International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 04, April-2015

Structural Analysis of Non Return Control Valve using Finite Element Analysis

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Abstract **- The objective of this paper is to perform analysis and optimization of the critical component of Check Valve, its Body or Housing. Housing is mainly prone to its internal fluid pressure which passes through it. Circumferential Shell thickness of the Check valve housing is an important factor which decides the life of the valve. Wall thickness maintained should be an optimized one, else more thickness will lead to cost implication and less wall thickness will lead to failure of the vessel. Based on the ASME Standards, Check Valve body is categorized as a Pressure Vessel which contains only internal pressure. This project set out to verify finite element analysis, or FEA, when applied to Check Valves. In this study, we carried out the structure analysis of the body of the Check valve using ANSYS v14.5. Validation of the FEA results is supported by stress analysis using classical theory of mechanics. Numerically calculated stresses are compared with the FEA results and the Wall thickness is finalised based on it. Having tested three dimensional symmetric models, the preliminary conclusion is that the FEA is an extremely powerful tool when employed correctly.to the original impeller.**

Keywords - Non Return Control Valve, FEA

1. INTRODUCTION

Valves are common components of Upstream and Downstream fields in the Oil and Gas Industries. They have been playing an important role in a variety of different industries as a hydraulic device for fluid flow control. Different types of valves have applications of their own. Valves are often used for safety reasons in flow control systems. When used for flow control, the dynamics of the valve has to match the dynamics of the flow system. The relation between valve position and the pipe system would make the pressure drop and flow highly non-linear. Most of the valves are in the category Thick Walled Pressure Vessels in which the internal line pressure acts as the main loading factor.

There are several types of fluid control valves, such as, globe valve, butterfly valve, gate valve, check valves etc. Among all the flow control valves, check valve is the only valve which is simple in construction and does not require any actuation mechanism to operate.

1.1 Non Return Valves

A non-return valve can be fitted to confirm that a fluid medium flows through a pipe in the correct direction, else the pressure boundary conditions may cause reversed flow. A non-return valve allows a fluid medium to pass through in only one direction. Relatively large pressure drop is caused when the fluid flows through the non-return valve causes. This pressure drop has to be accounted when designing the system.

1.2 Swing Check Valve

A check valve [1] (as shown in Figure 1.1) is a typical valve from the family of valves non return control valves, commonly used in applications where the reversible flow of fluid highly restricted. The swing check valve works by directing flow forces to move the disc from the closed position to the fully open position. It travels in a sweeping arc motion against the hinge-stop inside of the valve body as shown in Figure 1.2. Due to the weight and center-of-gravity location of the disc and hinge-arm assembly, the valve will return to the closed position when the flow is interrupted or reversed. External counterweights are mounted on the hinge pin. They are sometimes used to inflate or deflate the reaction time and speed of the disc returning to its original position.

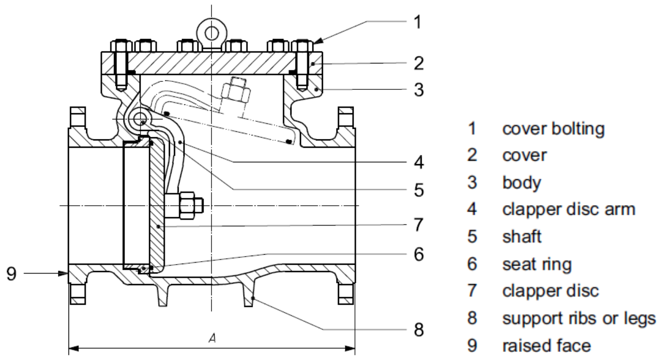

Fig. 1.1 Check Valve and its Parts

1.3 Working Principle Of Check Valves

The main function of check valves is to close upon forward-flow stoppage and prevent or minimize the development of reverse flow. This function helps to protect pumps and systems from damage caused by reverse flow. Check valves are also used to isolate areas of plants, such as nuclear power plants, from over-pressurizing or being contaminated.

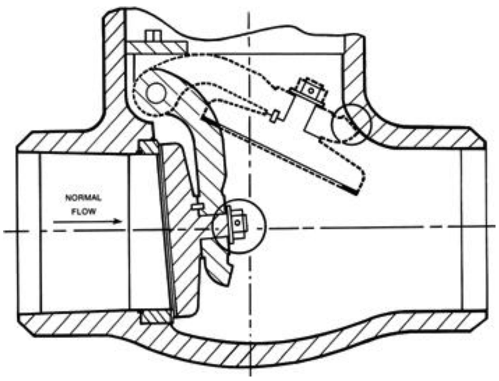

Fig. 1.2 Working of a Swing Check Valve

Among several types of check valve designs, Swing Check Valves are the simplest in construction and often used in industry.

Swing check valve discs are not stable unless they are in systems with steady flow and are in the fully open position as shown in Figure 1.2. As the disc/hinge assembly center of rotation is a fair distance from its hinge pivot point, it will take a relatively long period of time for swing check valves to close when the flow is stopped or reversed. During this period, reverse-flow forces will experience a large increase in energy from flow and pressure buildup. This situation can cause high-energy water hammer when the disc slams onto the seat.

2. REVIEW OF LITERATURE

Provoost (1983) [2] described a decisive characteristic of un-damped non-return check valves which is the correlation between the maximum back flow velocity through the valve and the deceleration of the flow. The maximum reversed velocity is said to almost only depend on the deceleration, given a stationary initial condition and a constant deceleration. This is an important dynamic characteristic since once the maximum velocity is obtained, the Joukowsky equation can be used to predict the maximum pressure rise that can occur on both sides of the valve, which also gives an idea of the unsteady phenomena of the valve after closure can be obtained.

In a work carried out by Li & Liou [3], a fundamental approach using that equation to model the movement of the swing check valve disc is defined as “The net torque imposed on the disc is compared to the time rate of change of the angular momentum of the rotating mass”. The torques applied are those imposed by the weight of the disc, buoyancy, friction at the hinge pin, any external load such as a spring or a lever, and the torque caused by the flow, the so called Hydraulic torque.

Another master thesis by Turesson (2011) [4] investigates the modelling of check valves in the 1D-code Relap5 and compares results of simulations of closure of a valve with CFD-simulations of the same case. It turns out that all 1D-models under predicts closure time compared to CFD-simulations. The probable reason is said to be that the 1D model have got deficiencies modeling the contribution of the moving disc. Also a thorough investigation of the CFD-settings is performed. Turbulence model, mesh density, the usage of prismatic layers in the boundary layer, parallelization of computations and time step are investigated.

Also Thorley [5] and Koetzier, H., Kruisbrink [6] studied the dynamic behavior of the clapper arm in the check valves.

Several researches have been done on check valves as stated earlier. However, such researches mainly focused on the fluid analysis of check valve rather than the structural analysis with the view of check valve as pressure vessel. From this reviews of research papers it is seen that there is a scope for structural analysis of check valve body by using methodologies like finite element analysis.

3. SWING CHECK VALVES AS PRESSURE VESSELS

Check valve [7] as a pressure vessel in-service poses extreme potential danger due to the high pressure and varying operating temperature; hence there should be no complacency about the risks. Unfortunately, pressure vessels accidents happen much more than they should.

Due to the differential operating pressure of pressure vessels, they are potentially dangerous and accidents involving pressure vessels can be deadly and poses lethal dangers when vessels contents are flammable/explosive, toxic or reactive.

Stress is the internal resistance or counterforce of a material to the distorting effects of an external force or load, which depends on the direction of applied load as well as on the plane it acts. At a given plane, there are both normal and shear stresses. However, there are planes within a structural component subjected to mechanical or thermal loads that contain no shear stress. Such planes are principal planes, the directions normal to those planes are principal directions and the stresses are principal stresses.

4. DESIGN OF PRESSURE VESSELS

In general, pressure vessels designed in accordance with the ASME Code, Section VIII, Division 1 [8], are designed by rules and does not require any detailed evaluation of all stresses. It is recognized that all highly localized and secondary bending stresses will exist but they are allowed for by use of a higher safety factor and design rules for details. It is required, however, that all loadings (the forces applied to a vessel or its structural attachments) must be considered.

When the wall thickness of a cylindrical pressure vessel is about one-twentieth, or less, of its radius then it is called as Thin Walled Vessels; if it is greater, then it is called Thick Walled Pressure Vessel. Check Valve is a thick walled pressure vessel.

4.1 DESIGN PARAMETERS OF SWING CHECK VALVE

The design parameters of the swing check valve are listed below in the Table 4.1.

Details	Values
Size of the Valve	10”
Pressure Class	600
Maximum Allowable Working Pressure (MAWP)	10.0 MPa
Maximum Allowable Working Temperature (MAWT)	-50^{o}F

Table 4.1 Design Parameters of the Check Valve

4.2 MATERIAL PROPERTIES OF THE CHECK VALVE HOUSING

Material of valve body	-	A352 Grade LCC
Tensile Yield Strength (YS)	-	275.0 MPa
The tensile ultimate strength	-	515.0 MPa
Young’s Modulus	-	$2.0 * 10^5$ MPa
Poisson’s Ratio	-	0.3

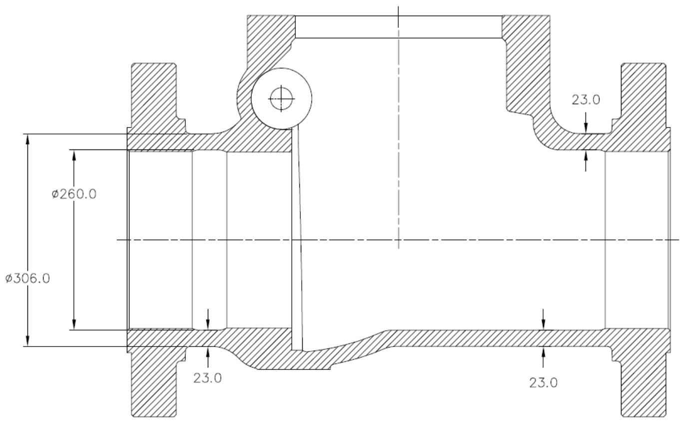

Fig. 4.1 Check Valve Housing

Check Valve with line pressure is considered as the scenario of Shell under Internal Pressure. Wall thickness of the shell is under prime concern and it is to be calculated.

As per the standard ASME B16.34 [9] Table A-1, Minimum Inside Diameter of a 10” CL. 600 Valve body is 247.7 mm. The maintained inside diameter is 260.0 mm (as shown in Figure 4.1) to meet the standard requirements and customer requirements.

Inlet, Outlet and Top flanges of the check valve are designed based on the standard ASME B16.5 (Pipe Flanges and Flanged Fittings)

Also based on the standard ASME B16.34 Table VI – 1, Basis Equations for Minimum Wall Thickness for 10” Cl. 600 for diameter ranging $50 < d \leq 1300$ is

$$tm\,(600) = 0.06777 * ID + 2.54$$

where ID, actual inside diameter of the valve body

$$tm\,(300) = 0.06777 * 260.0 + 2.54$$

$$tm\,(300) = 20\ \text{mm}$$

So the minimum wall thickness to be maintained as per ASME B16.34 is

$$Ts = t_m\,(300) + CA$$

where CA is the Corrosion Allowance of 3.0 mm per side.

Hence Ts = 23.0 mm

Subsection UG 27 of ASME Section VIII Division 1 calculates the thickness of the shell under internal pressure using the below formula.

Body Shell Thickness = ((P * R) / (S*E – 0.6*P)) + CA

Where,

P is the Maximum Pressure applied in the body,

ie Hydrostatic Test Pressure (TP)	=	15.0 MPa
R is the Radius of the Shell	=	130.0 mm
S is Maximum Allowable Stress	=	229.2 MPa
E is joint efficiency of cylindrical shells	=	1
CA is the Corrosion Allowance	=	3 mm

Body Shell Thickness (BST) is calculated using the above equation and the result is BST = 17.0 mm.

5. INTRODUCTION TO FINITE ELEMENT ANALYSIS

Finite element analysis (FEA) [10] is a numerical method that models a region by dividing it into small discrete elements composed of interconnecting nodes. Finite element analysis obtains the solution to the model by determining the behavior of each element separately, then combining the individual effects to predict the behavior of the entire model. The interconnecting nodes of the elements make the solution of one element dependent on another, meaning that to reach an accurate solution; FEA must solve each element several times, possibly thousands of times, to reach a solution.

5.1 MODEL GEOMETRY

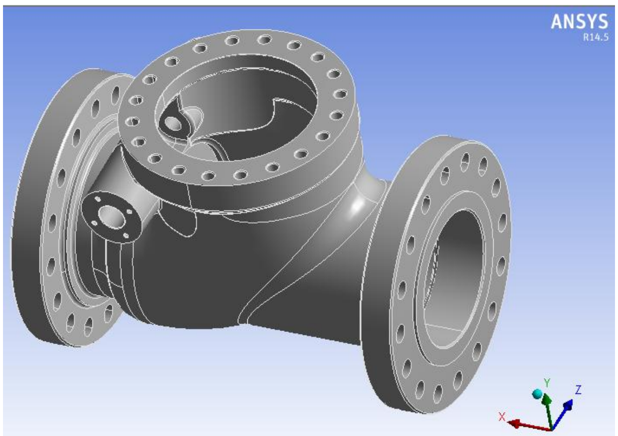

Fig. 5.1 Check Valve Housing

International Journal of Engineering Research & Technology (IJERT)
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Finite element analysis can be used with either 2-D or 3-D models. 3-D models generally offer a more accurate analysis as they include all three planes of the physical world. A 3-D model is also composed of a great deal more nodes and elements as well, drastically increasing solution time.

In this work, the 3D model of the check valve housing has been created using the 3D modeling sofware Pro/E Creo as shown in Figure 5.1.

5.2 PREPROCESSING

FEA package ANSYS R14.5 [11] is used in this analysis. Meshing has been done by using the method of hex dominant mesh. These types of mesh control create a free hex dominant mesh. It is useful for meshing bodies that cannot be swept. It is highly recommended for meshing bodies with large interior volumes. And it is not recommended for thin or highly complex shapes.

Object Name	*Mesh*
State	Solved
Sizing	
Use Advanced Size Function	On: Curvature
Smoothing	Medium
Transition	Fast
Curvature Normal Angle	36.0 °
Min Size	Default (0.160620 mm)
Object Name	*Hex Dominant Method*
State	Fully Defined
Method	Hex Dominant
Free Face Mesh Type	Quad/Tri
Nodes	579565
Elements	183967

Table 5.1 Meshing Details

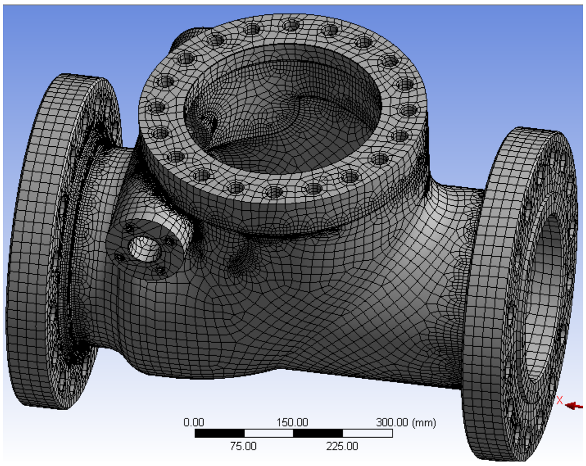

Fig. 5.2 Meshed model of the Valve Housing

5.3 BOUNDARY CONDITIONS

The check valve housing is fixed on both the flange area as depicted in Figure 5.3, because the valve flanges are fastened with the pipe flanges. Thus all the degrees of freedom of the valve housing are arrested.

Normally during working condition, the maximum allowable working pressure of the fluid medium 10.0 MPa acts as the load. But as per the standard ISA-75.19.01 [12], during hydrostatic shell test, the load is calculated by multiplying the 38°C (100°F) working pressures by 1.5. So the hydrostatic test pressure of 15.0 MPa is applied to the internals of housing as shown in the Figure 5.3.

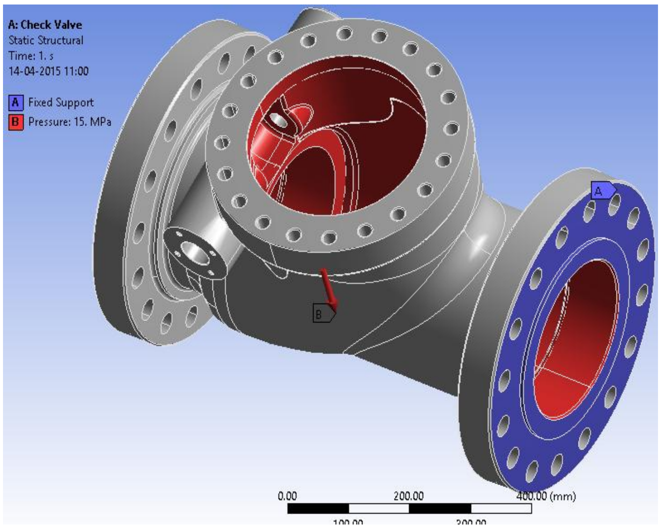

Fig. 5.3 Boundary Conditions in the Housing

5.4 RESULTS

After applying the boundary conditions, the structural analysis is carried out. The results of the check valve housing analysis are shown in the Figure 5.4 to Figure 5.7.

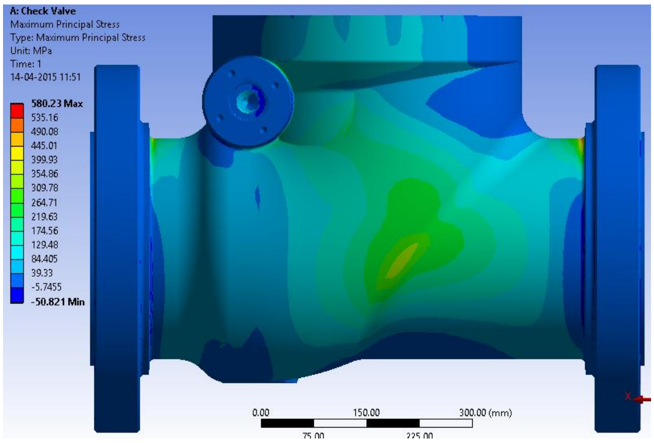

Fig. 5.4 Equivalent Stress Plot 1

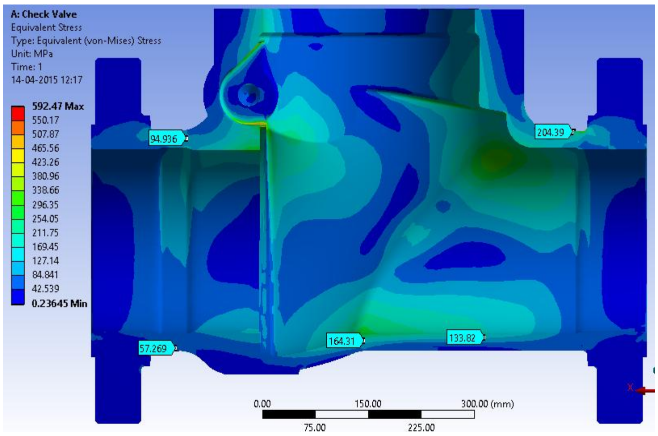

Fig. 5.5 Equivalent Stress Plot 2 (Sectional View)

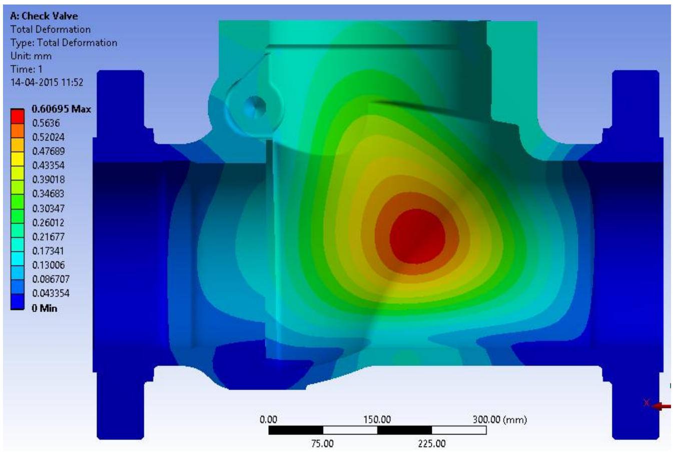

Fig. 5.6 Total Deformation Plot (Sectional View)

From the Figure 5.4 and Figure 5.5 the equivalent stress [13] acting at minimum cross sectional area or at minimum wall thickness region lies in the range of 42 MPa to 211 MPa. This is within the limit as described by the ASME Section VIII, Division 2 [14], Appendix 4 ie. Design-allowable stresses, S_T, is the maximum allowable general primary membrane stress intensity at hydrostatic test pressure,

$$S_T = 5/6 * \text{Yield Strength} = 229 \text{ MPa}$$

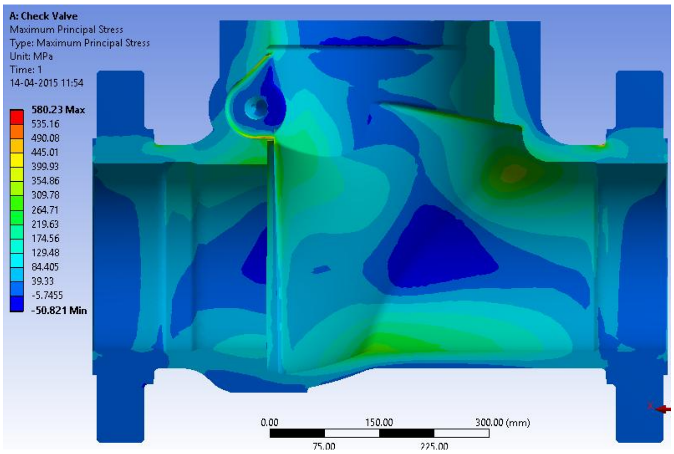

Fig. 5.7 Max Principal Stress (Sectional View)

Also the deflection and the maximum principal stress acting at the minimum wall thickness region are maximum of 0.2 mm and 219 MPa respectively, which are within the safety limit. And the calculated Factor of Safety is more than 2 in all states like membrane stress intensity and maximum stress intensity.

As per ASME B16.34 calculated wall thickness Ts is 23 mm. And as per UG 27 of ASME Section VIII Division 1 calculated wall thickness is 17 mm. So the maximum value based on the ASME B16.34 which is taken into consideration and the maintained wall thickness is 23.0 mm which satisfies the pressure vessel design requirements.

Hence the design is safe and will withstand the maximum hydrostatic test pressure of 15.0 MPa efficiently. Hence the wall thickness of 23.0 mm is an optimized value per the standards and the theoretical calculations as well as structural analysis. And the check valve housing is with the high degree of structural stability

6. CONCLUSION

Pressure vessels are the most important parts of any Oil and Gas Plant. And the design should satisfy the criteria's given in the international code, ASME Section VIII Division 1 and Division 2. They have to be designed carefully to cope with operating temperature and pressure.

This work presented critical design analysis of stress development using 3D CAD models of check valve housing and finite element engineering simulation of various stress and deformation tests at high pressure.

Theoretical calculated values by using different formulas quoted from various standards are very close to that of the values obtained from ANSYS analysis which is suitable for pressure vessels.

FEA is a powerful tool in analyzing the various structures and the results provided by ANSYS v14.5 proved once again its reliability. The current capabilities of FE software on desktop computers provide pressure vessel design engineers with the ability to employ FE analysis on a nearly routine basis.

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