid flow stream. Now, in the existing Series 9100 ARC provides economical as well as effective protection for centrifugal pumps against the serious damage from overheating or instability that can result even from a few minutes of low flow. By providing recirculation flow to the inlet of the pump, the 9100 ARC assures a minimum flow for proper pump operation.

On the other hand, Existing ARC valve system is complicated and it develops high pressure and velocity that leads to damage of the bearings, shafts etc. and it also involves high cost valve, to control the bypass of fluid flow for the safety of Multi stage centrifugal pump. It also requires long duration shut down for repairs and wear of parts. The existing ARC details of internal parts are shown in **Fig 2.1**

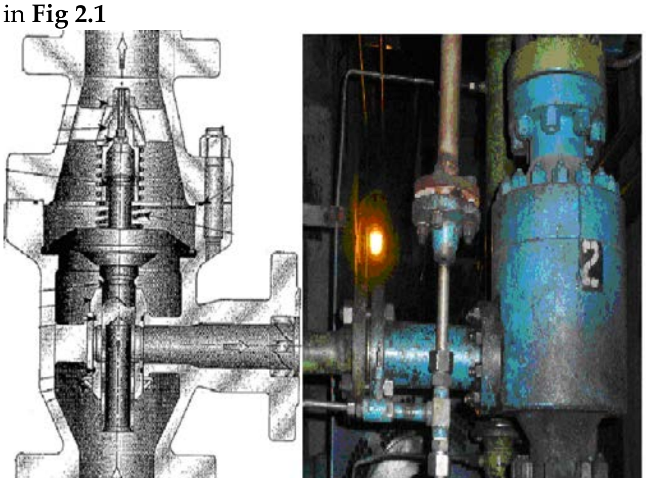


Fig 2.1. Automatic Recirculation control Valve (ARC) – Existing System

2.2 Problem identification in existing system

In the existing system due to high pressure & velocity, the following failures are observed.

2.2.1Thrust bearing failure in multi stage centrifugal pump [3]

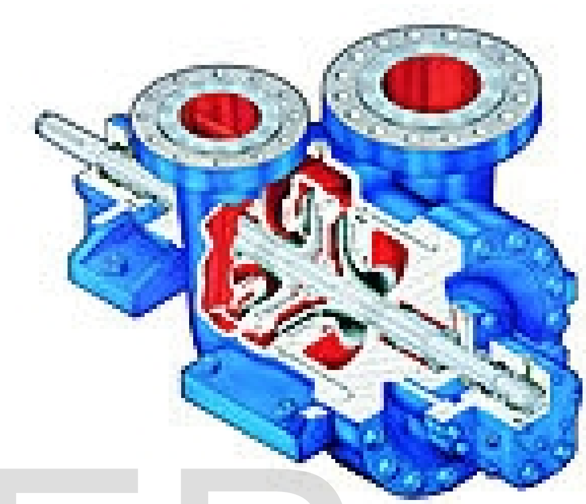


Fig: 2.2 Failures in thrust bearing

The axial bearing of the above structure includes a ring, pads and a base. Multiple, independent pads are orientated in a curved or circular configuration with respect to one another. The pads are retained by recesses in the base. Each pad conforms in shape and size to a corresponding recess's shape and size. The pads have sides which face a face of the ring. The pads preferably tapered in axial height to increase the circulation of cooling and lubricating fluid between the pads and the face. If appropriately positioned in a centrifugal pump, the axial bearing of the pump can perform both the functions of axial bearing and a wear ring .The axial bearing is preferably installed between the front of the impeller and the housing. The recesses are located in a front shroud of the impeller so that the need for a separate base is obviated. In other words, the base may be integrated into the impeller itself the impeller is optimally made from a plastic or polymer that can flex during operation to dampen axial load changes. When the axial bearing is used as a wear ring, a leakage joint is formed between the pads and the ring.

The leakage joint limits the flow of fluid from the pump discharge region to the intake region. To keep flow through the leakage joint to a minimum, the pads and the ring are made from ceramics or Polytetrafluoroethylene(PTFE)so sliding contact may occur. If other materials are used for the pads and the ring, the pads need to be separated from the ring by a

ISSN 2229-5518odd page

To further increase the performance of the axial bearing as a wear ring, the axial shroud extension is optionally added to the front shroud of the impeller. Each pad is optimally placed in a recess or pocket that can potentially retain an individual, broken pad. If a pad breaks during operation of the pump, the broken pad is retained within an isolated pocket or recess region while the other pads can often continue to function normally.

In practice, whether the pad is broken or not,it is adequately retained and it depends upon the number of pad fractures among other factors. The recess may be lined with a resilient member to avoid damage to the face when operating with a broken pad. The resilient member compensates for angular misalignment. The pads are retained in recesses in the base which allows the pads to be readily removed during repair and replacement of the axial bearing. Each recess has a first interlocking cross section. The first interlocking cross section corresponds in size and shape to a second interlocking cross section of any pad.

The first interlocking cross sections and the second interlocking cross sections are united to frictionally retain the pads in the recesses. The peripheral surface of the pads are prevented from radial movement and rotational movement by the recesses. The walls of the recess prevent radial and rotational movement. While the frictional retention of the pads may be accomplished by a press-fit, the clearances between the pads and recess are preferably such that the pads are readily removable from the base during repair and inspection of the axial bearing.

Each pad is individually placed in a corresponding recess, reducing the amount of force necessary to assemble an axial bearing in comparison to certain axial bearings of the background art. The efficiency of the pump manufacturing Process is increased by reducing delays from broken axial bearing rings. In addition, repair and disassembly of pumps in the field is simplified through use of the axial bearing of the present invention.

2.2.2 Armature failure of end ring [4]

Damage to stator winding when reverse flow passes through the impeller accelerating its rotation in opposite direction, it creates substantial amount of hydraulic imbalance inside the casing across the impeller.The unbalance hydraulic forces give rise to radial and axial load of variable magnitude and direction acting on rotating parts and static parts as well.The thrust bearing which is generally angular contact by design gives in first undergoing fatigue under this kind of reciprocating impact.

As the speed approaches runaway speed, the rubbing speed of rolling elements also increases leading to high vibration and noise level and the rotor tends to shift towards lower pressure side of the impeller. This causes failure of radial bearing and opening of seal faces, rubbing of impeller with casing or diffuser etc. More over in case of multistage pump having higher shaft length the bending of shaft also takes place due to uncontrolled radial load.

3. PROJECT METHODOLOGY

To overcome the problems (i.e. Failures) in the existing system, a new orifice bypass system is designed &analysed. To achieve the objective of the project following methodology is used.

- ✓ Understanding of 2D representation of the model
- ✓ Modeling of the components using Solid works parametric language.
- ✓ The sections to be optimized are represented with scalar parameters
- ✓ Assigning steel properties for the model.
- ✓ Application of support boundary conditions and pressure boundary conditions
- ✓ Solving the problem using Solid Works default solver
- ✓ Post processing the results and capturing images for report generation
- ✓ Specifying design and state variables with constraints
- ✓ Selecting Subproblem optimization tool for weight optimization and executing the problem

Design of Orifice Meter with theoretical calculations, Modelling of Orifice meter using 2D axisymmetric approach, Meshing and applying fluid pressure loads, Stress analysis and further improvement in the model for better results analyzing the problem in 3D domain to study the effect of nozzle holes on the geometry. Modelling of whole orifice assembly using Solid work mixed approach meshing of the object Initial stress analysis to validate the safety of the structure .Final flotran analysis to check the drop in pressure and velocity results.

Table 3.1 Specification of the System

Pipe ID	42 mm
Orifice ID	4.8 mm
Head	2300 M
Coefficient of discharge	0.61
Flow per hole	184.5 lpm
Thickness	5 mm
Velocity	130 m/sec
Calculated coefficient of discharge	0.757
No. of holes per orifice	6
Total Pressure drop required	210 bar
No of orifice	6
Distance from one orifice	42 mm
Total length of orifice set	84 mm
Actual length	230 mm
Total length	314 mm
Material	SS316
Yield stress	550 Mpa

3.1 Orifice Modelling and Meshing Details

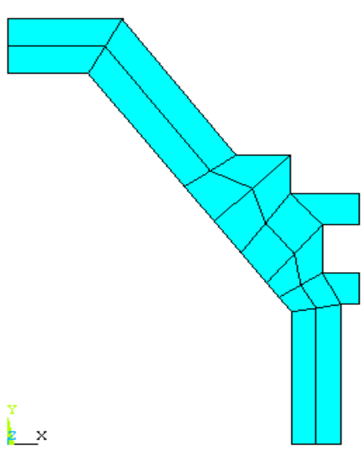


Fig 3.1 Initial geometry of the model

The **Fig 3.1** shows in geometry built for the Solid Works drawing are represented. For initial axisymmetric Analysis, holes are neglected. The geometry has been divided for meshing of the structure. Generally, meshing of Structure yields better results.

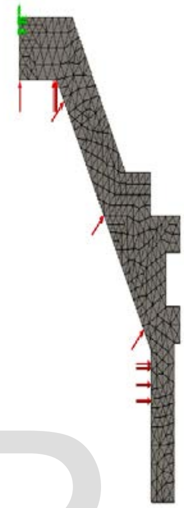


Fig 3.2 Meshed and boundary conditions plotwithout fillet

The **Fig 3.2** Shows meshed plot of the axisymmetric model. Pressure load is applied on the inner surface, a value of 28.75 Mpa is considered.

Modified Geometry

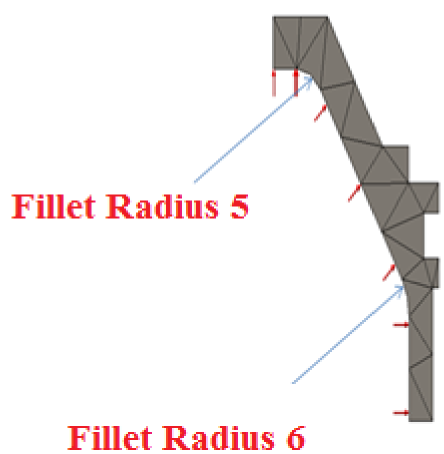


Fig 3.3 Meshed and boundary conditions plotwith fillet

To avoid stress concentration as well as sharp edges, the edges are smoothened with fillet option of radius 5 & 6, as shown in **Fig 3.3**

Full 3D orifice assembly

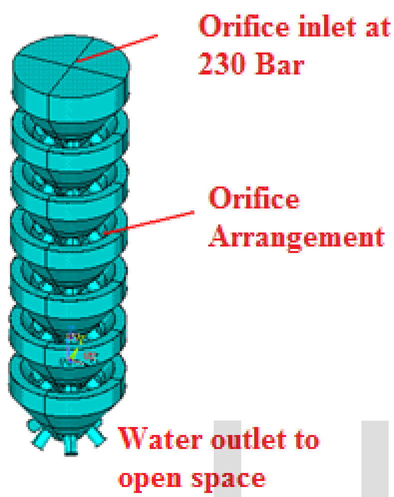
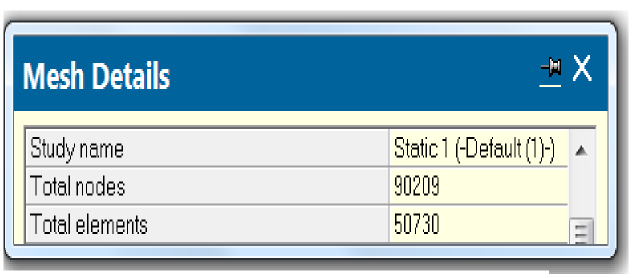


Fig 3.4 3D Representation of the model

The **Fig 3.4** shows overall orifice assembly arrangement of water inlet and out let positions. This 3D geometry is built using solid Works Software with mixed approach.



Study name	Static 1 (-Default (1)-)
Total nodes	90209
Total elements	50730

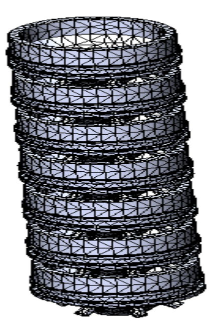


Fig 3.5 Meshed plot of the model

The **Fig 3.5** Shows meshed plot of the fluid region Solid

4. RESULTS & DISCUSSION

Theoretical Design Calculations: [5]

<u>**Design Calculation**</u>

For the design calculation the values are referred in**table 3.1**

Inlet pipe ID = 42mm
Orifice hole ID = 4.8 mm x 6 holes per orifice
Total head= 2300 mtr = 230 bar
No of orifice hole = 6 holes per orifice.
Co-efficient of discharge (cd) e= 0.61 (as per theory)
Velocity = 130 m/sec

Here, in this design calculation following 6 steps are performed to find out various parameter values.

Step-1 Flow through one orifice
Step-2 Flow Resistance per orifice
Step-3 Velocity flow through one hole
Step-4 Co-efficient of discharge
Step-5 Flow resistance for multiple hole
Step-6 Flow through all 6 orifice & pressure drop

Step-1 Flow through one orifice

Flow through one orifice hole i.e. ID=4.8

$= 0.206 \times cd \times \{Orifice.ID\}^2 x \sqrt{total\ \ head\ \ in\ \ mtrs}\ x$
$[1/1- (orifice\ hole\ ID/Pipe\ ID)\ 4]^{0.5}$

$=0.206x0.61 \times (4.8)^2 x \sqrt{2300\ \ mtr} \times [1/1 - (4.8/42)^4]^{0.5}$

$= 0.206 \times 0.61 \times 23.04 \times 47.95 \times [(1/1-(0.114)^4]^{0.5}$

=141.1 1pm per one hole per orifice

For 6 hole per orifice = 141.1 lpm x 6 holes

= **848.6 lpm** through one orifice

Step-2 Flow Resistance per orifice

Flow resistance per orifice multi holes

= Total head / total flow in one orifice (6 holes)

= 2300 mtr / 848.6 lpm

Resistance = **2.71**

<u>Step-3</u> Velocity flow through one hole

Velocity = flow through one hole x 1000 / 60 x 100 x Q 785 x (orifice/10)^2

= 141.1 lpm x 1000/60 x 100 x 0.785 x (4 8mm/10)^2

= 141.1 lpm x 1000/60 x 100 x 0.785 x 0.2304

Velocity = **130 m/s**

<u>Step-4</u> Co-efficient of discharge (Cd)

Calculated coefficient of discharge = 0.7 + (0.73-0.52) *x* thickness

= 0.7+ [(0 73-0 52) x (thickness of orifice / orifice ID)-0.5]/2

= 0.7+ [0.21 x (5/4.8 mm)-0.5]/2

= 0.7-0.14

Cd **= 0.56** =calculated cd

i.e., *cd*=0.56

<u>Step-5</u> Flow resistance for multiple hole

No.of orifice = 6 Nos

Resistance for all multi holes orifice= resistance per single multi hole orifice x no of orifices + calculated cd

=2.716x6 +0.56

= 16.856

i.e., Resistance for all multi hole is **16.856**

<u>Step – 6</u> Flow through all 6 orifice & pressure drop

Flow through all orifice = Total head / Resistance

= 2300 mtr / 16.856

= 141.1 lpm @ a velocity 21 .7 m/s

Velocity = Velocity x (Flow Through/ orifices/6 holes)/flow through one hole

= 130.1 m/sec x (141/6)/141.1 lpm
i.e., Velocity at end of orifice = **21.7 m/sec**

Pressure rating calculation for orifice drum / cylinder i.e. We have considered (standard values) 2 ½ "sch 160 SS pipe;

D= outside diameter of the pipe / drum = 2.875"
t= design thickness of drum / pipe =0.374"
C= allowance in inches for corrosion or mechanical strength =0.05"
Y= a coefficient having value for 0 .4 up to 900^0 F
Calculation Formula = P= 2S (t-c) / D-2y (t-c) =
P= 2X 16000PSI (0.374 "-0.05") / 2.875"-2X0.4 (0.374"-0.05")
P= 32000 PSI X 0.324 / 2.875"-0.2592
P= 10 368 / 2.615
P= 3964 PSI = **273.43 bar**
i.e. calculated pressure is **273.43 bar** but actual working pressure in the system is 210 bar. Maximum design pressure in the 2½ "Sch 160 pipe is 53081 KPa i.e. 530 Kg / cm^2 (530 bar)sodesign is safe**.(**By referring the standard pressure reading chart for stainless steel pipes)

4.1 The stress Results plots

The following stress results for Radial, Hoop & Vonmises stress neglecting hole effect shown below.

4.1.1 Axisymmetric Results (For initial model without fillet)

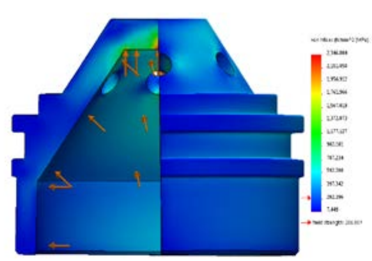
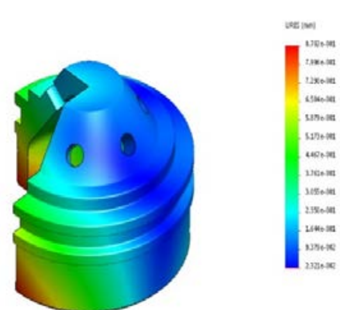


Fig 4.1 Radial Stress Distribution

The **Fig 4.1** shows axisymmetric results of the model. Maximum radial stress value is around **111.33 MPa.**



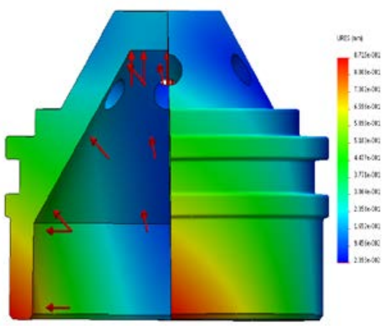


Fig 4.3 Vonmises Stress Distribution

The **Fig 4.3** shows Vonmises stress generated in the model. Maximum Vonmises stress is around **280.19 Mpa**

4.1.2 Axisymmetric Results (For modified with fillet)

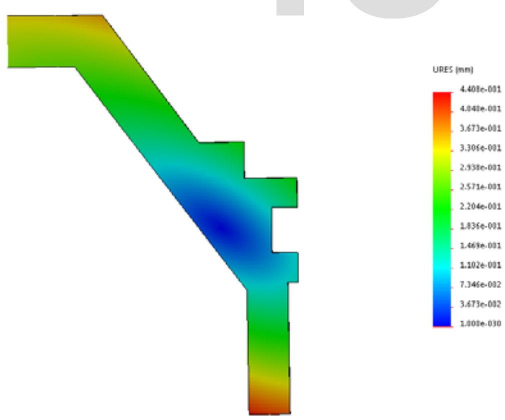
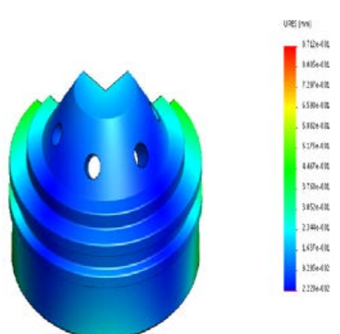


Fig 4.4 Radial Stress Distribution

For the modified geometry results are shown in **Fig 4.4**Maximum Radial stress is observed to be around **82.16 MPa.**



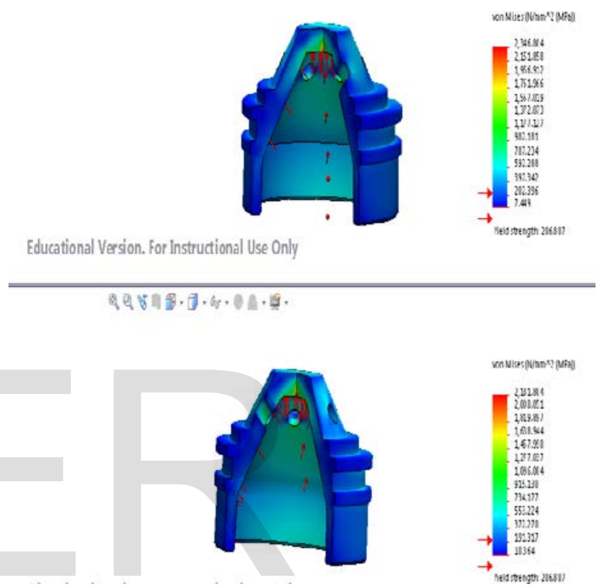


Fig 4.6 Comparison between the Vonmises stress before and after doing modification.

The **Fig 4.6** shows maximum Vonmises stress in structure. Maximum Vonmises stress observed to be around **180.51 MPa**, which is less than the initial design Vonmises stress. So, by providing fillet, stress concentration has been reduced.

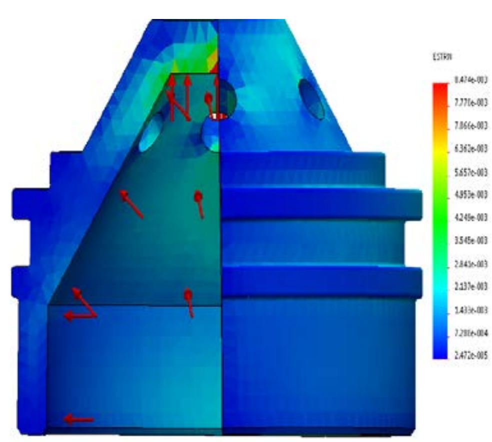


Fig 4.7 Stress concentration on holes.

The **Fig 4.7** The above analysis is carried out to find holes effect on stress concentration. The result displays that an extreme Vonmises stress of **214.034**

Table 4.1: comparison of stresses

Stresses	Before Modification	After Modification
Radial	111.33 MPa	82.16 MPa
Hoop	150.18 MPa	147.0 MPa
Vonmises	280.19 MPa	180.51 MPa

Table 4.1 shows the before & after model modification For Radial, Hoop & Vonmises stresses. In above table for after modification the stress values are less compare to before modification, because fillet is applied to after modification model hence, the stress concentration is less.

.

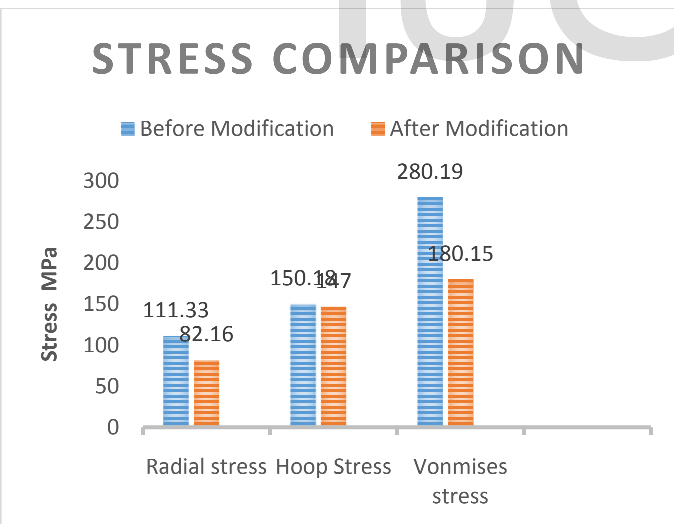


Fig 4.8 Graphical Representation

The Fig 4.8 shows, graphical representation of radial, Hoop and Vonmises stress for before & after modification. In the above graph it shows that for after modification the stress values are less compare to before modification, because the fillet has been applied for after modification model, hence the stress concentration is reduced.

4.2 Discussion

The multistage centrifugal pump delivering 85 m^3/hr. at 210 bar is used for de-scaling application in the steel plant to remove scale on the boiler plates. But the pump runs periodically and when it is not working, the water has to be by-passed at least to an extent of 15 to 20% of total flow, to protect the pump from jamming and for the safety of impeller thrust bearings. The sudden release of water from pipe lines destroys whole pressure of the system and when required, pressure will not be developed immediately. To avoid this problem, and to maintain the constant pressure at the pump Side and to deliver 20% of water to outlet, orifice system has been designed. So the orifice bypass system needs to maintain the inlet at higher pressure and outlet at negligible pressure.

So an orifice bypass system has been designed with theoretical calculations and shown in the beginning of the results in section4. The calculation is carried out for 230 bar considering the pressure changes. Total head considered is 2300 Meters. Corresponding to a discharge factor of 0.206, flow for individual orifice is calculated at 141.1 lpm and total flow is calculated for 6 orifices as 846.6 lpm. The resistance per single multi hole orifice is calculated as 2.716, and to get over all resistance in the flow path it is multiplied with number of orifices (6). In step 3 & 4 of section 4 the velocity and co-efficient of discharge calculations are carried out and values are 130 m/sec and 0.56 respectively. The final outlet velocity from orifice system is calculated based on overall resistance and total design head.

Stress analysis has been carried out on the orifice bypass system to check the safety of the problem. The axisymmetric structure has been built using solid works mixed approach, and meshing by dividing the structure using Solid works mixed approach. Then boundary conditions are applied and stress results are obtained. The results are presented for Radial, Hoop and Vonmises stresses. Initial model results show 111.33, 150.188, 280.19 MPa for Radial, Hoop and Vonmises stresses respectively which are located at the sharp corners. So the geometry has been modified to avoid sharp corners, as shown in **Fig 3.3** and further meshing and analysis is carried out using 4

The results are obtained for Vonmises which is around 214.034 MPa, showing slight stress concentration effect with 3D modeling. But this stress value is well within the allowable range of stress for the model.

The final results for velocity and pressure are represented in section 4. The results show negligible outletpressure and velocity flow which are supporting the theoretical calculations carried out.

5. Conclusions

The orifice bypass system model has been designed and analyzedfor stress concentration using Solid works FEM software. And results are summarized below.

- Theoretically calculation of Orifice bypass system has been designed with 6 steps of orifices in the flow path. For overall orifice path, velocity drop has been calculated. The theoretical calculations are shown in **section 4**
- The finite element axi-symmetric model has been built. A pressure load of 28.75 MPa is applied on the inner surface shown in **Fig 3.2**. The stress is determined and found that stress are concentrated at sharp edges.
- For the initial model stress results are obtained for Radial, Hoop and Vonmises stresses. The values of Radial, Hoop & Vonmises are 111.33, 150.188, 280.19 MPa respectively. Shown in **Table 4.1**
- For the modified model the stress concentration as reduced. The values are 82.163, 147.0 & 180.515 MPa respectively, so drop can be observed for Radial, Hoop and Vonmises stresses are shown in **Table 4.1**

- Analysis has been continued with three dimensional modelling to study the effect of holes on the geometry. The results shows, small increase in stress levels due to the presence of orifice holes. The Vonmises stress is observed to be around 214 MPa. Shown in **Fig 4.7**.The stress is mainly concentrating around the holes region.
- The result shows drop in velocity of flow and

- The following advantage after observation:

1. Newly designed orifice system can be replaced easily.
2. This orifice system is economic and has low maintenance.
3. This system is very simple in construction and it has low maintenance and less man power required.

REFERENCES

1. **G. A. Tokaty** (1994). A History and Philosophy of Fluid Mechanics (Dover).
2. **Tecmara task technology-solution**," Automatic recalculation –check valve SVV10 & SVV 12 tecmara gmbh . Im Wiesengrund 4. D-51580 Reichshof/Br. Germany. =. www.tecmara.de
3. **Eugene A. Availone and Theodore Baumeister** "Mark's Standard Handbook for Mechanical Engineers", III, (9th Ed. 1987) at page 8-126. Kingsbury bearings include levelling or equalizing plates located in a base ring. "Failure of thrust bearing in multi stage pumps"
4. **James Mc** menus school of engineering science, University of Southampton, Southampton S0 171BJ,UK
5. Hydraulic data hand book "Ingersoll-Rand data hand book", for orifice design & pressure drop calculation.