Design Methodology of Pressure Vessel Nozzle

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Abstract— Nozzle is the most essential component of the pressure vessel. The main objective of this work is to design nozzle for pressure vessel according to ASME code. The design methodology of components of the nozzle like neck, reinforcement, and flange has been explained through this work. Design of 32” nozzle has been carried out according to required flow rate of hydrocarbon fluid based on internal pressure and temperature and external loading. This work gives the analytical result for all the component of pressure vessel. It also shows effect of thickness of Gasket and effect of no of bolt of weld neck flange on seating pressure.

Key words: Nozzle, Flange, Pressure Vessel, ASME BPVC Code, Nozzle Reinforcement

I. INTRODUCTION

Nozzle is one type of opening in pressure vessel. A pressure vessel cannot be imagined without a nozzle. It is a device designed to control the direction or characteristics of a fluid flow, different types of nozzles used are inlet, outlet, manways, sump, indicators, and safety valves. In the nozzle there are components like neck, reinforcement pad, flanges and their methodology is explain in this paper. All this design methodology are done based on ASME boiler and pressure vessel code, pressure vessel design manual Dennis moss and some research papers. For pressure vessel there are various construction available for nozzle according to code are 1. Nozzle with pipe & Flange, 2. Nozzle with pipe & reinforcement 3.long weld neck type nozzle, 4. Self-reinforcement nozzle. A neck is small pipe used in determining diameter of nozzle. The annular circular plate is provided around the opening in the form of pad known as reinforcement pad. A piping flange is device used to connect piping and components in piping system by use of bolted connections and gasket. Result contain complete calculation of 32” nozzle for Knock out Drum and Some Graphs regarding to flange and stress.

II. BASIC CONSIDERATION IS CONCERN BEFORE DESIGNING OF PRESSURE VESSEL NOZZLE

Before the start the design of the nozzle keep following consideration first,
1) Location
2) Material Selection
3) Maximum allowable nozzle load

A. Location

Nozzle can be attach to different components of pressure vessel i.e. shell, head and sump at various positions. Basic general categories of head are hemispherical, tori spherical and ellipsoidal. For all the load categories, high stresses occurred at the vessel-nozzle junction as there is a severe geometric discontinuity. At the knuckle region of head high compressive stress occurred so design of nozzle in radial direction must be careful at knuckle region. Hoop stress in the knuckle become compressive when the ratio of R/h exceeds 1.42, where R is radius of head and h is depth of head [1]. The best solution for locating the nozzle in vessel head is to place it in the axis of symmetry of the vessel axis. When the nozzle is placed at various distance from the axis of symmetry of the vessel the high stress occurred in nozzle junction. So the limiting distance from axis of symmetry is 0 to 0.8R where R is radius of the vessel [2]. In pressure vessel heads the flexibility of nozzle has to be investigated. The flexibility of nozzles play an important role in both piping flexibility analysis and vibration analysis .flexibility of nozzles in ellipsoidal heads can be appreciably higher than those in hemispherical heads for same vessel diameters [9].

Fig. 1: Nozzle Locations, [1]
three directions: forces perpendicular to axis of the vessel and parallel to the nozzle axis, perpendicular to the axis of the vessel and perpendicular to the nozzle axis and forces parallel to the axis of the vessel & perpendicular to the nozzle axis [3]. Stress analysis of nozzle on cylindrical vessel for external load is carried by comparing the experimental results for different models, with parameters in the ranges: $50 \leq \frac{D}{T} \leq 2530$, $0.01 \leq \frac{d}{D} \leq 0.5$ and $6.5 \leq \frac{d}{t} \leq 240$. Hence, the calculation should be useful for an improvement in nozzle design [4].

B. Material Selection

The material selection of nozzle is based on design conditions like internal pressure, temperature, external loadings and allowable stresses. Generally Allowable Stress and properties of material for nozzle is selected from ASME BPVC Section II Part D.

C. Maximum Allowable Nozzle Load

It establishes the minimum criteria for design by providing the maximum allowable nozzle load by size and class of flange rating. This is an alternative work process to determining each individual nozzle load after the piping system is designed. This procedure eliminates much of the late design changes that occur when late date on nozzle loads impact either the design of the piping system or the stresses in the vessel shell. This procedure does not take into account any of the vessel parameters such as diameter, thickness, material, temperature, allowable stress, internal pressure, etc [1].

III. METHODOLOGY

A. Nozzle Neck Design

Nozzle Neck methodology says, first find the diameter of nozzle from the flow rate of hydrocarbon fluid $Q=AV$. Here, the velocity of fluid must be less than 1 m/s and then evaluate the time required for filling the vessel. For Nozzle neck design ASME Section VIII, UG-45 is used to find minimum nozzle neck thickness [5]. Following flowchart determine the steps to find the minimum wall thickness of nozzle neck.

Nomenclature
1) $t_a$ - minimum neck thickness required for internal & External using UG-27&UG-28 respectively
2) $t_{b1}$ - thickness under internal pressure UG-16(b)
3) $t_{b2}$ - thickness under external pressure UG-16(b)
4) $t_{b3}$ - the thickness given in Table UG-45
5) $t_{UG-45}$ = minimum wall thickness of nozzle necks

Formulas to be used as per ASME UG-45
1) $t_{UG-45} = t_a$
2) $t_b = \text{min}(t_{b3}, \text{max}(t_{b1}, t_{b2}))$
3) $t_{UG-45} = \text{max}(t_a, t_b)$

B. Nozzle Reinforcement

The basic principle of the area compensation method is when the opening is cut in the pressure vessel, an area is removed from the shell and head. It must be reinforced by an equal amount of area near the opening. The area removed should be equal to the area added. The area is added by providing a reinforcing pad in the form of annular circular plate around the opening. Here we are considering cross-sectional area in the form of a rectangular strip. It is not the compensation of volume of metal that has been cut due to the opening by means of the reinforcing pad.
opening in cylindrical and conical shell under internal pressure [5].

The methods used to design openings in cylindrical shells under internal pressure reinforced by set-on nozzles. The set-on nozzle is nozzle which is outside the weld that penetrates through the nozzle wall so the height of the weld is equal to the thickness of nozzle wall. The methods used to design openings in cylindrical shells under internal pressure reinforced by set-on nozzles are EN 13445-3: Unfired Pressure Vessel, Part 3: Design (European standard), ASME pressure vessel code (American standard) and PD 5500: specification for unfired fusion welded pressure vessels (British Standard). It can be observed that ASME design method mostly gives the smallest values of permissible design pressure, which proves that ASME provides greatest design safety but require largest amount of reinforcement material [6]. The design of reinforcement pad in nozzle is based on some limit in reinforcement pad are as follows:

1) This above limit appears to be contradictory to the provisions for isolated circular openings which states that the reinforcement is not required if: \( d \leq 0.2 \sqrt{RT} \) where, \( R \) =radius of shell or head, \( T \) =thickness of shell or head, \( d \) =diameter of opening [7].
2) The analytical solution of two orthogonally intersecting cylindrical shell is worked out and the rest are applicable to reinforcement design up to \( d/D=0.8 \), where \( D \) =shell or head diameter, \( d \) =nozzle diameter [8].
3) The guidelines based on ASME Sec VIII Div. 1 are used to design reinforcement pad according to UG-36 to UG-43 ,UW-14 to UW-16 & Appendix 1 to 7 are limits ,strength, width, thickness , openings in head and shell ligaments ,multiple openings ,plane of reinforcement [5].

C. Flange

A pipe flange connects piping and components in piping system by use of bolted connections and gasket. Flanged joints are the popular for connection piping system and this piping system have fluid or gas under pressure. There are various types of flanges such as weld-neck flanges, slip on flange, lap-joint flange, threaded flange and socket weld flange.

There are different flanges available having various pressure capacity. Lap joint type, slip on type and thread type flange have low pressure capacity and socket weld type & weld neck type flange have high pressure capacity also blind type flange is used when there is very high pressure capacity. All the flanges have different applications. Blind type flange is used for closing pipes and testing flow pressure. When it is necessary to attach flange without welding then threaded type flange is used. Weld neck flange is used in petrochemical industry and due to its best stress distribution it is used in most of the cases. Among all type flanges slip on flange have low installation cost and simply assemble but it is available in only small size. Weld-neck flanges is most suitable for hydrocarbon fluid because it bored to match the inside diameter of the mating pipe or fitting so there will be no restriction of product flow. It is integral type flange. It prevents turbulence at the joint and reduces erosion. They also provide excellent stress distribution through the tapered hub and are easily radiographed for flow detection. Weld-neck flange is used for high pressures and extreme temperatures.

![Fig. 3: Flange Design Procedure](image)

Flange design is governed by two condition Gasket seating force and Hydrostatic end force. High pressure flanges and low pressure flange are design with reference to hydrostatic force and seating force respectively. Flange Methodology say, first select appropriate flange size according to pipe or nozzle size from ASME-B16.1 Cast iron pipe flanges and flange fittings, B16.5 – Pipe flanges & flange fittings, B16.47 – Large diameter steel flanges or B16.42 – Ductile iron pipe flanges & flange fittings [1]. Now gathering the information about design conditions, allowable stress for bolt, flange and gasket. By selecting the geometry of flanges from raised face type, flat face type and groove face type. Follow the procedure to find effective gasket width and diameter of location of gasket load reaction from the ASME Section VIII DIV 1 mandatory Appendix 2 - rules for bolted flange connections with ring type Gaskets [5]. Flange design is the process of determining the load applied at the gasket due to seating and operation conditions. Find the bolt load for seating and operation condition and select the larger bolt load for further design.

![Fig. 4: Number of Bolts vs Seating Stress](image)

Basically number of bolts and spacing between two bolts mainly effect on leakage and stresses. When bolt spacing is small the leakage rate will increase as magnitude of pressure increase and when bolt spacing is large leakage process can be divided into two ways; the leakage rate will increase linearly up to critical pressure and drop sharply after critical pressure. Number of bolts affect the stress acting on flanges [12]. Fig 4 shows with increase in number of bolts N which increases the longitudinal stress \( S_h \), Radial Stress \( S_r \) and \( (S_h+S_r) \).

Here, Flange material is SA 105 and size 32” and take standard B16.47 #300 Weld neck flange. Gasket Material is SS308 Graphite filler spiral wound and dimensions are standard as ASME 32” gasket. Design
pressure is 3.6 bar and temperature is 350°C. In these experiment after 50 numbers of bolts the flange will fail Gasket stress.

Flange deflection angle affect the tightness of the flange connection system and greater the flange deflection angle, more uneven distribution of gasket compression stress along the width of the gasket. So the leakage rate increase. By ASME deflection angle of the flange should be limited to 0.3°. The increase in flange rotation is found beneficial in terms of tightness, noting that the gasket apparently was not damaged due to the high local contact stress at the outside diameter [11].

D. Gasket Details

Generally, the most common types of gasket are cut gasket, Spiral wound gasket, API ring joints type gasket, PTFE envelop type gasket and corrugated metallic type gasket for piping and flanges.

The selection of particular gasket is based on temperature & pressure of fluid and compatibility of gasket material for the flow. Basically, temperature is the first concern for the selection of gasket. Gasket must withstand the system temperature without any serious impairment of its performance properties and also cryogenic temperatures must be consider in some situations. The most important information under application is the gasket type is selected based on the flange type (raised face type, flat face type and groove face type, ring joint type etc.) used. Gasket material must be chemically resist the fluid to prevent any serious impairment of its physical properties.

The gasket must have sufficient strength to resist crushing under applied load. Gasket must be strong enough to resist internal pressure. Thinner gasket is used in the flange arrangement but thick enough to compensate the unevenness of the flange surface, surface finish, parallelism, rigidity etc.

Gasket can withstand higher bolt load with thinner gasket. Also the gasket area will be lower which will be exposed to attack from the internal pressure and aggressive media. So gasket must be as thin as possible. The gasket performance decreases as material thickness.

These both graph shows the effect of thickness of gasket on longitudinal hub stress, Radial Flange Stress and Tangential Flange Stress. Values of stresses and thickness of gasket with regarding to 32” nozzle and gasket material is spiral wound SS304 Graphite Filler. Pressure and temperature are as mention in earlier graph. Gasket and flange dimension mentions earlier too.

IV. DESIGN RESULT

A. Design Condition


B. Nozzle neck result as per ASME Section VIII Div 1 UG-45

Nozzle Neck Thickness = 11.34 mm ≅ 12 mm

C. Nozzle reinforcement pad area calculation as per UG 37

Required reinforcement area = 5257.8 mm²

Given area by reinforcement pad =16385.1 mm²

D. Flange Stress according to ASME Mandatory Appendix-2

Operating Stresses,

Longitudinal Hub Stress = 20.59 N / mm²
Radial Flange Stress = 11.11 N / mm²
Tangential Flange Stress = 5.02 N / mm²
Average of Longitudinal Hub Stress and Radial Flange Stress = 13.84 N / mm²
Average of Longitudinal Hub Stress and Tangential Flange Stress = 12.81 N / mm²
Seating Stress,

Longitudinal Hub Stress = 142.35 N / mm²
Radial Flange Stress = 76.78 N / mm²
Tangential Flange Stress = 34.73 N / mm²
Average of Longitudinal Hub Stress and Radial Flange Stress = 109.56 N / mm²
Average of Longitudinal Hub Stress and Tangential Flange Stress = 88.54 N / mm²

V. CONCLUSION

From this paper we conclude the methodology for Nozzle design through ASME code. From ASME code the nozzle location selection is been easily justified from this paper. The material selection for nozzle components can be obtained from ASME code section II. Weld neck flange is selected based on design pressure and temperature. From the above methodology of Nozzle, Result for 32” Nozzle has been obtained for Pressure Vessel.
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REFERENCES