PARAMETRIC STUDY AND COMPARATIVE ANALYSIS OF FLANGES IN HEAT EXCHANGER APPLICATIONS

By:

SOUMIL VYAS (20MMCC16)

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

YEAR: 2021-2022

PARAMETRIC STUDY AND COMPARATIVE ANALYSIS OF FLANGES IN HEAT EXCHANGER APPLICATIONS

Submitted in partial fulfillment of the requirements for the award of degree of

MASTERS OF TECHNOLOGY

IN

MECHANICAL ENGINEERING (CAD/CAM)

Submitted by:

SOUMIL VYAS
(20MMCC16)

Guided By

Prof. K.M. Patel

MECHANICAL ENGINEERING DEPARTMENT

SCHOOL OF ENGINEERING,

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

YEAR: 2021-2022

Declaration

This is to certify that

- The thesis comprises my original work towards the degree of Masters of Technology in CAD CAM at Nirma University and has not been submitted elsewhere for degree.
- Due acknowledgement has been made in the text to all other material used.

Sign:

Name:

Roll No:

Undertaking for Originality of the Work

I, Soumil Vyas (20MMCC16), give undertaking that the Major Project entitled ("Parametric Study and Comparative Analysis of Flanges in Heat Exchanger Applications ") submitted by me, towards the partial fulfillment of the requirements for the degree of Masters of Technology in CAD CAM of Nirma University, Ahmedabad, is the original work carried out by me. I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

Signature of Student

Date:

Place: Nirma University, Ahmedabad

Endorsed by

(Signature of Guide)

Certificate

This is to certify that the Major Project Report entitled "Parametric Study and Comparative Analysis of Flanges in Heat Exchanger Applications" submitted by Mr. Soumil Vyas (20MMCC16), towards the partial fulfillment of the requirements for the award of Degree of Master of Technology in Mechanical Engineering (CAD/CAM) of School of Engineering, Nirma University, Ahmadabad is the record of work carried out by him under our supervision and guidance. In our opinion, the sub- mitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

Date:

Guide Name: Dr. Kaushik Patel,
Head and Professor,
Department of Mechanical Engineering,
Institute of Technology
Nirma University, Ahmedabad.

Industry Guide Name: Safwan Chaklasiya
Sr. Design Engineer,
L&T Heavy Engineering,
Ranoli, Vadodara.

Dr. Kaushik Patel
Head and Professor,
Department of Mechanical Engineering
Institute of Technology
Nirma University, Ahmedabad.

Approval Sheet

The Project entitled **Parametric Study and Comparative Analysis of Flanges in Heat Exchanger Applications** by **Soumil Vyas (20MMCC16)** is approved for the degree of Masters of Technology in CAD CAM.

Examiners

____

____

____

Date: _____________________

Place: ____________________

Acknowledgments

It gives me immense pleasure to express my sincere gratitude to the people who have been a source of constant motivation and relentless support in my endeavor. I would like to express my deepest gratitude to my project guide internal Prof. K.M Patel as well as industrial guide Mr. Safwan Chaklasiya, who have done their best to make me understand everything which I came across during my project work with utmost patience and in the best possible way. I feel myself very lucky to have meet with these two and got opportunity to work under them. I am very much indebted to Prof. Jatin Dave and Prof. B.A Modi for their valuable support, guidance and constant encouragement throughout my study. I am also very much thankful to Mr. Safwan Chaklasiya to constantly show the areas where I am lacking and encouraged to reach to a satisfactory level. I would like to give my special thanks to Mr. Hardic Shah (DGM, Design Department, L&T Heavy Engineering, Ranoli, Vadodara) for giving me the opportunity to carry out my project work from the esteemed organization and like to thank the organization as well. I cannot imagine any better place than this for my project work. I am thankful to all its employee who are always eager to help in the best possible way they can. Last but not the least I am very much thankful to Mr. Chintan Hingoo, Mr. Ashish Joshi, Mr. Nikunj Patel, Mr. Harikrushna Barot, Mr. Nirav Mehta, Mr. Rajat Gupta (Design Department, L&T Heavy Engineering, Ranoli, Vadodara) for providing me continuous guidance to carry out my project.

Name and signature of student

Date:

Place:

Abstract

In today’s world, due to constantly depleting resources and cut- throat competition, it becomes very much essential to optimize our resources in best possible way, such that not only the manufactured object can satisfy all its operational parameter and have long life but also it should be manufactured with right quantity of material. In a pressure vessel, the flange plays an important role in providing leak- proof joints as well as connections between two pieces of pipe, pipes and nozzles. Flange joint permits disassembly and removal for maintenance purposes. So, the design of Flanges must be very accurate in order to avoid leakages or failure of joints during the operation of the equipment. Appropriate guidelines for the design of Flanges are provided in the ASME codes (The American Society of Mechanical Engineers). The purpose of this project is to design a Flange which has very critical design parameters and to ensure that while in operation, the flange doesn’t encounter leakages or any other types of failure. Along with the safety factors, the flange should ensure correct structural strength and also withstand high stresses generated during the processes which are taking place in the pressure vessel. For this, the target flange in the equipment is designed according to ASME Section VIII Div.1 Mandatory Appendix 2 and ASME PCC-1. In addition to that, Finite Element Analysis and Parametric study is conducted in order to get the optimized flange design.

Key words: Flange, Weld neck Flanges, ASME Section VIII Div.1, ASME PCC-1, Finite Element Analysis, Parametric Study.

Table of contents **Page No.**

List of Figures

Page No.

List of Tables **Page No.**

CHAPTER 1

Introduction

1.1 About L&T Heavy Engineering, Vadodara:

Vadodara Heavy Engineering Works is an organization which is involved in the manufacturing of equipment and systems in a unique facility which involves use of exotic materials such as SS, DSS, Urea grade Steel, Inconel, Titanium, Zirconium etc.

The Facility caters to a number of industries like Oil & Gas, Fertilizer, Petrochemicals, Specialty Chemicals, Nuclear Power. This facility consists of ASME approved workshops with U, U2, N, NA, NS, NPT & MO stamps and they are equipped to supply highly critical equipment & assemblies, that match international standards. VHEW is constantly delivering to the industries at a global level by being a trusted supplier for the reactor internals.

Vadodara Heavy Engineering Works has extensive experience and expertise in the manufacturing and supply of Ammonia Converter Baskets, Heat Exchangers & other critical equipment for refinery & fertilizer industries. The Facility manufactures and supplies Dry Shielded Canisters, thereby becoming India's top exporter of nuclear power equipment. This unit is also contributes & partners to many strategic programmes for the Government of India.

The unit's strength lies in a highly skilled workforce and adaptation of innovative methods and processes. Currently there are four patents under process from this facility.

VHEW's core strength resides in its capability to engineer custom-built machines that suit niche products. Some examples of custom-built machines are high speed CNC turn-mill center, Vacuum sputtering for 0.6μ aluminum coating on carbon epoxy shells, Robotic CMM machine and LASER tracker.

A wide variation of welding processes is executed which comprises of mechanized TIG, GTAW, ESSC, resistance spot welding, PTAW, automatic tube-to-tube sheet welding, GMAW, FCAW, submerged arc and automatic nozzle welding.

1.2 What are Flanges?

Flanges are often used to dismantle the piping joints. The use of flanges is mostly at equipment's, valves and specialties. Flanges at regular intervals offer good features for regular maintenance of certain pipelines. The components that make a flange joint which are usually independent but are mostly interconnected which are: bolts, gaskets and flange. An appropriate combination of all the three components provides a good leak-proof joint.

Figure 1.1 Flanges (Reference Website: shutterstock.com)

1.3 Function of Flanges in Heat Exchanger:

Flanges are used to provide Leak proof connections between two pieces of pipe, pipes and nozzles. Flange joint permits disassembly and removal or maintenance purposes. Flange joints are utilized for the connection of pipes and instruments to the heat exchangers, as well for other pressure vessels it can be used as manway covers, and for vessel heads that can be removed for easy access [11]. Often the flanges are used in vessel body in form of sections for easy transportation and when division of pressure vessel is required.

1.4 Classification of Flanges based on method of Attachment:

Flanges can be classified based on the method of attachment:

1) Slip On Flange:

An attachment of two fillet welds, one inside and outside of the flange is used for Slip-on Flange. The strength of a Slip-On Flange is about two-thirds of that of a Weld-neck flange and the fatigue life is also about one-third of the latter.[9] Generally, a hub is provided in these flanges and they are of forged construction. Sometimes, these flanges are fabricated from plates and are not provided with the hub. A combination of flange and elbow or a tee is impossible as these fittings do not have straight end that can completely slid on the Slip-On Flanges.

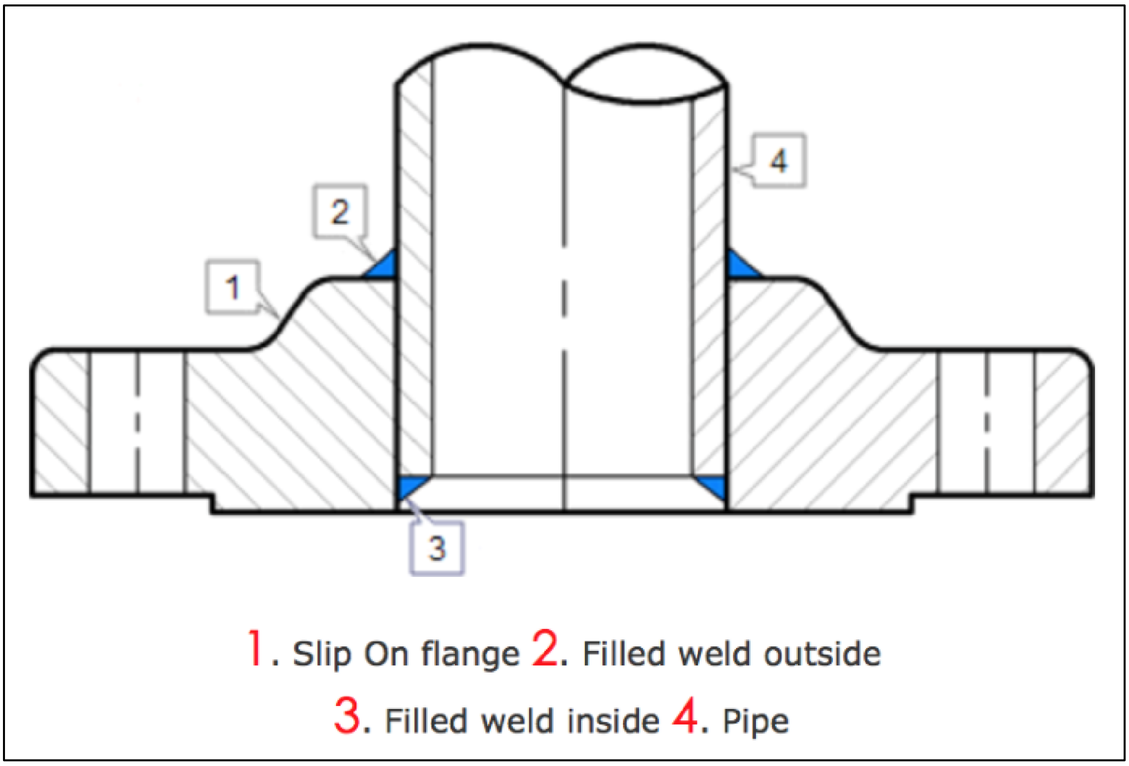


Figure 1.2 Section view of Slip-on Flange (Image Reference. engineeringlearn.com)

2) Socket Weld Flanges:

The Socket weld flanges are bonded by single fillet weld, which is on the outside, and which are not suggested for severe conditions. These are mostly used for small bore lines. Static strength of these flanges is same as that of Slip-on flanges, but the fatigue strength is 50% higher than double-welded Slip-On flanges.[2] For ensuring adequate bore dimension, the thickness of connecting pipe must be specified. A space should be created between flange and pipe, before welding in a Socket Weld Flange.

The disadvantage of socket weld flange is that a perfect gap that should be made. In corrosive environments, and mainly in stainless steel piping systems, the crack between pipe and flange will lead to corrosion problems. This flange is not permitted for some processes.

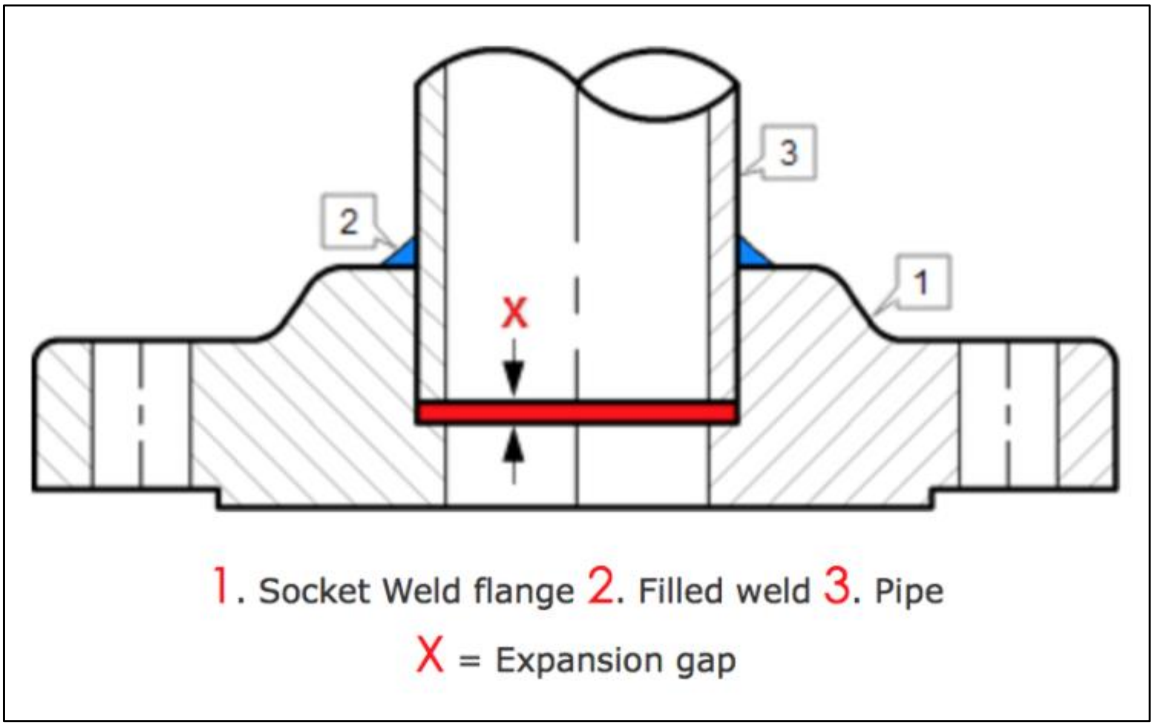


Figure 1.3 Section view of Socket weld Flange (Image Reference: theprocesspiping.com)

3) Screwed Flanges:

The Screwed or Threaded flanges are utilized in the pipes where welding cannot be done. A Screwed Flange cannot be used with the adjacent pipe of very less thickness as the threads on the pipe is impossible to cut.

Figure 1.4 Screwed Flanges (Image Reference: wermac.org)

4) Lap Joint Flanges:

The Lap joint flanges are used with stub ends when piping is of a costly material. For example, as the CS flange will not be able to be in contact with the product, it can be used in stainless steel piping system. Butt-weld will be there between stub ends to the piping and the flanges are kept loose over the same. There is chamfering on inside radius of these flanges to remove stub end radius.

These flanges are similar to a Slip-On flange and as an exception for the radius as well as the bore to take in the flanged part of the Stub End. Their pressure retaining capability is higher than that of Slip-On flanges and the fatigue life is only one tenth compared to Weld Neck flanges.[6] Thus, this flange connections are applied in low-pressure and non-critical applications.

Figure 1.5 Lap Joint Flange (Image Reference: wermac.org)

5) Weld Neck Flanges:

Welding Neck Flanges can be recognized easily by its long-tapered hub, that converges gradually to the wall thickness of a pipe or fitting. The long-tapered hub gives a vital reinforcement for several applications which involve high pressure, sub-zero or elevated temperatures.[12] The smooth transition of the hub from the flange thickness to pipe wall thickness provided by the taper is in a way highly beneficial, for the conditions of repeated bending, imposed by line expansion or any other forces that varies.

The bore of these flanges is made exact to the inside diameter of mating pipe so that there won’t be any obstruction to the flow of product which significantly reduces the erosion as well as the turbulence in the flow. These flanges offer an excellent distribution of stress through the hub. Butt-welding attachment is given between pipe and Weld Neck flanges. Such type of flanges is readily used for severe conditions where the weld joints are inspected radiographically. The thicknesses of welding ends are also specified in addition to the flange specifications.

The gradual transition of thickness from the base of the hub to the pipe thickness at the butt weld provides important reinforcement of the Weld Neck flange.

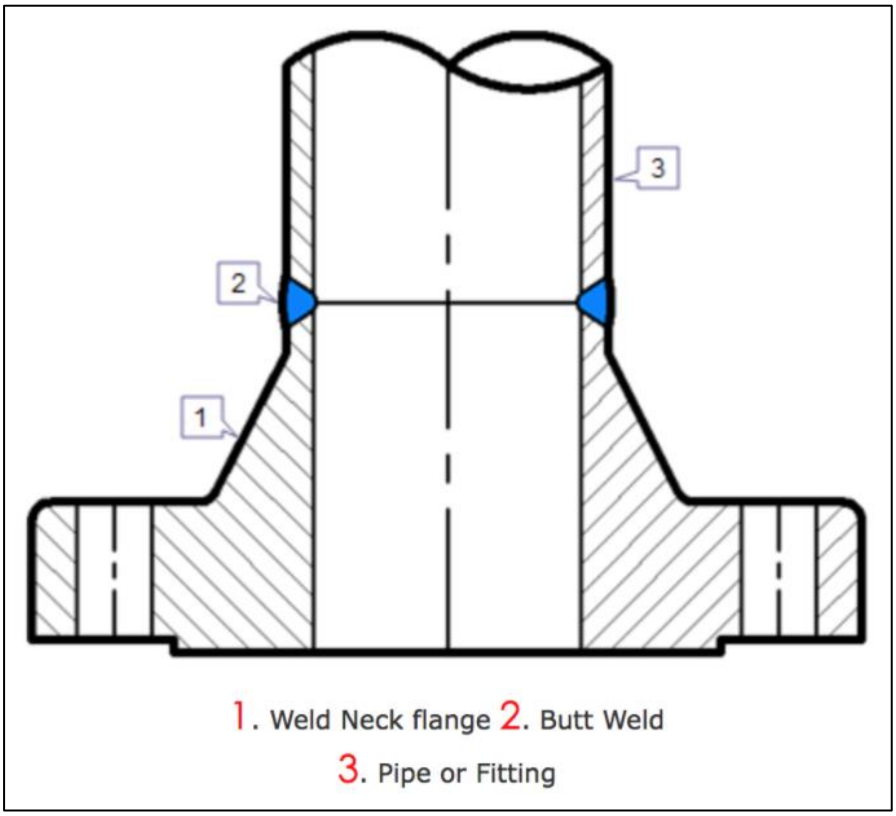


Figure 1.6 Section View of Weld Neck Flange (Image Reference: theprocesspiping.com)

6) Blind Flanges:

Blind Flanges are manufactured without a bore and are utilized for blanking off the ends of piping, Valves and pressure vessel openings. From a point of view of internal pressure as well as bolt loading, blind flanges, are the types which are very highly stressed.[1] Since the stresses induced are mostly bending stresses at the center and as there is no standard inside diameter, these flanges are suitable for higher pressure temperature applications.

Figure 1.7 Blind Flange (Image Reference: wermac.org)

1.5 Classification of Flanges based on Type of Facing:

Flanges can be classified on the basis of type of facings as follows:

1) Raised Face Flange (RF):

The Raised Face flange is most common type and is utilized in process plant applications, and it gets identified easily. It is called as a raised face as the gasket surfaces are raised above the bolting circle face. This face type allows the use of a good combination of gasket designs, including flat ring sheet types as well as metallic composites like spiral wound and double jacketed types. The reason for use of a RF flange is to increase the pressure on smaller gasket areas and hence, increasing the pressure containing capacity of the joint.

Figure 1.8 Raised Face Flange (Image Reference. engineeringlearn.com)

2) Flat Face Flange (FF):

The Flat Face flange has a gasket surface in the plane same as the bolting circle face. When the mating flange is made from casting, that’s when a Flat Face flange are used. Flat face flanges can never be bolted to the Raised Face Flange.

Figure 1.9 Flat Face Flange (Image Reference. engineeringlearn.com)

3) Tongue and Groove (T/G) Flanges:

One of the flange faces has a raised ring (Tongue) which is machined on the flange face while the mating flange has a matching depression (Groove) machined on its face.[5] The matching of Tongue and Groove is necessary.

Tongue-and-groove facings are available in both large and small types. They vary from male and female in that the inside diameter of the tongue and groove do not extend into the flange base, hence, holding on the gasket on its inner as well as outer diameter.

Figure 1.10 Tongue and Groove Flanges (Image Reference: wermac.org)

4) Male and Female (M/F) Flanges:

There must be a match between these types of flanges too. On one flange face, an area that extends after the normal flange face (Male). The mating flange has a matching depression (Female) which is machined on its face. The female face is 3/16-inch deep, and the male face is 1/4-inch high, and both have smooth finish.

The outer diameter of the female face functions for locating and retaining the gasket. Many male and female flanges are found on the Heat Exchanger shell to channel and cover flanges.[19] There is a smooth finish on male and female faces.

Bolting of general faces of flange such as T&G, F&M and RTJ must not be done. This is because the flange surface type does not match as well as the other reason will be the gasket. None of the gaskets have one type one side and another type on the other side.

1.6 Classification of Flanges based on Pressure – Temperature rating:

The flanges are also classified by the pressure temperature rating in ASME B 16.5 as below:

1. 150#
2. 300#
3. 400#
4. 600#
5. 900#
6. 1500#
7. 2500#

Pressure temperature rating charts, in the standard ASME B 16.5, specify the nonstock working gauge pressure to which the flange can be subjected to at a particular temperature. Flanges can withstand different pressures at different temperatures. As there is a temperature increase, pressure rating of the flange decreases.[7]

The pressure classes of 150#, 300#, etc. are the basic ratings and the flanges can withstand higher pressures at lower temperatures. Allowable pressure for various materials against the temperature are indicated in ASME 16.5.

1.7 Classification of Flanges based on Face finish:

There are three types of finishes done on to the facings:

1) **Stock Finish Flange:**

This flange surface finish is mostly used, because practically, is suitable for all ordinary service conditions. A good level of friction is created between mating surfaces as the soft face of gasket will set into the finish under compression.

2) **Smooth Finish Flange:**

Tool markings are not apparent in this finish. These finishes are generally used for gaskets with metal facings like corrugated metal, double jacketed and flat steel. A seal is created when two smooth surfaces mate and flatness of opposite face will affect the seal.

3) **Serrated Finish Flange:**

It is different from stock finish even though there is continuous or phonographic spiral groove, because the groove is generated using a right-angled tool which creates a "V" geometry with 45° angled serration.[17] When the fluid in the equipment has low density or if it can find paths for leakages easily from cavity, concentric serrations are suggested for the face finish.

1.8 Problem Definition:

Even though the design of flanges is as per the standard procedures followed by internationally accepted codes, during the operating conditions, due to various factors, there is a problem of leakage of the bolted flange joint. This is because in the design of Non-standard Flanges and Body Flanges, sometimes the gasket seating forces are not appropriate or may be the bolt forces are very low. So, such issues may take place.

The weight of the flange is another issue as the flanges that are used in operation for the heat exchanger, often have more weight than the optimized one. Just by increasing the parameters of the flange design will not satisfy other factors for design such as weight of flange. So, by designing the optimized flange can decrease the material requirement of the flange, and ultimately it has the direct impact on the cost of the flange.

1.9 Objectives:

1. To design a Body Flange for the equipment with minimum leakage characteristics.
2. To design the Flange according to ASME Section VIII Division 1 Mandatory Appendix 2.
3. To design the Flange according to ASME PCC-1.
4. FEA of the given flange in order to validate the theoretical results.
5. Parametric study of the flange to obtain the optimized design of Flange.
6. To design a flange for optimized weight.

1.10 Thesis Organization:

Chapter 1 discusses about flanges, types of flanges, classification of flanges based on method of attachment, types of facing, based on temperature- pressure ratings and types of face finish.

Chapter 2 is the detailed summary of the research papers which are published by researchers who had worked in the field of study on Flange design.

Chapter 3 is the design of a body flange used in a heat exchanger and the design is entirely as per ASME Section VIII Division 1 Mandatory Appendix 2 "Rules for Bolted Flange Connections with Ring Type Gaskets "and ASME PCC-1 Appendix O "Assembly Bolt Stress Determination".

Chapter 4 is the detail Finite Element Analysis of the given flange under the given loading and boundary conditions. In this chapter, the results obtained from ASME Section VIII Division 1 Mandatory Appendix 2 calculations are validated. The Parametric Study of the given flange in order to get the optimized design of the flange is discussed in the chapter. Under the influence of input parameters, the effects on the output parameters are studied and the best possible solution is taken as the best design for the flange.

Chapter 5 is the comparison of weight of the given flange and that of the optimized design of the flange. From that, an economical solution is obtained that helps in material reduction for the flange and cost incurred for the flange is reduced.

CHAPTER 2
Literature Review – Research Papers

2.1 Summary of Research papers

Literature Title: Performance Analysis of Flange Seal Assembly for Heat Exchanger

Conference/Journal- IEEE 22nd International Conference on Computer Supported Cooperative Work in Design (2018)

Summary- In this paper, the static strength of the fastening bolts and the sealing gasket in the flange connection seal assembly were analysed. Results show that the maximum contact pressure and Von Mises stresses of the sealing gasket are located near the bolt holes where the leakage happens mostly.

Literature Title: Comparative study of the behavior of conventional gasketed and compact non-gasketed flanged pipe joints under bolt up and operating conditions

Conference/Journal- International Journal of Pressure Vessels and Piping (2003)

Summary- In this study, two-dimensional non-linear finite element studies have been performed for both gasketed and non-gasketed bolted flange pipe joints. Based on the stress results for the flange and the bolt and the flange rotation/displacement, compact non-gasketed flange joints are shown to be a viable and preferable alternative to the conventional gasketed flange joints.

Literature Title: An Accurate Method of Evaluating Relaxation in Bolted Flange Connections

Conference/Journal- ASME Pressure Vessels and Piping Conference (1997)

Summary- This paper presents a simplified analytical method for relaxation of bolted flange joints for heat exchanger applications along with the test results on rigidity of the joint. This enables strategic design for the bolted joint connections.

Literature Title: The Design of Flanges Based on Flexibility and Tightness

Conference/Journal- ASME Pressure Vessels and Piping Conference (2003)

Summary- In this paper, a general comprehensive method based on gasket-bolt-flange elastic interaction is presented for the analysis of the joint. The results show the effect of gasket creep and thermal expansion on the leakage behaviour of the flange connection.

Literature Title: Optimization and Standardization of Flanged and Flued Expansion Joint Design

Conference/Journal- Journal of Pressure Vessel Technology (2019)

Summary- This paper is directed towards the standardization and optimum design approach of flange and flued expansion bellow fulfilling ASME VIII-1 and TEMA standard requirement. But there is a limitation of not completely fulfilling the thermal expansion factor as the accuracy of the results are 78%.

Literature Title: Leakage Characteristics of Flanged Pipe Joints

Conference/Journal- Journal of Strain Analysis (1996)

Summary- In this paper, detail research about the gasket properties is studied keeping in mind the aspect of increasing leakage problem in the flange pipe joints. The results obtained from the study suggests that increasing the gasket thickness reduces the leakage pressure for given conditions.

Literature Title: A parametric study of metal-to-metal full face taper hub flanges

Conference/Journal- International Journal of Pressure vessel and Piping (2000)

Summary- This paper conducts a parametric study of full-face taper hub flanges and it suggests that pre-stress in bolts should be made equal to their operating design stress. This gives a suitable design which can have balanced conditions for getting optimum bolt size and correct number of bolts for the design of bolted joint connections.

Literature Title: Determination of Temperature Limits for Heat Exchanger Joint Assembled of Solid Stainless Tube Sheet with Girth Flanges

Conference/Journal- ASME Pressure Vessels and Piping Conference (2017)

Summary- The paper depicts the thermal time dependent transient analysis of the above model is conducted to compute the temperature distribution in the flanged joint assembly for different time steps. The study determines both the permissible heating rates during start-up and the temperature limits.

Literature Title: On the Characteristics of Rectangular Bolted Flanged Connections with Gaskets Subjected to External Tensile Loads and Bending Moments

Conference/Journal- Journal of Pressure Vessel Technology (1994)

Summary- The purpose of this paper is to contribute to an optimal design of practical bolted connections with gaskets subjected to external tensile loads and bending moments. Experiments are performed concerning load factors and maximum stresses produced in bolts. Analytical results are compared with experimental.

Literature Title: Creep analysis of bolted flange joints

Conference/Journal- International Journal of Pressure vessel and Piping (2007)

Summary- In this paper, the development of a simple analytical solution to the creep-relaxation problem encountered in bolted flange connections of the float type is carried out. Particular emphasis is put towards relaxation caused by the flange and bolt material creep.

Literature Title: Strength of Integral Pipe Flanges

Conference/Journal- Bulletin of JSME (1979)

Summary- In this paper, the stress distribution in a standard integral flange is analysed by Finite Element Method and the effects of design factors on the stress concentration are clarified. The results of the calculations shows that the conventional standards are not always in a good agreement with results of FEA.

Literature Title: Thermal Stress Analysis and The Sealing Performance Evaluation of Bolted Flange Connection at Elevated Temperature

Conference/Journal- ASME Pressure Vessels and Piping Conference (2009)

Summary- The effects of gasket properties and nominal diameter of flanges on the above characteristics are examined numerically. In the experiments, the amount of helium gas leakage in the connection was measured. Using the obtained gasket stress at the elevated temperature, a method for estimating the amount of leakage is proposed.

Literature Title: On the deformation of bolt head and nut in a bolted flange connection for heat exchanger

Conference/Journal- Bulletin of JSME (2014)

Summary- In this paper, a study of increment in the axial force produced in the bolt when the load is applied on the bolted flange joint for a heat exchanger. Here, the deformations of bolts and nuts are compared with the experimental methods and found the value of factor Kt as precise as Kc for the given design considerations.

Literature Title: The Sealing Performance of a Large Diameter Bolted Joint Under Elevated Temperature

Conference/Journal- ASME Pressure Vessels and Piping Conference (2007)

Summary- This paper proposes and analyzes sealing performance of the pipe flange connection was evaluated by measuring the amount of gas leakage at 50 ℃ and 100 ℃ under internal pressure. The estimated results were compared with the experimental results. The results were also compared with the small nominal diameter test result.

Literature Title: Effects of Partial Cooling on Tightness of Heat Exchanger Girth Flange

Conference/Journal- ASME Pressure Vessels and Piping Conference (2017)

Summary- In this paper, the effects of partial cooling on flange tightness were studied. The flange tightness was evaluated by wideness of partially cooled region (liquid level) of heat exchanger and gasket recovery characteristics parametrically. Based on these studies, it was concluded that the gasket contact pressure was decreased by the partial cooling of heat exchanger.

Literature Title: Bolt Preload Control for Bolted Flange Joint

Conference/Journal- ASME Pressure Vessels and Piping Conference (2002)

Summary- In this paper, Step like increment of the bolt tension under the repeated tightening with small increment of the tightening torque is applied to observe the effects on the bolt force as well as the effects on the precision of the flange alignment is discussed.

Literature Title: FEM Stress Analysis of Bolted Flange Joints in Elevated Temperature Service Condition

Conference/Journal- ASME Pressure Vessels and Piping Conference (2018)

Summary- In this paper, the effects of internal pressure, temperature as well as the heating rate on the variations of bolt load, bolt stress and gasket contact stress have been evaluated. The results show that the maximum bolt stress increases while average gasket contact stress decreases with increasing the temperature under steady-state thermal loading. Results of this paper shows that with application of internal pressure and increasing temperature, the degree of flange rotation increases, and the contact stress increases gradually form inner to outer radius of the bolted Flange Joints.

Literature Title: Inelastic Analysis of Sealing Characteristics of Flanges with Metal Ring Joint Gasket at elevated temperature

Conference/Journal- ASME Pressure Vessels and Piping Conference (2006)

Summary- In this paper, the heat transfer analysis and elastic plastic analysis were performed to investigate the sealing characteristics of dissimilar material flanges with a metal ring joint gasket at an elevated temperature. The effects of meteorological condition changes on the sealing characteristics and the behavior of the gasket ring through the first and second operation cycles were investigated. The results of this study suggests that the operation cycles will produce a progressive plastic deformation. This deformation may affect the sealing performance.

Literature Title: Effects of Temperature change on Bolt Load and Gasket Load of Bolted Flange connection with Ring Type Joint Gasket

Conference/Journal- ASME Pressure Vessels and Piping Conference (2008)

Summary- In this paper, finite element analysis is performed for the piping flange connection of 12-inch class 900 RTJ which is subjected to the conditions of internal pressure 10 MPa as well as temperature 450°C.As a part of the result, it is required to study the difference of thermal expansion as well as the stiffness between flange and bolt along with the temperature dependency.

Literature Title: Determination of Gasket effective width based on Leakage

Conference/Journal- ASME Pressure Vessels and Piping Conference (2004)

Summary- This paper presents some work related to the investigation of the effect of the non-uniform gasket contact stress due to flange rotation on the leakage behavior of gasket materials. It was found that the effective width depends on the applied load and the rotational flexibility of the flange and the type of gasket. For sheet gaskets, the maximum gasket stress located at the gasket outer periphery and hence the thickness of the gasket at this location is found to be one of the key parameters that controls leakage.

CHAPTER 3

Design of the Flange

- **Design of Flange according to ASME Section VIII Div. 1 Mandatory Appendix 2:**

The rules mentioned in this Appendix applies to the design of bolted flange connections with the gaskets that are within the circle enclosed by the bolt holes and with no contact outside this circle, and are to be used in conjunction with the applicable requirements of this appendix. The design of a flange involves the selection of the gasket (material, type, and dimensions), flange facing, bolting, hub proportions, flange width, and flange thickness. Flange dimensions shall be such that the stresses in the flanges do not exceed the allowable flange stress.

➢ Input Parameters for the design of Flange are as follows:

Internal design pressure, P = 0.448159 Mpa (65 psi)

Design temperature = 671.11 ℃

Minimum Design Metal Temperature for equipment = -28.9 ℃

Ambient Temperature = 30 ℃

Material of Body Flange = SA 965M GR. 304H

Corrosion allowance, CA = 1.6 mm

Material of bolting = SA 193 GR B8M

For Gasket, given parameters are:

Gasket factor = m = 3

Gasket or joint contact surface unit seating load = y = 68.95 Mpa

Gasket width = N = 30 mm.

The assumed dimensions of Flange as per standard industrial practice are as follows:

Flange I.D = B= 4953 mm

Flange O.D = A= 5405 mm

Flange BCD = C= 5265 mm

$g0$ = 49 mm

$g1$ = 65 mm

Gasket I.D = 4993 mm

Gasket O.D = 5055 mm

Corroded Flange I.D = B_c = 4953+(2*1.6) = 4956.2 mm.

Corroded g0 = 49-1.6 = 47.4 mm

Corroded g1 = 65-1.6 = 63.4 mm

Now, according to ASME Sec. II Part D Table 1A,

Allowable design stress for the material of Flange at design temperature = S_f = 34.2692 Mpa

Allowable design stress for the material of Bolting at Ambient temperature = S_{fa} = 138 Mpa

Allowable design stress for the material of Bolting at design temperature = S_b = 40.57 Mpa

Allowable design stress for the material of Bolting at Ambient temperature = S_a = 130 Mpa

3.1 Gasket Design:

From Table 2-5.2 of Appendix 2,

Basic Gasket seating Width,

$$bo = \frac{N}{2} \tag{3. 1}$$

$$= \frac{30}{2}$$

$$= 15 \text{ mm.}$$

Effective Gasket Width,

$$b = 2.5 * \sqrt{bo} \tag{3. 2}$$

$$= 2.5 * \sqrt{15}$$

$$= 9.76 \text{ mm.}$$

Diameter at location of Gasket load reaction [12],

$$G = G.O.D - 2b \tag{3. 3}$$

$$= 5055 - (2 * 9.76)$$

$$= 5035.48 \text{ mm.}$$

Figure 3.1 Kammprofile Gaskets (Reference website: gobizkorea.com)

3.2 Bolting Loadings and sizing of bolts with the designed Gasket:[19]

Total hydrostatic end force,

$$H = 0.785 G^2 P \tag{3. 4}$$

$$= 0.785 * 0.448159 * 5035.315^2$$

= 8924471 N

Total Joint contact Compression Load,

$$Hp = 2b * \pi G m P \tag{3. 5}$$

$$= 2 * 9.76 \ * \pi * 5035.315 * 3 * 0.448159$$

= 415130.5 N

Minimum required Bolt loads for operating conditions,

$$Wm1 = Hp + H \tag{3. 6}$$

= 8924471+ 415130.5

= 9339601 N

Figure 3.2 Bolt Loads acting on the Flange joint (Reference website: hextechnologies.com)

Minimum required Bolt loads for Gasket seating,

$$Wm2 = \pi bGy \tag{3. 7}$$

$$= \pi * 9.76 * 5035.315 * 68.95$$

= 10644371 N

Total Cross-Sectional Area of Bolts at Root of Thread or Section of least diameter under stress, required for the Operating conditions,

$$Am1 = \frac{Wm1}{Sb} \tag{3. 8}$$

$$= \frac{9339601}{40.57}$$

= 230209.53 mm^2

Total Cross-Sectional Area of Bolts at Root of Thread or Section of least diameter under stress, required for the Gasket Seating [5],

$$Am2 = \frac{Wm2}{Sa} \tag{3. 9}$$

$$= \frac{10644371}{130}$$

= 81879.77 mm^2.

Total required cross-sectional area of bolts,

$$Am = Greater\ of\ Am1\ and\ Am2$$

Here, $Am1 > Am2$

Hence, $Am = 230209.53$ mm^2.

According to TEMA Section 9- Table D-5M, for the assumed diameter of bolts,

Root Area of single bolt, $Ab = 3392.8964$ mm^2.

Flange design Bolt load, for Operating Conditions,

$$W = \frac{Sa}{2} * (Am1 + Ab) \tag{3. 10}$$

$$= \left(\frac{130}{2}\right) * (230209.53\ + 352861.225)$$

= 37899600 N

3.3 Total Flange Loads and Flange Moments:

Flange Loads:

Hydrostatic End Force on Area Inside of Flange,

$$Hd = 0.785B^2P \tag{3.11}$$

$$= 0.785 * (4956.2)^2 * 0.448159$$

= 8645663 N

Gasket Load [2],

$$Hg = Wm1 - H \tag{3. 12}$$

$$= 9339601 - 8924471$$

= 415130.16 N

Difference between Total Hydrostatic End Force and the Hydrostatic End Force on Area Inside of Flange [11],

$$Ht = H - Hd \tag{3. 13}$$

$$= 8924471 - 8645663$$

= 278808.406 N

Lever Arms,

Radial distance from the bolt circle, to the circle on which Hd acts [11],

$$hd = R + 0.5g1 \tag{3. 14}$$

where, $R = \left(\frac{C-B}{2}\right) - g1$ (3. 15)

$$= \left(\frac{5265-4953}{2}\right) - 65$$

= 91 mm.

$$hd = 91 + (0.5 * 63.4)$$

= 122.7 mm

Radial distance from gasket load reaction to the bolt circle,

$$hg = \frac{C - G}{2} \tag{3. 16}$$

$$= \frac{5265-5035.315}{2}$$

= 114.75 mm

Radial distance from the bolt circle to the circle on which Ht acts [18],

$$ht = \frac{R + g1 + hg}{2} \tag{3. 17}$$

$$= \frac{91+63.4+114.84}{2}$$

= 134.57 mm

Flange Moments [7]:

Component of moment due to Hd,

$$Md = Hd * hd \tag{3. 18}$$

$$= 8645663 * 122.7$$

= 1060822850.1 N.mm

Component of moment due to Ht,

$$Mt = Ht * ht \tag{3. 19}$$

$$= 278808.406 * 134.57$$

= 37519247.19 N.mm

Component of moment due to Hg,

$$Mg = Hg * hg \tag{3. 10}$$

$$= 415130.16 \quad * 114.75$$

= 47636185.85 N.mm

Total moment acting upon the flange under Operating conditions,

$$Mo = Md + Mt + Mg \tag{3. 11}$$

$$= 1060822850.1 \ + 37519247.19 + 47636185.85$$

= 1146448 KN.mm

Total moment acting upon the flange under Gasket Load,

$$Mog = W * hg \tag{3. 12}$$

$$= 37899600 * 114.76$$

= 4340458 KN.mm

Taking the higher value from the above-mentioned total moments,

$$Mo > Mog$$

Hence, the total moment for the flange considered is Mo =1146448 KN.mm

3.4 Shape Constants used for calculations [13]:

1) Shape Constant K:

$$K = \frac{A}{B} \tag{3. 13}$$

$$= \frac{5405}{4953}$$

$$= 1.1$$

2) Shape Constant T:

$$T = \frac{K^2(1 + 8.55246\, log_{10} K) - 1}{(1.04720 + 1.9448K^2)(K - 1)} \tag{3. 14}$$

$$= \frac{1.1^2(1+8.55246 \log_{10} 1.1)-1}{(1.04720+1.9448(1.1)^2)(1.1-1)}$$

$$= 1.8804$$

3) Shape Constant U:

$$U = \frac{K^2(1 + 8.55246\, log_{10} K) - 1}{(1.36136(K^2 - 1))(K - 1)} \tag{3. 15}$$

$$= \frac{1.1^2(1+8.55246\, log_{10} 1.1)-1}{(1.36136(1.1^2-1))(1.1-1)}$$

$$= 24.341$$

4) Shape Constant Y:

$$Y = \frac{1}{k-1}\left[0.66845 + 5.71690 \frac{k^2\, log_{10} k}{k^2 - 1}\right] \tag{3.26}$$

$$= \frac{1}{(1.1-1)}\left[0.66845 + 5.71690 \frac{1.1^2\, log_{10} 1.1}{1.1^2-1}\right]$$

$$= 22.151$$

5) Shape Constant Z:

$$Z = \frac{k^2 + 1}{k^2 - 1} \tag{3.27}$$

$$= \frac{1.1^2+1}{1.1^2-1}$$

$$= 11.48$$

6) g_1/g_0:

$$\frac{g1}{g0} = \frac{63.4}{47.4} = 1.338 \tag{3.28}$$

7) h_0 :

$$\text{h0} = \sqrt{}Bg0 = \sqrt{(4956.2 * 47.4)} = 484.69 \tag{3.29}$$

8) h / h_0 :

$$\frac{h}{h0} = \frac{120}{484.69} = 0.247 \tag{3.30}$$

9) From Fig. 2-7.2, ASME Section VIII Div.1 Mandatory App.2, F = 0.891

From Fig. 2-7.3, ASME Section VIII Div.1 Mandatory App.2, V = 0.425

From Fig. 2-7.6, ASME Section VIII Div.1 Mandatory App.2, f = 1.042

10) Factors e and d:

$$e = \frac{F}{h0} = \frac{0.891}{484.69} = 0.0018 \tag{3.31}$$

$$d = \frac{U}{V} * h0 * g0^2 = \frac{24.341}{0.425} * 484.69 * 47.4^2 \tag{3.32}$$

$$d = 66651882.66$$

3.5 Stresses acting on the flanges:

Assuming the flange thickness to be t=392 mm, calculating factor L,

$$L = \frac{te + 1}{T} + \frac{t^3}{d} \tag{3.33}$$

$$= \frac{(390.4*0.002)+1}{1.8804} + \frac{390.4^3}{66651882.66}$$

$$= 1.806$$

Longitudinal Hub Stress:

$$SH = \frac{fM0}{Lg1^2B} \tag{3.34}$$

$$= \frac{1.042*1146411805.6}{1.806*65^2*4953}$$

$$= 32.22 \text{ MPa}$$

Radial Flange Stress:

$$SR = \frac{(1.333te + 1)M0}{Lt^2B} \tag{3.35}$$

$$= \frac{(1.333 * 390.4 * 0.0018 + 1) * 1146411805.6}{1.806 * 390.4^2 * 4956.2}$$

$$= 1.627 \text{ Mpa}$$

Tangential Flange Stress:

$$ST = \left(\frac{YM0}{t^2B}\right) - Z * SR \tag{3.36}$$

$$= \frac{(22.151*1146411805.6)}{390.4^2*4956.2} - (11.48 * 1.627)$$

$$= 15.41 \text{ Mpa}$$

3.6 Check for Allowable Stresses [17]:

Conditions required to be satisfied:

1) $SH \leq 1.5Sf$ (3.37)

$$1.5 * (34.26928) = 51.4 > 32.22$$

Hence, the condition is satisfied.

2) $SR \leq Sf$ (3.38)

$$1.627 < 34.26928$$

Hence, the condition is satisfied.

2) $ST \leq Sf$ (3.39)

$$15.41 < 34.26928$$

Hence, the condition is satisfied.

4) Check for Combined Stress:

$$\text{Greater of } \left(\frac{SH+SR}{2}\right), \left(\frac{SH+ST}{2}\right) < Sf \quad (3.40)$$

$$\text{Greater of } \left(\frac{32.22+1.627}{2}\right), \left(\frac{32.22+15.41}{2}\right) < 34.26928$$

$$23.815 < 34.26928$$

Hence, the thickness of the Flange is sufficient. But the flange design can still be optimized and the design can be improved.

- **Design of Flange according to ASME PCC-1 Appendix O:**

For the design of Flange according to ASME PCC-1 Appendix O, we shall apply Joint Component Approach.

Input Parameters obtained by the Gasket Vendor as mentioned in the code:

1) Maximum Permissible single flange rotation for gasket at the maximum operating temperature, $\theta g\ max = 0.5$ degree
2) Target Assembly Gasket Stress, Sgt = 180 Mpa
3) Maximum Assembly Gasket Stress, $Sg\ max$ = 295 Mpa
4) Minimum Gasket Seating Stress, Sg min $\_s$ = 140 Mpa
5) Minimum Gasket Operating Stress, Sg min $\_o$ = 97 Mpa
6) Gasket Relaxation Fraction, ϕg = 0.7

Now, from the calculated value from ASME Section VIII Div.1 Mandatory App.2,

Bolt Area, $Ab = 3392.8964$ mm^2.

Number of bolts required, nb =104.

Following are the steps to check the maximum gasket stress and flange rotation limit according to the code:

3.7 Step 1: To determine the Target bolt Stress in accordance to the following equations

Now, Gasket Partition Length = $GPI = 1.5 * GID$ = 1.5*4993 = 7489.5 mm

Gasket Partition Width = GPW = 10 mm.

Gasket Area,

$$Ag = \frac{\pi(GOD^2 - GID^2)}{4} + \frac{GPI * GPW}{2} \tag{3.41}$$

$$= \frac{\pi(5053^2 - 4993^2)}{4} + \frac{7489.5 * 10}{2}$$

= 510854 mm²

Target Bolt Stress,

$$Sb\ sel = \frac{Sgt * Ag}{nb * Ab} \tag{3.42}$$

$$= \frac{180 * 510854}{104 * 3392.8964}$$

= 375 Mpa

3.8 Step 2: To determine if the bolt lower limits are controlled

Maximum Permissible Bolt Stress, $Sb\ max = 0.7 * Sy$ = 507 Mpa

$$Sb\ sel = min.(Sb\ sel, Sb\ max) \tag{3.43}$$

= 375 Mpa

3.9 Step 3: To determine if the bolt higher limits are controlled

Minimum Permissible Bolt Stress, $Sb\ min = 0.35 * Sy$ = 276 Mpa

$$Sb\ sel = max.(Sb\ sel, Sb\ max) \tag{3.44}$$

= 276 Mpa

3.10 Step 4: To determine if the flange limits are controlled

Maximum Permissible bolt stress prior to flange damage, $Sf\ max = 430$ Mpa

Flange Rotation at this load, $\theta f\ max$ = 0.32 degree

$$Sb\ sel = min.(Sb\ sel, Sf\ max) \tag{3.45}$$

= 375 Mpa

3.11 Step 5: To check if the gasket assembly seating stress is achieved

$$Sb\ sel \geq\ Sg\ min\ \_s\ * \frac{Ag}{Ab * nb} \tag{3.46}$$

$$= 140\ * \frac{510854}{3392.8964\ * 104}$$

$$= \ 130 \text{ Mpa}$$

Here, L.H.S > R.H.S, hence the condition is satisfied.

3.12 Step 6: To check if the gasket operating stress is maintained

$$Sb\ sel \geq \frac{\left[Sg\ min\ \_o * Ag + \left(\frac{\pi}{4}\right) Pmax * GID^2\right]}{Ab * nb * \phi g} \tag{3.47}$$

$$= \frac{[140 * 510854 + 0.8618 * 0.785 * 4993^2]}{3392.8964\ * 104 * 0.7}$$

$$= 173 \text{ Mpa}$$

Here, L.H.S > R.H.S, hence the condition is satisfied.

3.13 Step 7: To check if the gasket maximum stress is exceeded

$$Sb\ sel \leq\ Sg\ max\ * \frac{Ag}{Ab * nb} \tag{3.48}$$

$$= 295 * \frac{510854}{3392.8964\ \ * 104}$$

$$= \ 274 \text{ Mpa}$$

Here, L.H.S < R.H.S, hence the condition is satisfied.

3.14 Step 8: To check if the Flange rotation Limit is exceeded

$$Sb\ sel \leq\ Sf\ max\ * \frac{\theta g\ max}{\theta f\ max} \tag{3.49}$$

$$= 430 * \left(\frac{0.5}{0.32}\right)$$

$$= 672 \text{ Mpa}$$

Here, L.H.S > R.H.S, hence the condition is satisfied.

Now,

3.15 Torque required by the bolts for proper installment of the Flange,

$$Tb = \frac{Sb\ sel * K * Ab *\ \phi b}{1000} \tag{3.50}$$

$$= \frac{375*3392.8964\ *0.2*56}{1000}$$

$$= 22206 \text{ N.m}$$

CHAPTER 4

FEA of the Flange and Parametric Study of Flange

4.1 Introduction

In order to validate the results obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2, we need to conduct the Finite element analysis of the flange joint. Analysis is performed to study the behaviour of the given Flange joint under applied mechanical loadings. ANSYS software is used for this analysis.

4.2 Analysis Details:

The below details represent the results of finite element analysis of Flange Joint. Internal design pressure is applied to the Inner faces of the Flange. 3-Dimensional model is prepared in ANSYS Workbench.

Component	Flange Joint
Software	ANSYS 19.2
Analysis type	Static Structural, Mechanical (Elastic Stress Analysis)
Element Type	Second order solid elements SOLID 186
No. of nodes	2751624
No. of elements	678410
Design code	ASME Sec. VIII Div.1 Ed. 2021

Table 4.1 Details about Analysis of Flange

4.3 Material Properties:

The flange joint mainly constitutes of two flanges and a gasket in between to provide a leak proof joint. Below Table 4.2 represents the material of the flange as well as the core material of the gasket used in the flange.

Component	Material	Temp.	Modulus of Elasticity	Max. Allowable Stress at Design Temp.	Density
		^{o}C	MPa	MPa	Kg/m^3
Flange	SA-956 Gr. F304H	671	143460	34.34	8030
Gasket	SA-240 Gr. 304H	671	143460	34.34	8030

Table 4.2 Material properties of Flange components

- Poisson's Ratio = 0.31 for both the materials

Notes:

1. Allowable Stress of Material is obtained from ASME Sec. II Part D, Table 1A.
2. Modulus of Elasticity is obtained from Table TM1, ASME Sec. II Part D.
3. Density of Material & Poisson's ration is obtained from Table PRD of ASME Sec. II Part D.

4.4 Acceptance Criterion

The acceptance criterion is as per Part 5 of ASME Sec. VIII Div. 2 "Design based on Stress analysis".

4.4.1 Protection against Plastic Collapse [Elastic Stress Analysis] (According to Para 5.2.2 of Part 5)

a) Equivalent stress derived from the average value across the thickness of a section of the general primary membrane stresses **(Pm)** produced by internal pressure and other loads but excluding geometrical discontinuities and all secondary and peak stresses must be less than S, where S is the allowable stress of material at design temperature.

b) Equivalent stress derived from the average value across the thickness of a section of the local primary stresses **(PL)** produced by internal pressure and other loads including geometrical discontinuities but excluding all secondary and peak stresses must be less than SPL.

c) Equivalent stress derived from the average value across the thickness of a section of local primary membrane stress plus primary bending stress proportional to distance from centric produced only by mechanical load **(PL+Pb)** must be less than SPL. As per Para 5.2.2.4, SPL value is taken as larger of the quantities shown below,

(1) 1.5 times the allowable stress of material at corresponding temperature

(2) Sy for material, except above shall be used if room temperature ratio of minimum specified yield strength to ultimate strength exceeds 0.7.

4.4.2 Elastic check for Protection against Ratcheting / Secondary Stress evaluation

Equivalent stress derived from the highest value across the thickness of a section, of the linearized local primary membrane stresses plus primary bending stresses plus secondary stresses **(PL+Pb+Q)** produced by "load controlled" as well as by "strain-controlled" loads but excluding all peak stresses must be less than $\mathbf{S_{PS}}$; where S_{PS} is larger of the values below,

(1) Three times average value of tabulated S of material for highest and lowest temperature during operational cycle

(2) Two times the average tabulated yield strength for highest and lowest temperature during operational cycle, except (1) above shall be used if room temperature ratio of minimum specified yield strength to ultimate strength exceeds 0.7 or the value of S is governed by time dependent properties.

4.4.3 Protection against Local Failure (According to Para 5.3.2 of Part 5)

In addition to demonstrating protection against plastic collapse, the following elastic analysis criterion shall be satisfied for each point in the component. The sum of the local primary membrane plus bending principal stresses shall be used for checking this criterion.

4.5 Solid model:

Design modeller of ANSYS Workbench 19.2 was used for 3-D modelling the full body of geometric model. The components of the vessel are as shown in figure 4.1. As the flange is having 104 number of bolts, we have taken 3.5-degree section of the flange joint in order to reduce the computational time required for the analysis. The flange joint consists of two identical flanges and a gasket between them. Lines found in model are due to slicing operation in geometry to facilitate part meshing. The dimensions of the flange and the gasket is shown in Table 4.3.

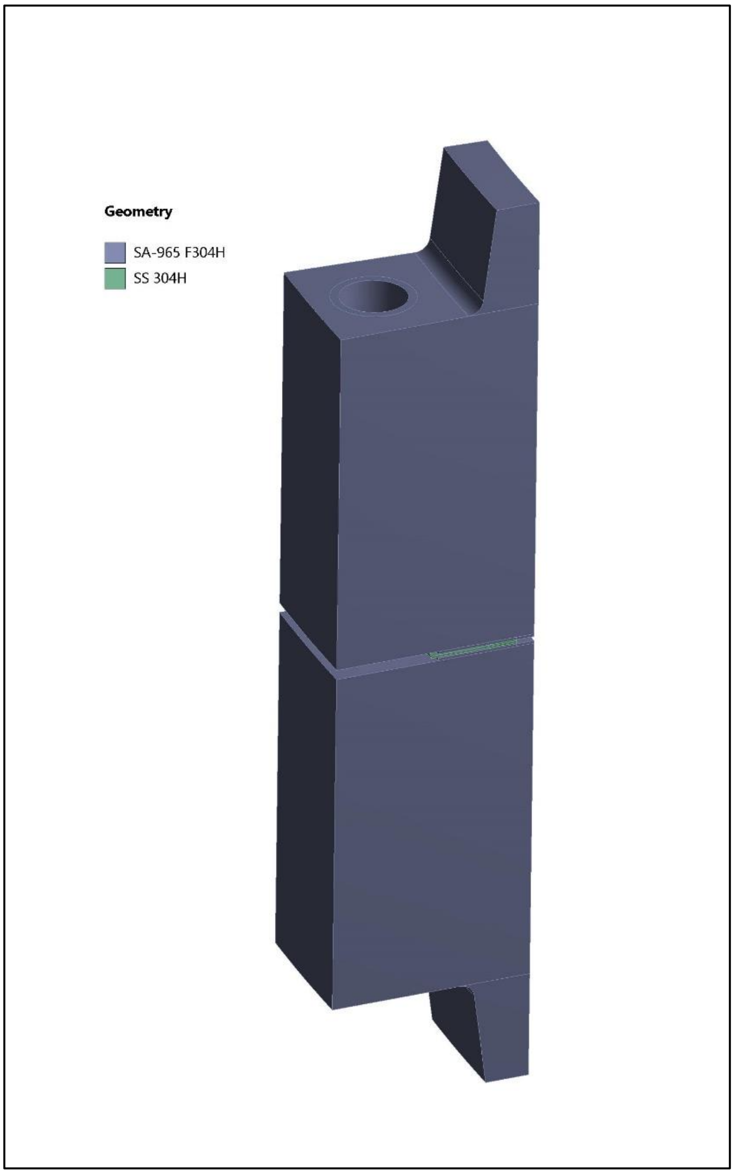


Figure 4.1 Geometry of the Flange

Sr. No.	Specifications	Dimensions (mm)
1.	Inside Diameter of Flange	4953
2.	Outside Diameter of Flange	5405
3.	Flange Thickness	392
4.	Thickness of hub at small end	49
5.	Thickness of hub at large end	65
6.	Length of Hub	120
7.	Diameter of bolt circle	5265
8.	Thickness of gasket	5
9.	Flange face Inside Diameter	4953
10.	Flange face Outside Diameter	5172
11.	Gasket Inside Diameter	4995
12.	Gasket Outside Diameter	5055
13.	Diameter of Bolt Hole	69.85

Table 4.3 Dimensions of the Flange

4.6 Solid Element Plot:

Meshing is a key component to obtain accurate results from a FEA model. The elements in the mesh should account for taking up many aspects for discretization of stress gradients more precisely. Smaller the mesh size, more accurate the solution as the design is better sampled across the physical domains. However, smaller the mesh, more time is required for computation of analysis results.

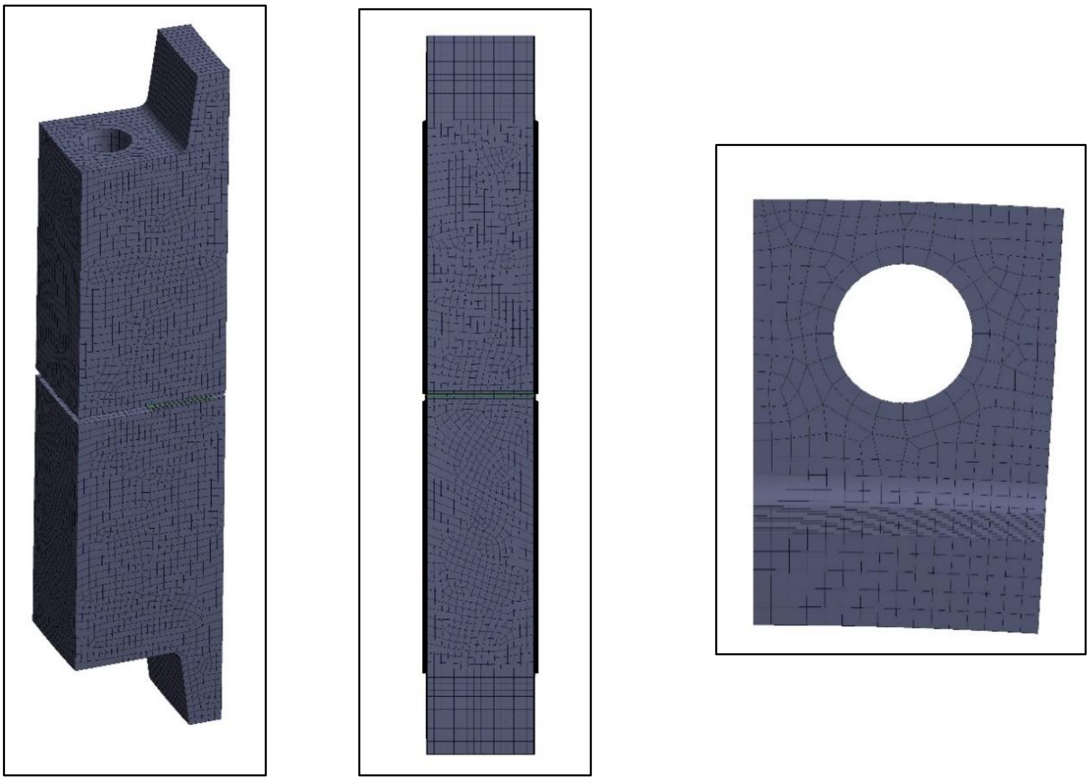


Figure 4.2 Meshed model of Flange

Analysis type	Static Structural, Mechanical
Element Type	Second order solid elements SOLID 186
Mesh Quality	
No. of nodes	2751624
No. of elements	678410
Aspect Ratio	4.014
Corner Angle	98.67 ˚
Jacobian Ratio	1.278

Table 4.4 Mesh Details

4.7 Loads and Boundary Conditions:

The cylindrical co-ordinate system is used for the analysis so as to keep the flange radially free to move and to fix the bottom hub face of the flange using nodal displacement free only in radial direction and restrict it to move in all other directions. For evaluating the stresses due to mechanical loads, the bottom nodes of the flange are fixed in all directions. Internal pressure (P) of 4.48 bar (i.e.,0.448 MPa) is applied on the inner faces of the flange and the gasket. Frictionless support is provided in order to restrict the faces of the flange to deform or move in the normal direction so that correct behavior of the flange could be obtained. Bolt loads are applied on the bolt contact faces in form of forces (Force and Force 2 in figure 4.3). The bolt loads applied in the analysis are corresponding to single bolt load as the flange is considered for a 3.5-degree section. A compensating pressure (Pressure 2 in figure 4.3) is applied on the top of the flange hub face. The compensating pressure is calculated as per the given formula:

$$Pc = \frac{Pi}{\left(\frac{Do}{Di}\right)^2 - 1}$$

$$= \frac{4.48}{\left(\frac{5051}{4953}\right)^2 - 1}$$

$$= 11.2 \text{ Mpa}$$

Where, Pc = Compensating pressure

Pi = Internal Pressure acting on the flange

Do = Outside diameter of opening of the flange

Di = Inside diameter of opening of the flange

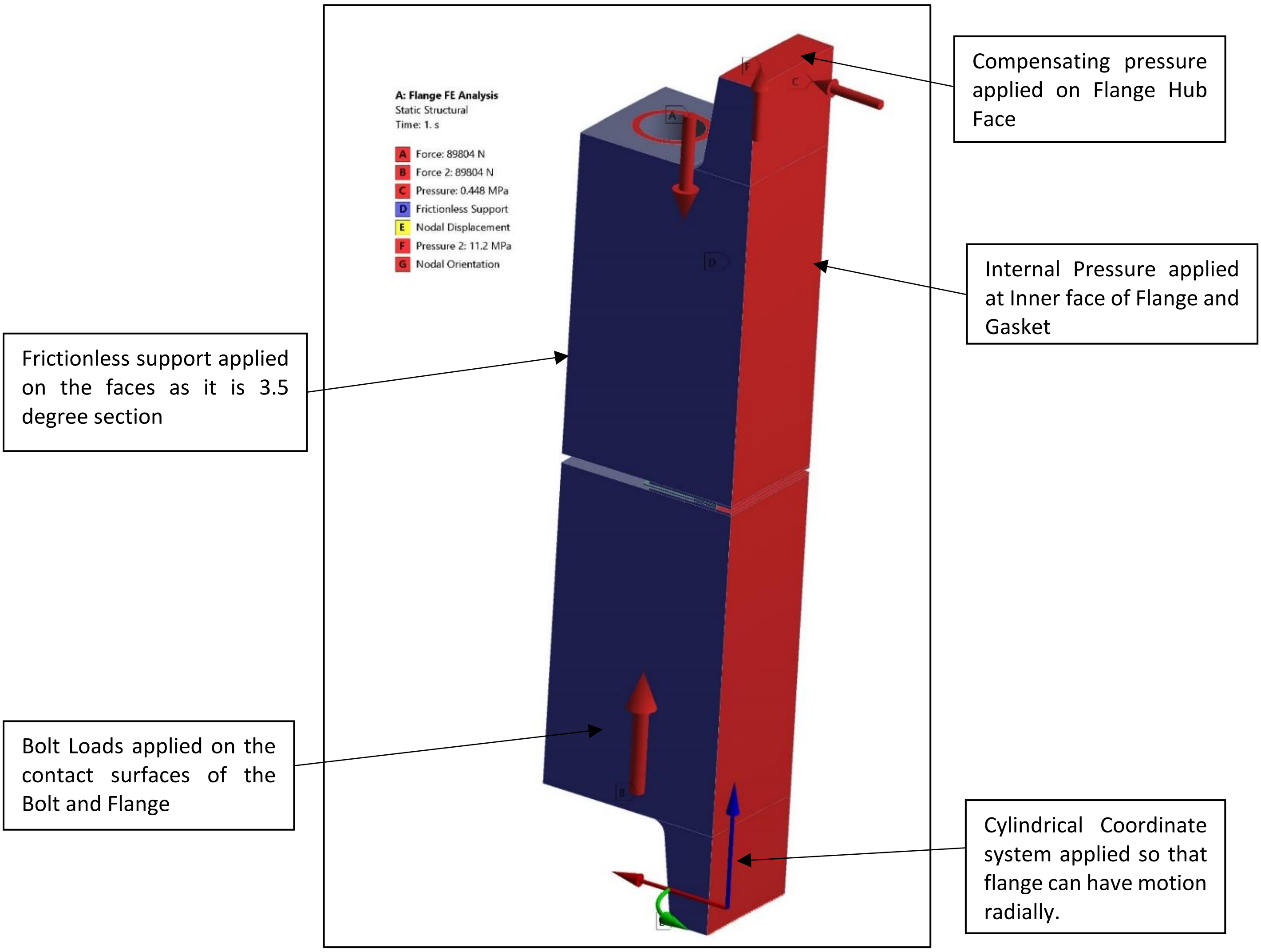


Figure 4.3 Loadings and boundary conditions acting on the Flange

4.8 Results and Discussion:

The results of the above applied boundary and loading conditions are shown in the Figure 4.4. Here, for validation of the mesh quality in the analysis, the average and the unaverage Equivalent von-mises stresses are compared simultaneously. If the difference in the values of average and unaverage Equivalent von-mises stresses are under 5% respectively, then the mesh quality is acceptable. In addition to that, to validate the results of theoretical calculations obtained from ASME Section VIII Div. 1 Mandatory Appendix 2, the stresses required while conducting the Finite element analysis are obtained through stress linearization. Stress linearization is a procedure in which the stress distribution along a line or section through the thickness in a solid is approximated with an equivalent linear stress distribution. The classes of stress i.e., primary, secondary as well as the peak stresses are evaluated and limited according to their allowable limits as guided in ASME Section VIII Division 2. Stress classification planes (SCPs) are transection across the thickness of a component through which membrane and bending stresses are developed. For planar geometry, the stress classification line (SCL) is taken by decreasing the two opposing edges of a SCP to a minute extent. These SCLs should be orthogonal to one of the two, maximum stress component or the mid-surface of the section, and are usually taken at very local structural or material discontinuities in order to examine the part in case of plastic collapse as well as the local failure.

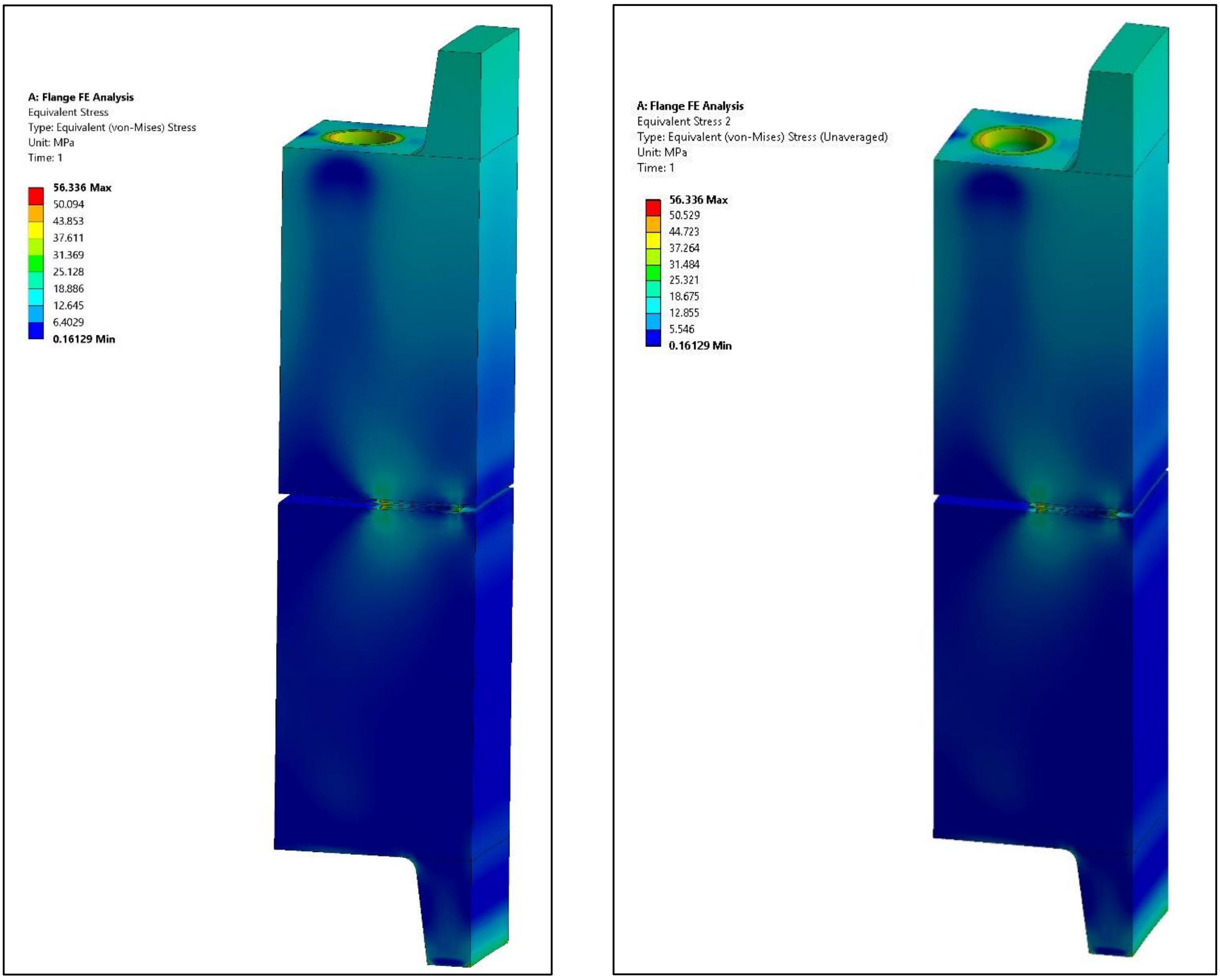
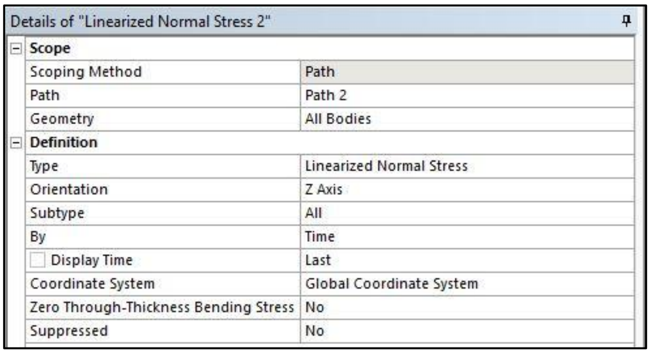


Figure 4.4 Average and Unaverage Equivalent von-mises Stresses

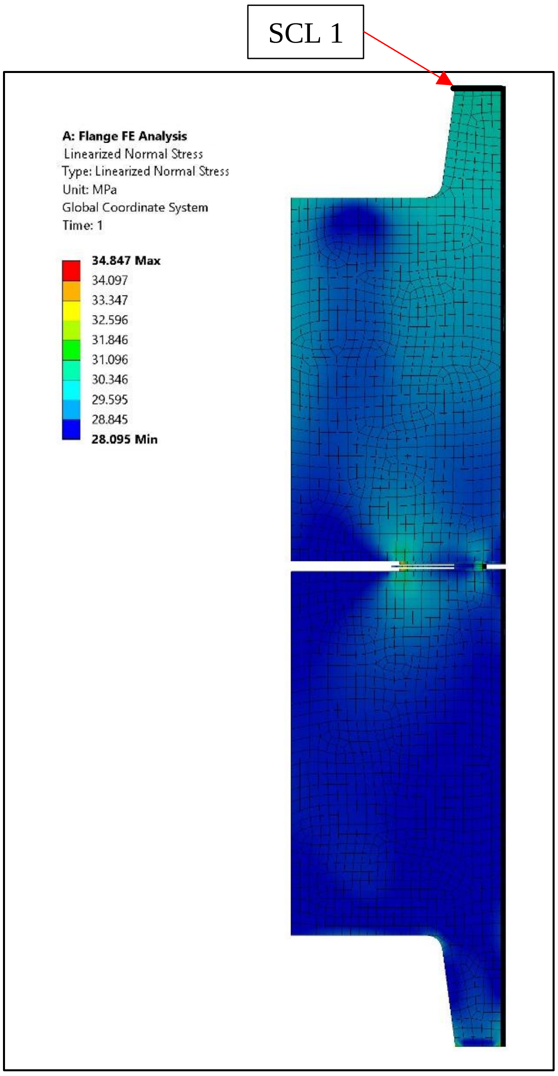
From figure 4.4, it is clear that the difference in values for Average and Unaverage Equivalent von-mises stresses is less than 5%, hence the mesh quality is adequate. Furthermore, the value of the stress is well under the allowable limit of 3S (103.2 MPa) as it represents local stress at the flange joint. Following Figure 4.5,4.6 and 4.7 represents the locations of SCL on the flange and the results for validation of flange stresses.

1) Longitudinal Hub Stress:



Details of "Linearized Normal Stress 2"	
Scope	
Scoping Method	Path
Path	Path 2
Geometry	All Bodies
Definition	
Type	Linearized Normal Stress
Orientation	Z Axis
Subtype	All
By	Time
Display Time	Last
Coordinate System	Global Coordinate System
Zero Through-Thickness Bending Stress	No
Suppressed	No

(a)

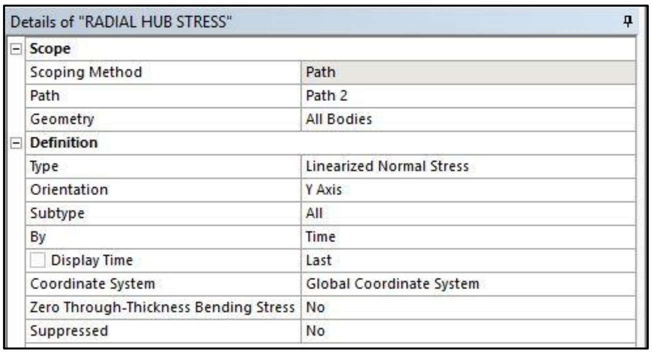


(b)

Figure 4.5 (a) Direction and orientation of SCL 1 (b) Longitudinal hub Stress

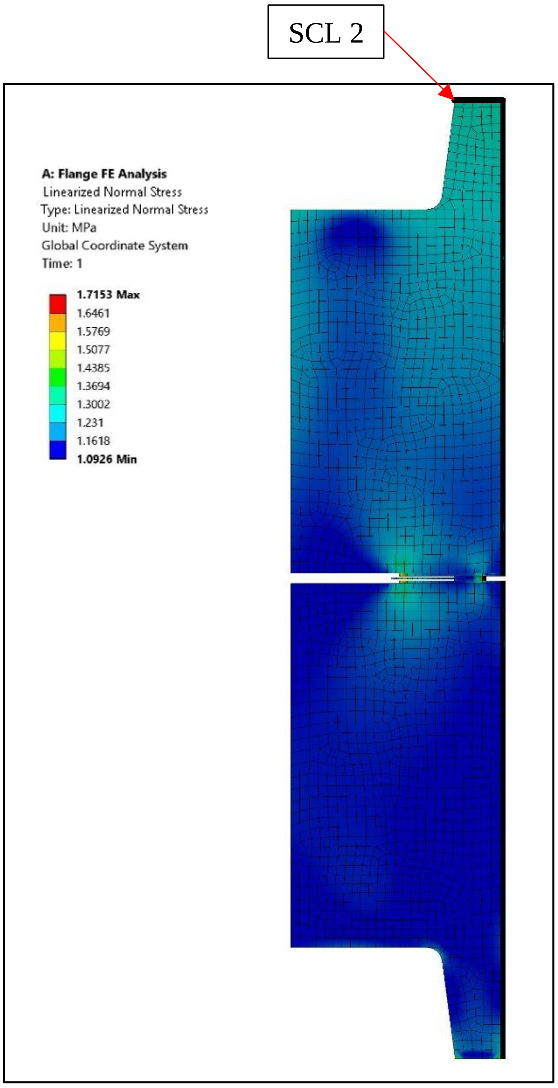
From Figure 4.5, the Linearized stress at the location of the SCL represents the Longitudinal hub stress at the location. Here, the SCL is taken in Z-axis direction which represents the behavior of Longitudinal stress in the flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

2) Radial Hub Stress:



Details of "RADIAL HUB STRESS"	
Scope	
Scoping Method	Path
Path	Path 2
Geometry	All Bodies
Definition	
Type	Linearized Normal Stress
Orientation	Y Axis
Subtype	All
By	Time
Display Time	Last
Coordinate System	Global Coordinate System
Zero Through-Thickness Bending Stress	No
Suppressed	No

(a)



(b)

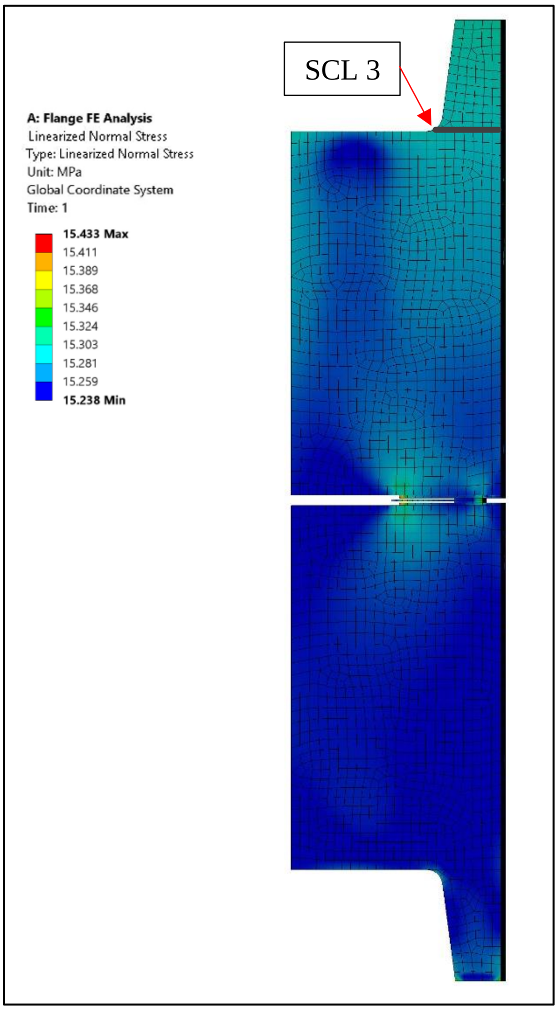
Figure 4.6 (a) Direction and orientation of SCL 2 (b) Radial hub Stress

From Figure 4.6, the Linearized stress at the location of the SCL represents the Radial hub stress at the location. Here, the SCL is taken in Y-axis direction which represents the behavior of Radial stress in the flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

3) Tangential Hub Stress:

Details of "TANGENTIAL HUB STRESS"	
Scope	
Scoping Method	Path
Path	Path 3
Geometry	All Bodies
Definition	
Type	Linearized Normal Stress
Orientation	X Axis
Subtype	All
By	Time
Display Time	Last
Coordinate System	Global Coordinate System
Zero Through-Thickness Bending Stress	No
Suppressed	No

(a)



(b)

Figure 4.7 (a) Direction and orientation of SCL 3 (b) Tangential hub Stress

From Figure 4.7, the Linearized stress at the location of the SCL represents the Radial hub stress at the location. Here, the SCL is taken in X-axis direction which represents the behavior of Radial stress in the flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

The total deformation plot for the flange under given boundary and loading conditions is shown in figure 4.8. Due to the compensating pressure acting on the hub on one of the flanges, it will try to break the flange joint which is opposed by the bolt loads. And on the other flange, the movement for flange is restricted only in radial direction to observe the behavior of the flange under the given loads. The total deformation for flange is very negligible as per figure 4.8 which shows that the flange joint will remain intact under operating conditions. There won't be any free contacts after the application of loads in the flange.

From the above study, it is clear that the theoretical calculations are substantiated and verified with the help of Finite Element Analysis of the flange as all the stress values are identical to the results obtained from the theoretical calculations.

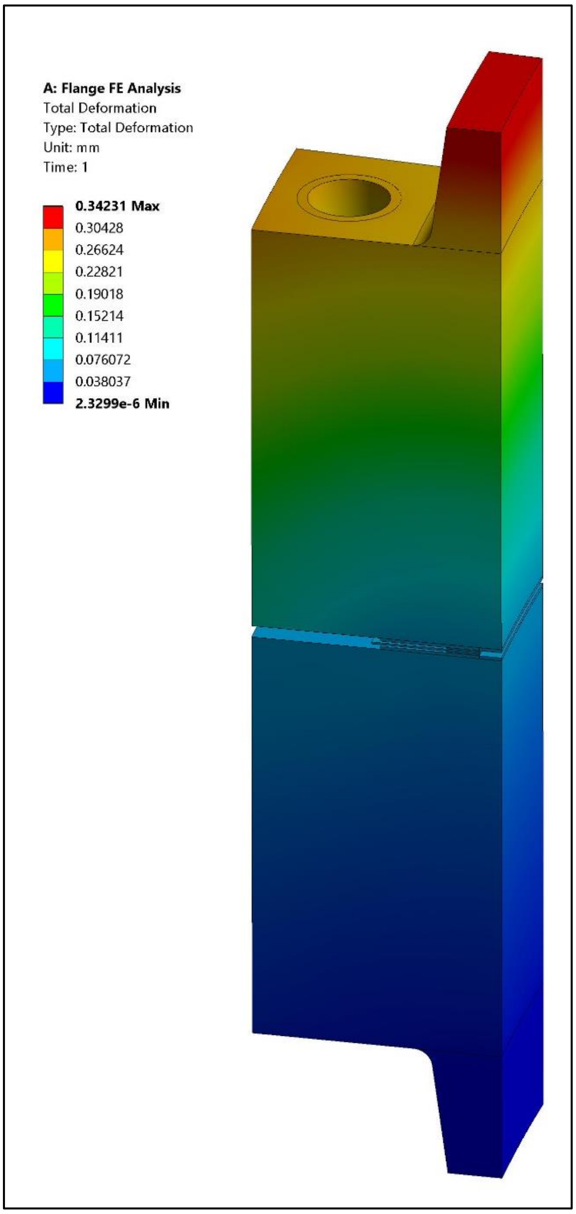


Figure 4.8 Total Deformation plot for the flange

From the above study, it is clear that the stresses induced on the flange are under the allowable limits of the stresses and there is a scope for optimization of the flange. While carrying out optimization of the flange, it must be noted that the optimized flange must also comply with ASME Section VIII Div. 1 Mandatory Appendix 2 calculations as well TEMA and so accordingly, the dimensions of the flange must be finalized. Parametric study is a good tool for carrying out optimization of the flange.

4.9 Introduction to Parametric Study

A parametric analysis, also known as a sensitivity analysis, is the study of the influence of different geometric or physical parameters or both on the solution of the problem. Parametric analysis is an important tool for the design aspect of the component. Many different sets of parameters are taken in order to observe the behaviour of the component as well as the effects of the parameters on the output stresses of the component. Here, for this case, the dimensional parameters are those that can be modified and optimal design can be obtained using the parameters. The dimensional parameters that can give optimized design are those which will contribute towards making the flange lighter i.e., towards the reduction of material of the flange so that the design would be optimal from a cost point of view. So, in this case, the input parameters that would contribute towards the reduction of weight of the flange is shown in the Table 4.5.

Sr. No.	Input Parameters	Output Parameters
1.	Diameter of Bolt circle	Longitudinal Hub Stress
2.	Outside Diameter of Flange	Radial Hub Stress
3.	Flange Thickness	Tangential Hub Stress
4.		Equivalent von-mises Stress

Table 4.5 Input and Output parameters for Parametric Study

The above input parameters largely contribute towards the reduction of material of the flange and hence by selecting different values for the input parameters, we can observe the behavior of the flange as well as study the best suitable parametric option out of all. At the same time, while selection for the values of the parameters, the parametric values must be in compliance with the ASME Section VIII Div. 1 Mandatory Appendix 2.

Following Table 4.6 represents different set of values of the input parameters for parametric study of the flange.

Input Parameters				Output Parameters			
Sr. No.	O.D. in mm	Bolt Circle Dia. (mm)	Thickness of Flange (mm)	Longi. Hub Stress (MPa)	Radial Hub Stress (MPa)	Tangential Hub Stress (MPa)	Equivalent Von- Mises Stress (MPa)
1.	5405	5263	387	40.21	1.42	10.242	58.845
2.	5405	5260	387	39.28	1.39	9.87	56.214
3.	5405	5260	383	40.52	1.46	10.16	60.014
4.	5405	5256	380	41.043	1.43	11.08	64.225
5.	5405	5255	375	41.89	1.49	11.41	65.287
6.	5400	5265	387	41.4	1.411	11.3	64.741
7.	5400	5265	380	42.82	1.503	12.17	67.229
8.	5400	5255	375	42.14	1.503	11.86	66.884
9.	5400	5255	368	43.68	1.606	12.92	68.543
10.	5400	5255	365	44.351	1.648	13.35	69.574
11.	5400	5255	363	44.81	1.6789	13.78	71.465
12.	5395	5255	366	44.393	1.648	12.47	68.486
13.	5389	5255	366	43.181	1.5046	10.497	70.117
14.	5395	5260	370	44.423	1.6171	11.68	67.541
15.	5410	5265	387	40.808	1.3905	9.421	57.575
16.	5410	5261	366	43.41	1.61	10.982	66.821
17.	5410	5261	369	42.75	1.56	10.06	63.784
18.	5410	5258	363	43.54	1.63	11.811	68.557

Table 4.6 Parametric study of Flange

From the Table 4.6, the results of the study suggests that the optimum design of the flange from the dimensional input parameters would be the 13^{th} row of the table. This is because it satisfies the calculations of ASME Section VIII Div. 1 Mandatory Appendix 2, the stresses are well under the permissible limit and this combination of dimensions will give the maximum reduction of the material of the flange. Now, from the above study, the results of the Optimized flange are discussed below.

4.10 Solid model for Optimized Flange:

The components of the vessel are as shown in figure 4.9. As the flange is having 104 number of bolts, we have taken 3.5-degree section of the flange joint in order to reduce the computational time required for the analysis. The flange joint consists of two identical flanges and a gasket between them. Lines found in model are due to slicing operation in geometry to facilitate part meshing. The dimensions of the flange and the gasket is shown in Table 4.7.

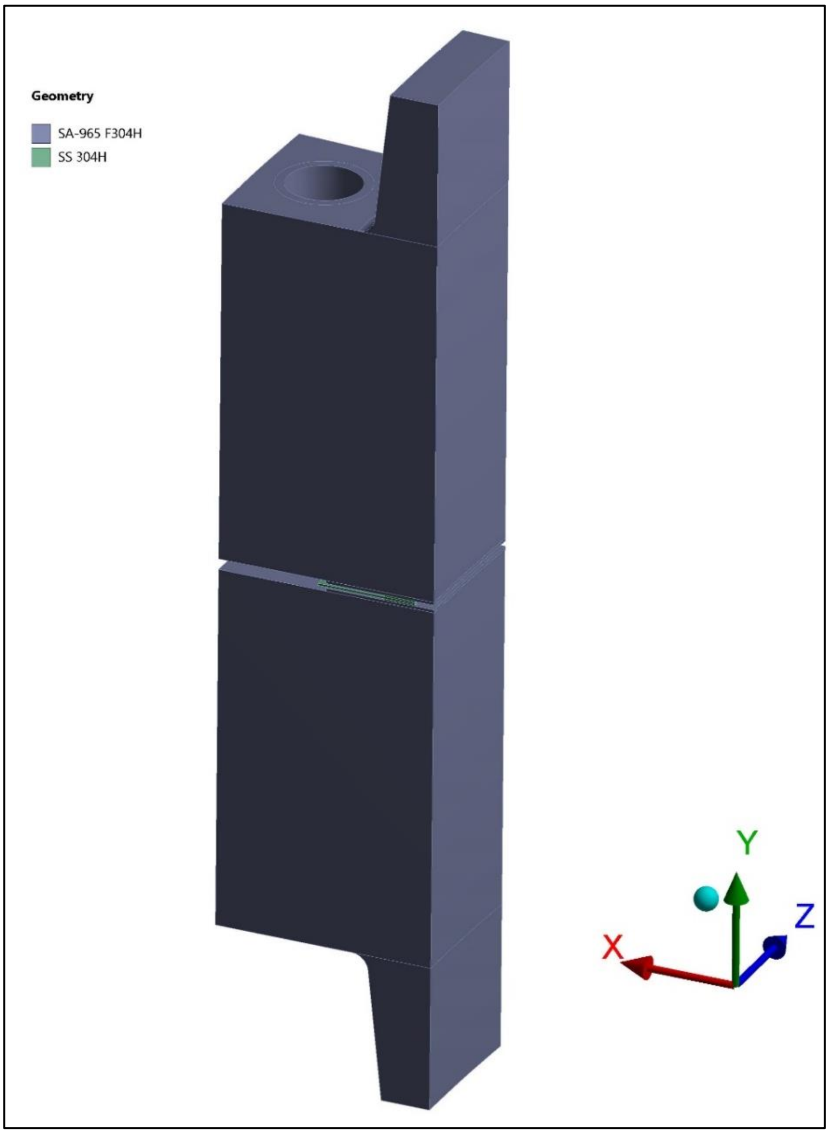


Figure 4.9 Geometry of the Flange

Sr. No.	Specifications	Dimensions (mm)
1.	Inside Diameter of Flange	4953
2.	Outside Diameter of Flange	5389
3.	Flange Thickness	366
4.	Thickness of hub at small end	49
5.	Thickness of hub at large end	65
6.	Length of Hub	120
7.	Diameter of bolt circle	5255
8.	Thickness of gasket	5
9.	Flange face Inside Diameter	4953
10.	Flange face Outside Diameter	5172
11.	Gasket Inside Diameter	4995
12.	Gasket Outside Diameter	5055
13.	Diameter of Bolt Hole	69.85

Table 4.7 Dimensions of the Optimized Flange

After obtaining the dimensions of the Optimized flange, it is necessary to validate the flange dimensions according to ASME Section VIII Div. 1 Mandatory Appendix 2 in order to proceed further. Table 4.8 shows the results of the calculations. For detail calculations refer Appendix I, as the calculations are repetitive.

Sr.	Specifications	Symbol	Value	Unit
1.	Basic Gasket seating Width	bo	15	mm
2.	Effective Gasket Width	b	9.76	mm
3.	Diameter at location of Gasket load reaction	G	5035.48	mm
4.	Total hydrostatic end force	H	8924471	N
5.	Total Joint contact Compression Load	Hp	415130.5	N
6.	Minimum required Bolt loads for operating conditions	$Wm1$	9339601	N
7.	Minimum required Bolt loads for Gasket seating	$Wm2$	10644371	N
8.	Total Cross-Sectional Area of Bolts required for the Operating conditions	$Am1$	230209.53	mm^2

9.	Total Cross-Sectional Area of Bolts required for the Gasket Seating	$Am2$	81879.77	mm^2
10.	Flange design Bolt load, for Operating Conditions	W	37899600	N
11.	Hydrostatic End Force on Area Inside of Flange	Hd	8645663	N
12.	Gasket Load	Hg	415130.16	N
13.	Difference between Total Hydrostatic End Force and the Hydrostatic End Force on Area Inside of Flange	Ht	278808.406	N
14.	Radial distance from the bolt circle, to the circle on which Hd acts	hd	117.7	mm
15.	Radial distance from gasket load reaction to the bolt circle	hg	109.7598	mm
16.	Radial distance from the bolt circle to the circle on which Ht acts	ht	129.5799	mm
17.	Component of moment due to Hd	Md	1018006912	N.mm
18.	Component of moment due to Ht	Mt	36142608	N.mm
19.	Component of moment due to Hg	Mg	45583068	N.mm
20.	Total moment acting upon the flange under Operating conditions	Mo	1099732608	N.mm
21.	Total moment acting upon the flange under Gasket Load	Mog	3845832192	N.mm
22.	Longitudinal Hub Stress	SH	39.945	Mpa
23.	Radial Flange Stress	SR	1.459	Mpa
24.	Tangential Flange Stress	ST	10.182	Mpa

Table 4.8 ASME Section VIII Div. 1 Mandatory Appendix 2 Calculations

4.11 Solid Element Plot for Optimized Flange:

Meshing is a key component to obtain accurate results from a FEA model. The elements in the mesh should account for taking up many aspects for discretization of stress gradients more precisely. Smaller the mesh size, more accurate the solution as the design is better sampled across the physical domains. However, smaller the mesh, more time is required for computation of analysis results.

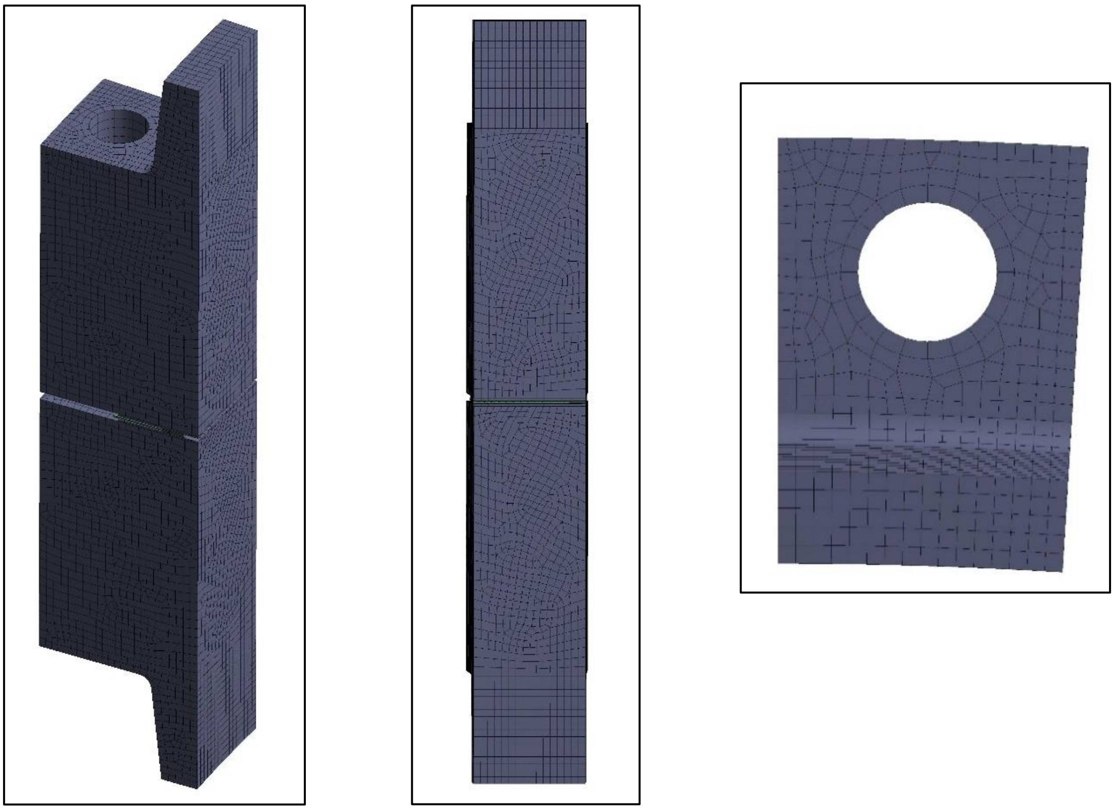


Figure 4.10 Meshed model of Optimized Flange

Analysis type	Static Structural, Mechanical
Element Type	Second order solid elements SOLID 186
Mesh Quality	
No. of nodes	2440966
No. of elements	654933
Aspect Ratio	3.457
Corner Angle	99.41°
Jacobian Ratio	1.1596

Table 4.9 Mesh Details for Optimized Flange

4.12 Loads and Boundary Conditions for Optimized Flange:

The cylindrical co-ordinate system is used for the analysis so as to keep the flange radially free to move and to fix the bottom hub face of the flange using nodal displacement free only in radial direction and restrict it to move in all other directions. For evaluating the stresses due to mechanical loads, the bottom nodes of the flange are fixed in all directions. Internal pressure (P) of 4.48 bar (i.e., 0.448 MPa) is applied on the inner faces of the flange and the gasket. Frictionless support is provided in order to restrict the faces of the flange to deform or move in the normal direction so that correct behavior of the flange could be obtained. Bolt loads are applied on the bolt contact faces in form of forces (Force and Force 2 in figure 4.11). The bolt loads applied in the analysis are corresponding to single bolt load as the flange is considered for a 3.5-degree section. A compensating pressure (Pressure 2 in figure 4.11) is applied on the top of the flange hub face. The compensating pressure is calculated as per the given formula:

$$Pc = \frac{Pi}{\left(\frac{Do}{Di}\right)^2 - 1}$$

$$= \frac{4.48}{\left(\frac{5051}{4953}\right)^2 - 1}$$

$$= 11.2 \text{ Mpa}$$

Where, Pc = Compensating pressure

Pi = Internal Pressure acting on the flange

Do = Outside diameter of opening of the flange

Di = Inside diameter of opening of the flange

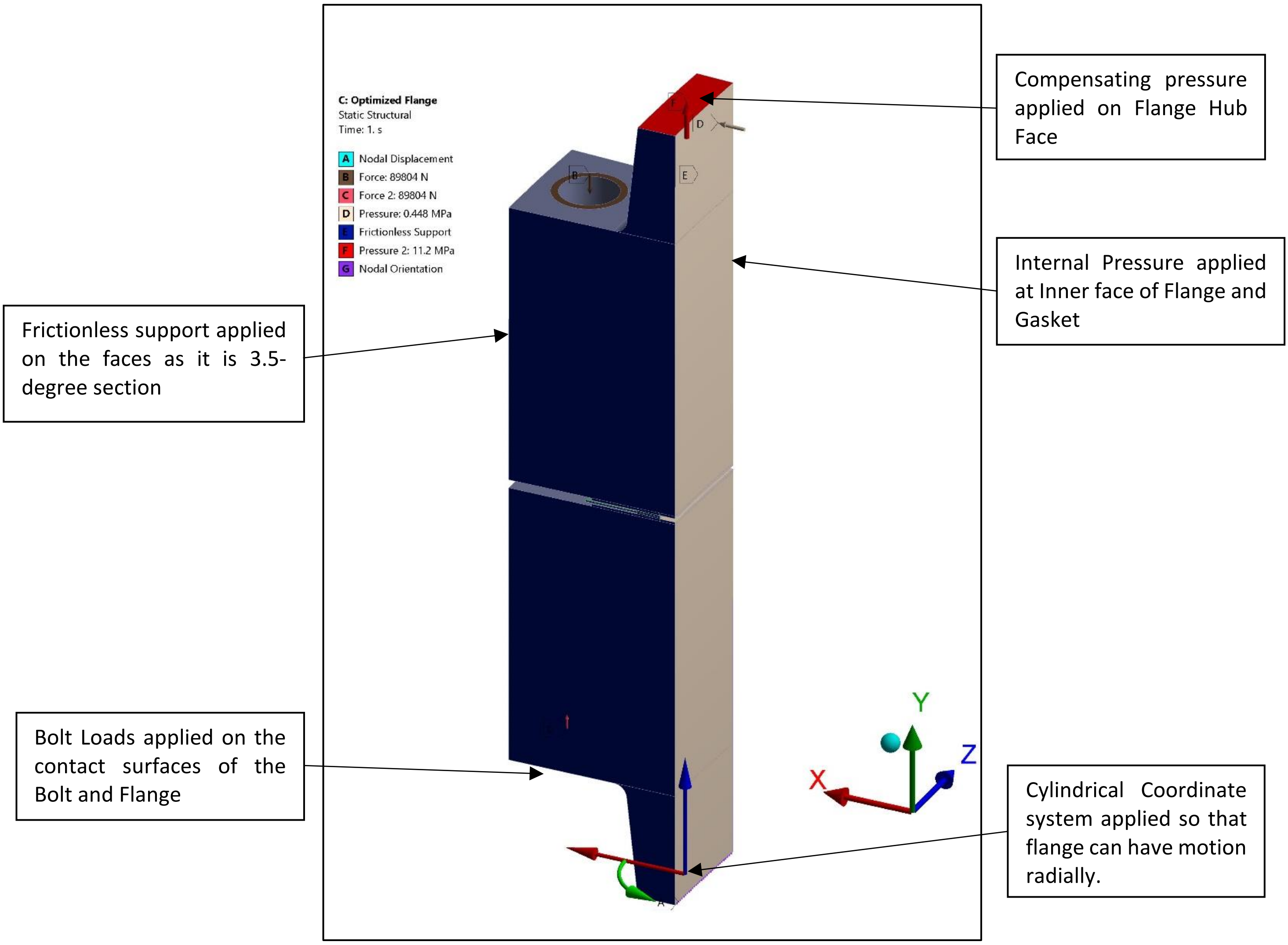
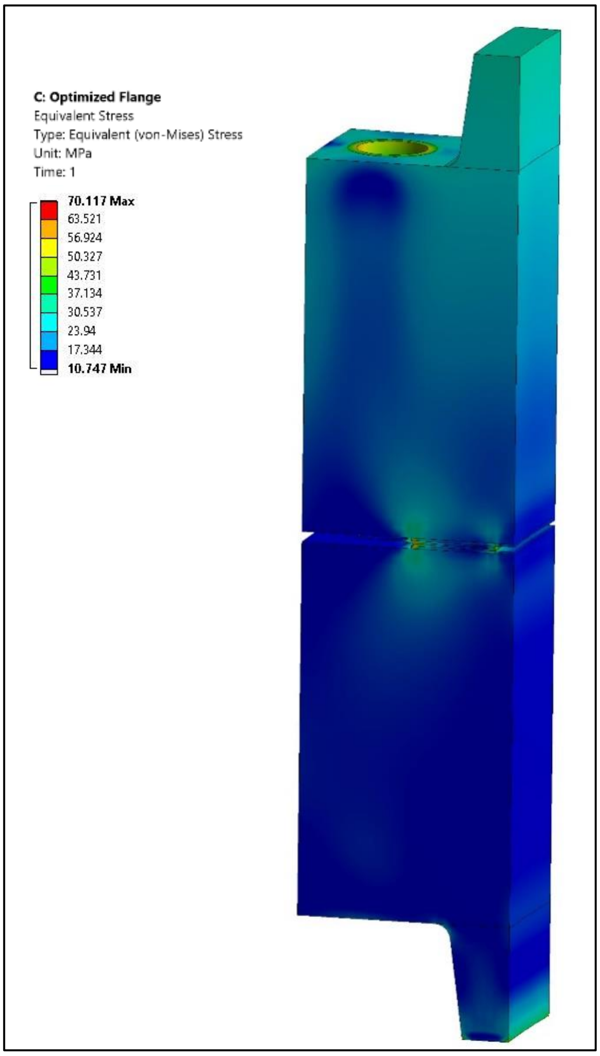


Figure 4.11 Loadings and boundary conditions acting on Optimized Flange

4.13 Results and Discussion for Optimized Flange:

The results for applied boundary and loading conditions are indicated in the Figure 4.12. Here, for validation of the mesh quality in the analysis, the average and the unaverage Equivalent von-mises stresses are compared simultaneously. If the difference in the values of average and unaverage Equivalent von-mises stresses are under 5% respectively, then the mesh quality is acceptable. In addition to that, to validate the results of theoretical calculations obtained from ASME Section VIII Div. 1 Mandatory Appendix 2, the stresses required while conducting the Finite element analysis are obtained through stress linearization. Stress linearization is a procedure in which the stress distribution along a line or section through the thickness in a solid is approximated with an equivalent linear stress distribution. The classes of stress i.e., primary, secondary as well as the peak stresses are evaluated and limited according to their allowable limits as guided in ASME Section VIII Division 2. Stress classification planes (SCPs) are transection across the thickness of a component through which membrane and bending stresses are developed. For planar geometry, the stress classification line (SCL) is taken by decreasing the two opposing edges of a SCP to a minute extent. These SCLs should be orthogonal to one of the two, maximum stress component or the mid-surface of the section, and are usually taken at very local structural or material discontinuities in order to examine the part in case of plastic collapse as well as the local failure.



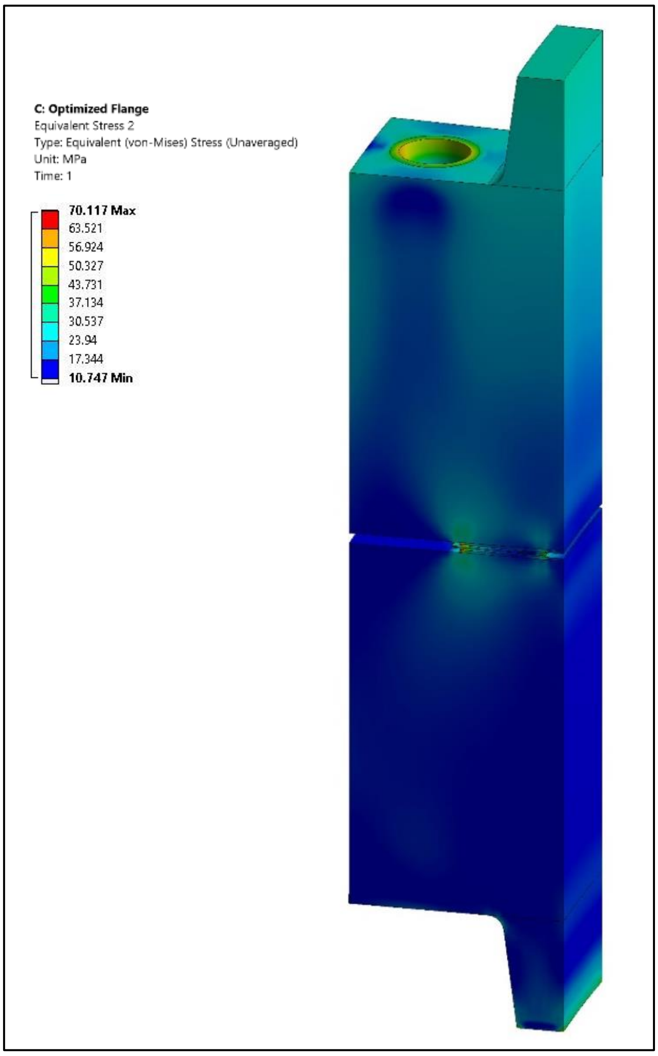
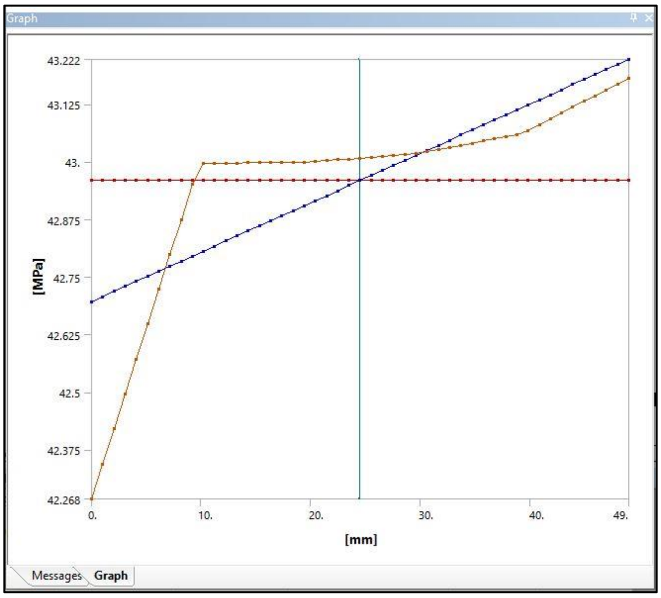


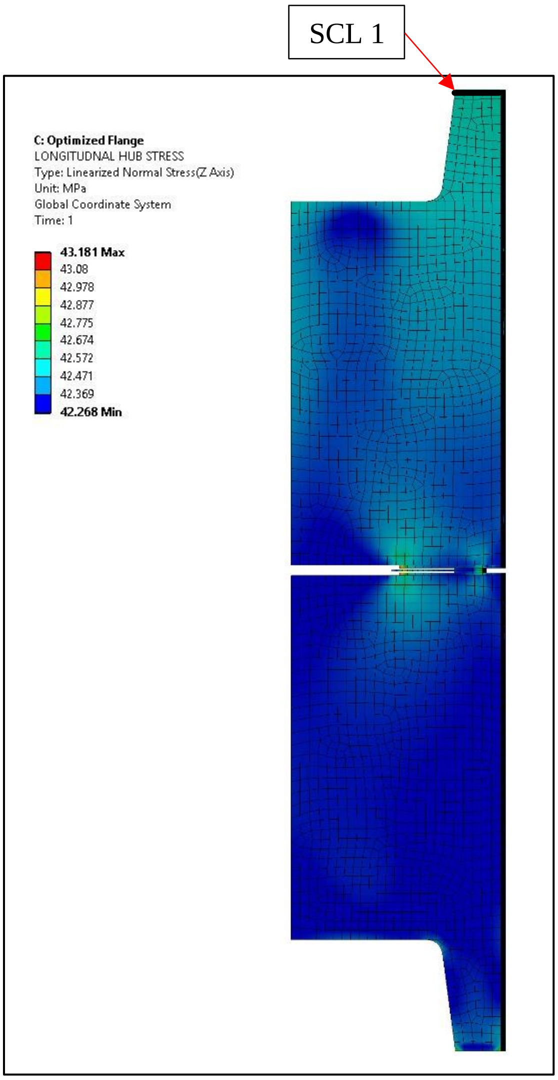
Figure 4.12 Average and Unaverage Equivalent von-mises Stresses for Optimized Flange

From figure 4.12, it is clear that the difference in values for Average and Unaverage Equivalent von-mises stresses is less than 5%, hence the mesh quality is adequate. Furthermore, the value of the stress is well under the allowable limit of 3S (103.2 MPa) as it represents local stress at the flange joint. Following Figure 4.13, 4.14 and 4.15 represents the locations of SCL taken for flange and moreover, the results for validation of flange stresses.

1) Longitudinal Hub Stress:



(a)



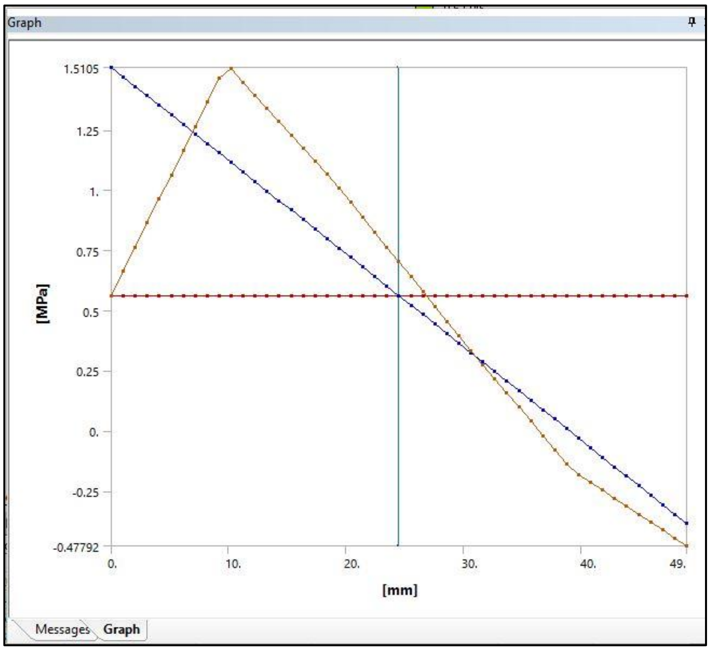
(b)

Figure 4.13 (a) Membrane and membrane plus bending stress v/s length of SCL 1

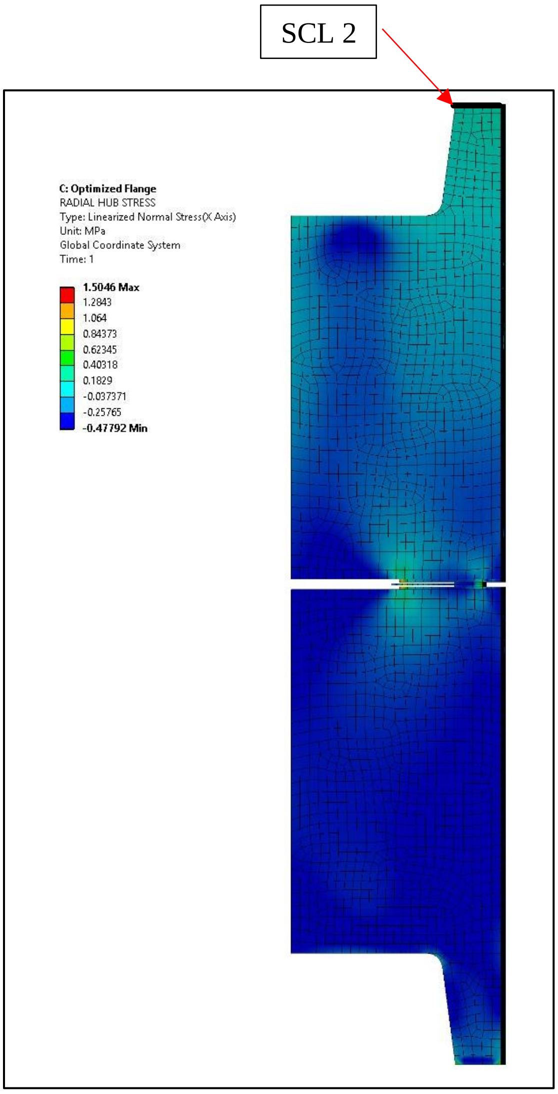
(b) Longitudinal hub Stress of Optimized Flange

From Figure 4.13, the Linearized stress at the location of the SCL represents the Longitudinal hub stress at the location. Here, the SCL is taken in Z-axis direction which represents the behavior of Longitudinal stress in the optimized flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

2) Radial Hub Stress:



(a)

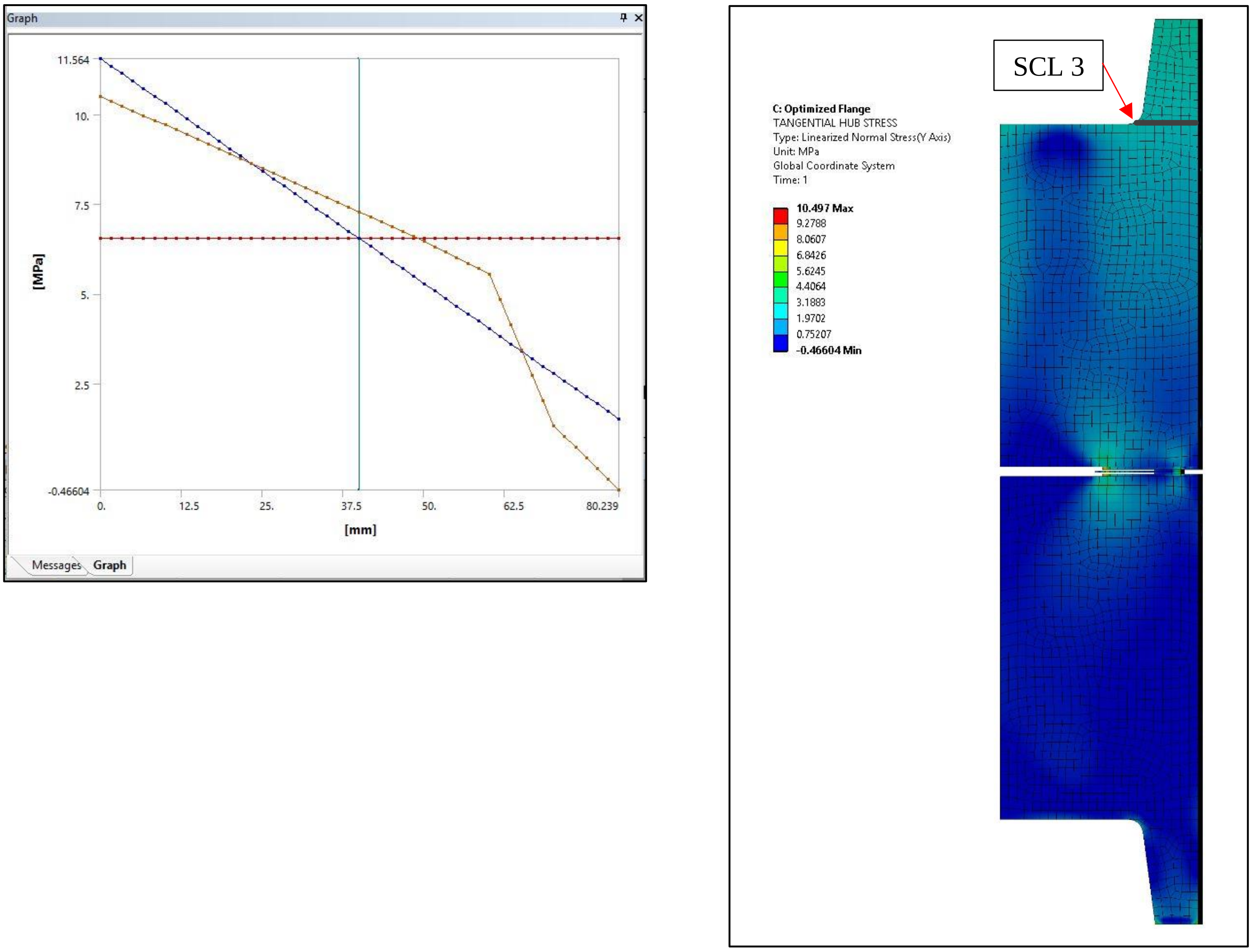


(b)

Figure 4.14 (a) Membrane and membrane plus bending stress v/s length of SCL 2
(b) Radial hub Stress of Optimized Flange

From Figure 4.14, the Linearized stress at the location of the SCL represents the Radial hub stress at the location. Here, the SCL is taken in Y-axis direction which represents the behavior of Radial stress in the optimized flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

3) Tangential Hub Stress:



(b)

Figure 4.15 (a) Membrane and membrane plus bending stress v/s length of SCL 3

(b) Tangential hub Stress of Optimized Flange

From Figure 4.15, the Linearized stress at the location of the SCL represents the Radial hub stress at the location. Here, the SCL is taken in X-axis direction which represents the behavior of Radial stress in the optimized flange. The above value of stress obtained is same as the value obtained from the theoretical calculations as per ASME Section VIII Div. 1 Mandatory Appendix 2. The difference in values of stress obtained from Finite Element Analysis and ASME Section VIII Div. 1 Mandatory Appendix 2 is below 5% which is acceptable.

The total deformation plot for the flange under given boundary and loading conditions is shown in figure 4.16. Due to the compensating pressure acting on the hub on one of the flanges, it will try to break the flange joint which is opposed by the bolt loads. And on the other flange, the movement for flange is restricted only in radial direction to observe the behavior of the flange under the given loads. The total deformation for flange is very negligible as per figure 6.8 which shows that the flange joint will remain intact under operating conditions. There won't be any free contacts after the application of loads in the flange.

From the above study, it is clear that the theoretical calculations are substantiated and verified with the help of Finite Element Analysis of the flange as all the stress values are identical to the results obtained from the theoretical calculations.

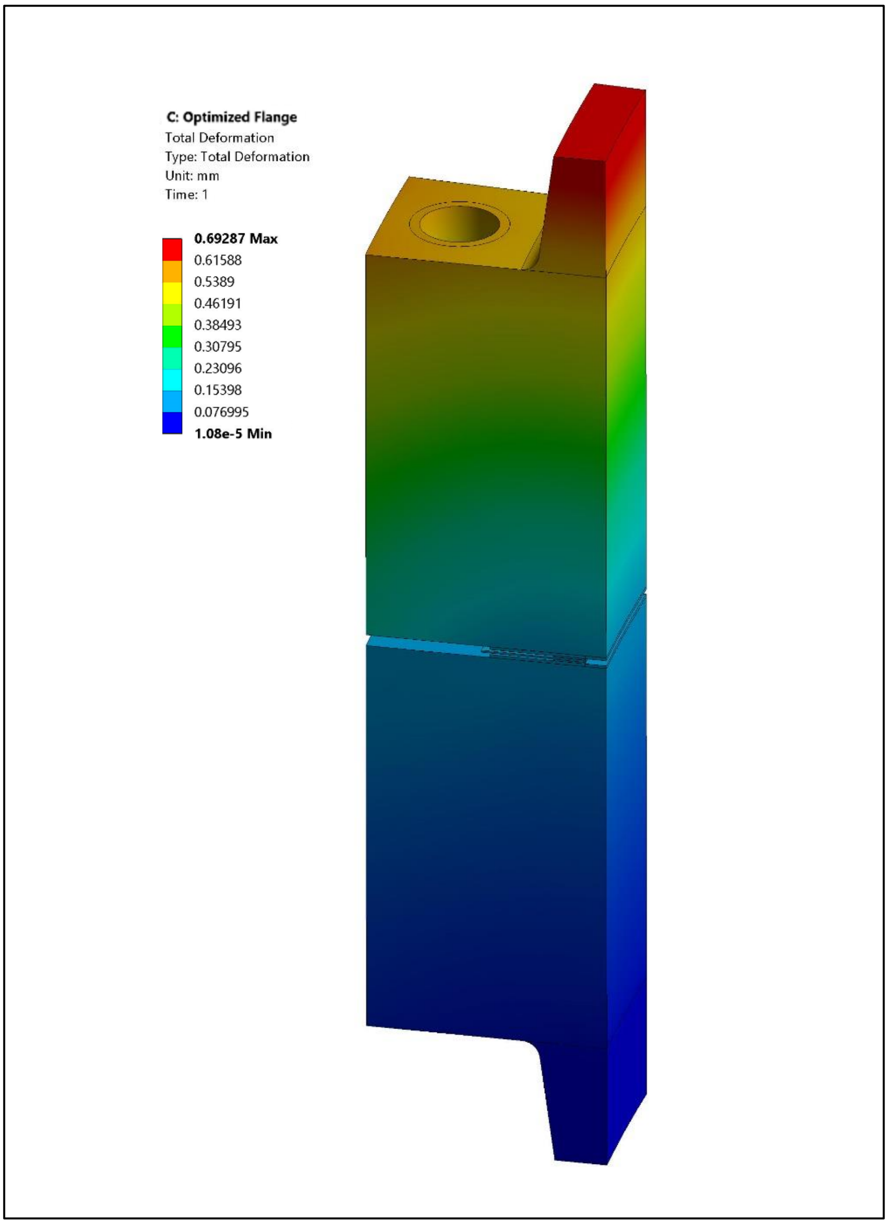


Figure 4.16 Total Deformation plot for Optimized flange

CHAPTER 5

Results and Conclusion:

5.1 Weight of the given Flange:

The dimensions of the flange which is designed according to ASME Section VIII Div. 1 and ASME PCC -1 are as follows:

Flange I.D =B= 4953 mm

Flange O.D = A=5405 mm

Raised Face = RF = 3 mm

R.F Face I.D of Flange = $RF.ID$ = 4953 mm

R. F Face O. D of Flange = $RF.\ OD$ = 5172 mm

g_0 = 49 mm

g_1 = 65 mm

Height of hub = h =120 mm

Thickness of Flange = t = 392 mm

Bolt Circle Diameter of the Flange = C = 5265 mm

Number of bolts = nb = 104

Nominal Diameter of Bolt = Dh = 69.85 mm

Now, Volume of the Flange,

$$Vf = \frac{\pi}{4} * (A^2 - B^2) * t \tag{7.1}$$

$$= \frac{\pi}{4} * (5405^2 - 4953^2) * 392$$

$$= 1441419158 \text{ mm}^3$$

Volume of the Raised Face surface,

$$Vrf = \frac{\pi}{4} * (RF.OD^2 - RF.ID^2) * RF \tag{7.2}$$

$$= \frac{\pi}{4} * (5172^2 - 4953^2) * 3$$

$$= 5224566 \text{ mm}^3$$

Volume occupied by the bolts,

$$Vb = \frac{\pi}{4} * Dh^2 * nb * t \tag{7.3}$$

$$= \frac{\pi}{4} * (69.85^2 * 104 * 392)$$

$= 156221969\ mm^3$

Volume of Hub,

$$Vh = \frac{\left(\frac{\pi}{4} * ((B + 2g1)^2 - (B + 2g0)^2) * h\right)}{2} + \left(\left(\frac{\pi}{4} * (B + 2 * go)^2 - B^2\right) * h\right) \tag{7.4}$$

$$= \frac{\left(\frac{\pi}{4} * (5083^2 - 5051^2) * 120\right)}{2} + \left(\frac{\pi}{4} * (5051^2 - 4953^2) * 120\right)$$

$= 107681481\ mm^3$

Total Volume of the Flange,

$$V = Vf + Vrf + Vh - Vb \tag{7.5}$$

$$= 1441419158 + 5224566 + 107681481 - 156221969$$

$= 1398103236\ mm^3$

Now, the density of the Flange Material SA-965 Gr. F304 according to ASME Sec. II Part -D Table PRD is $\rho = 8030\ kg/m^3$.

Hence, the Weight of Flange,

$$W = \frac{V * \rho}{1000000000} \tag{7.6}$$

$$= \frac{1398103236 * 8030}{1000000000}$$

= 11226 kg.

5.2 Weight of the Optimized Flange:

The dimensions of the flange which is designed according to ASME Section VIII Div. 1 and ASME PCC -1 are as follows:

Flange I.D =*B*= 4953 mm

Flange O.D = *A*=5389 mm

Raised Face = *RF* = 3 mm

R.F Face I.D of Flange = *RF.ID* = 4953 mm

R. F Face O. D of Flange = *RF. OD* = 5172 mm

g_0 = 49 mm

g_1 = 65 mm

Height of hub = *h* =120 mm

Thickness of Flange = *t* = 366 mm

Bolt Circle Diameter of the Flange = *C* = 5255 mm

Number of bolts = *nb* = 104

Nominal Diameter of Bolt = *Dh* = 69.85 mm

Now, Volume of the Flange,

$$Vf = \frac{\pi}{4} * (A^2 - B^2) * t \tag{7.7}$$

$$= \frac{\pi}{4} * (5389^2 - 4953^2) * 366$$

$$= 1296170072 \text{ mm}^3$$

Volume of the Raised Face surface,

$$Vrf = \frac{\pi}{4} * (RF.OD^2 - RF.ID^2) * RF \tag{7.8}$$

$$= \frac{\pi}{4} * (5172^2 - 4953^2) * 3$$

$$= 5224566 \text{ mm}^3$$

Volume occupied by the bolts,

$$Vb = \frac{\pi}{4} * Dh^2 * nb * t \tag{7.9}$$

$$= \frac{\pi}{4} * (69.85^2 * 104 * 366)$$

$$= 145860308 \text{ mm}^3$$

Volume of Hub,

$$Vh = \frac{\left(\frac{\pi}{4} * ((B + 2g1)^2 - (B + 2g0)^2) * h\right)}{2} + \left(\left(\frac{\pi}{4} * (B + 2 * go)^2 - B^2\right) * h\right) \tag{7.10}$$

$$= \frac{\left(\frac{\pi}{4} * (5083^2 - 5051^2) * 120\right)}{2} + \left(\frac{\pi}{4} * (5051^2 - 4953^2) * 120\right)$$

$$= 107681481 \text{ mm}^3$$

Total Volume of the Flange,

$$V = Vf + Vrf + Vh - Vb \tag{7.11}$$

$$= 1296170072 + 5224566 + 107681481 - 145860308$$

$$= 1263215811 \text{ mm}^3$$

Now, the density of the Flange Material SA-965 Gr. F304 according to ASME Sec. II Part -D Table PRD is ρ = 8030 kg/m^3.

Hence, the Weight of Flange,

$$W = \frac{V * \rho}{1000000000} \tag{7.12}$$

$$= \frac{1263215811 * 8030}{1000000000}$$

$$= 10143 \text{ kg.}$$

Hence, material saved for the Optimized flange = $11226 - 10143 = 1083$ kg.

5.3 Conclusion:

From the study it is obtained that the optimized flange is way lighter than the design of the flange at the initial stage. Because of which, the material cost for the flange is reduced. The above study also states that the stresses induced in the optimized flange are way under the allowable limits of the stress values. Hence, by conducting the parametric study and optimizing the dimensional parameters of the flange, the design can be improved. The proposed flange design is economical and, in a way, a cost-effective solution considering the entire cost for the flange.

In this case, while carrying out the parametric study, the requirement for the flange to be adequate according to ASME Sec. VIII Div. 1 Mandatory Appendix 2 is readily satisfied and then the optimized design is derived such that there is no code violation during the process of optimization of the flange. Hence, under operating conditions, the proposed flange design is the best suitable solution from design point of view.

References:

[1] Nash, D.H., Spence, J., Tooth, A.S., Abid, M. & Power, D.J. 2000, "A parametric study of metal-to-metal full face taper-hub flanges", International Journal of Pressure Vessels and Piping, vol. 77, no. 13, pp. 791-797.

[2] Bouzid, L. & Chaaban, A. 1997, "An accurate method of evaluating relaxation in bolted flanged connections", Journal of Pressure Vessel Technology, Transactions of the ASME, vol. 119, no. 1, pp. 10-17.

[3] Tsuji, H. & Nakano, M. 2002, "Bolt preload control for bolted flange joint", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 163.

[4] Abid, M. & Nash, D.H. 2003, "Comparative study of the behaviour of conventional gasketed and compact non-gasketed flanged pipe joints under bolt up and operating conditions", International Journal of Pressure Vessels and Piping, vol. 80, no. 12, pp. 831-841.

[5] Nechache, A. & Bouzid, A.-. 2007, "Creep analysis of bolted flange joints", International Journal of Pressure Vessels and Piping, vol. 84, no. 3, pp. 185-194.

[6] Mohamed, B.Y., Hamdy, M.A. & Eid, T.I. 2017, "Determination of temperature limits for heat exchanger joint assembled of solid stainless tube sheet with girth flanges", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP.

[7] Takahashi, K., Miyashita, T., Kataoka, S., Uno, Y. & Sato, T. 2017, "Effects of partial cooling on tightness of heat exchanger girth flange", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP.

[8] Wang, L., Chen, X., Fan, Z. & Xue, J. 2018, "FEM stress analysis of bolted flange joints in elevated temperature service condition", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP.

[9] Fessler, H. & Perry, D.A. 1977, "Leakage characteristics of flanged pipe joints", The Journal of Strain Analysis for Engineering Design, vol. 12, no. 1, pp. 29-36.

[10] Morohoshi, T. & Sawa, T. 1994, "On the characteristics of rectangular bolted flanged connections with gaskets subjected to external tensile loads and bending moments", Journal of Pressure Vessel Technology, Transactions of the ASME, vol. 116, no. 2, pp. 207-215.

[11] Kumano, H., Sawa, T. & Hirose, T. 1994, "Mechanical behavior of bolted joints under steady heat conduction", Journal of Pressure Vessel Technology, Transactions of the ASME, vol. 116, no. 1, pp. 42-48.

[12] Chikhaliya, K.M. & Patel, B.P. 2019, "Optimization and Standardization of Flanged and Flued Expansion Joint Design", Journal of Pressure Vessel Technology, Transactions of the ASME, vol. 141, no. 3.

[13] Chu, M., Su, C. & Mi, X. 2018, "Performance Analysis of Flange Seal Assembly for Heat Exchanger", Proceedings of the 2018 IEEE 22nd International Conference on Computer Supported Cooperative Work in Design, CSCWD 2018, pp. 17.

[14] Morita, Y. & Kawashima, H. 1980, "STRENGTH OF INTEGRAL PIPE FLANGES - 3. THE FORMULA FOR CALCULATING THE MAXIMUM MERIDIONAL STRESS).", Bulletin of the JSME, vol. 23, no. 181, pp. 1245-1251.

[15] Bouzid, A.-. & Beghoul, H. 2003, "The design of flanges based on flexibility and tightness", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 31.

[16] Sawa, T., Takagi, Y., Kawasaki, N. & Torii, H. 2008, "The sealing performance of a large diameter bolted joint under elevated temperature", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 201.

[17] Omiya, Y. & Sawa, T. 2010, "Thermal stress analysis and the sealing performance evaluation of bolted flange connection at elevated temperature", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 397.

[18] Bouzid, A.-., Diany, M. & Derenne, M. 2004, "Determination of gasket effective width based on leakage", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 105.

[19] Nagata, S. & Sawa, T. 2008, "Effects of temperature change on bolt load and gasket load of bolted flange connection with ring type joint gasket", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP, pp. 69.

[20] Sato, T. & Kado, K. 2006, "Inelastic analysis of sealing characteristics of flanges with metal ring joint gasket at elevated temperature", American Society of Mechanical Engineers, Pressure Vessels and Piping Division (Publication) PVP.

[21] ASME Sec. VIII Div. 2 Part 5.

[22] ASME Sec. VIII Div. 1.

American Society of Mechanical Engineers – https://www.asme.org/

L&T Heavy Engineering - https://www.larsentoubro.com/heavy-engineering/

APPENDIX-I

Here, Appendix-I shows the calculations of the optimized flange according to ASME Section VIII Div. 1 Mandatory Appendix 2.

- **Gasket Design:**

From Table 2-5.2 of Appendix 2,

Basic Gasket seating Width,

$$bo = \frac{N}{2} \qquad \text{(Eq. 16)}$$

$$= \frac{30}{2}$$

$$= 15 \text{ mm.}$$

Effective Gasket Width,

$$b = 2.5 * \sqrt{bo} \qquad \text{(Eq. 2)}$$

$$= 2.5 * \sqrt{15}$$

$$= 9.76 \text{ mm.}$$

Diameter at location of Gasket load reaction [12],

$$G = G.O.D - 2b \qquad \text{(Eq. 3)}$$

$$= 5055 - (2 * 9.76)$$

$$= 5035.48 \text{ mm.}$$

- **Bolting Loadings and sizing of bolts with the designed Gasket:[19]**

Total hydrostatic end force,

$$H = 0.785G^2P \qquad \text{(Eq. 4)}$$

$$= 0.785 * 0.448159 * 5035.315^2$$

$$= 8924471 \text{ N}$$

Total Joint contact Compression Load,

$$Hp = 2b * \pi GmP \qquad \text{(Eq. 5)}$$

$$= 2 * 9.76 \ * \pi * 5035.315 * 3 * 0.448159$$

$$= 415130.5 \text{ N}$$

Minimum required Bolt loads for operating conditions,

$$Wm1 = Hp + H \qquad \text{(Eq. 6)}$$

$$= 8924471 + 415130.5$$

$$= 9339601 \text{ N}$$

Minimum required Bolt loads for Gasket seating,

$$Wm2 = \pi bGy \tag{Eq. 7}$$

$$= \pi * 9.76 * 5035.315 * 68.95$$

= 10644371 N

Total Cross-Sectional Area of Bolts at Root of Thread or Section of least diameter under stress, required for the Operating conditions,

$$Am1 = \frac{Wm1}{Sb} \tag{Eq. 8}$$

$$= \frac{9339601}{40.57}$$

= 230209.53 mm^2

Total Cross-Sectional Area of Bolts at Root of Thread or Section of least diameter under stress, required for the Gasket Seating [5],

$$Am2 = \frac{Wm2}{Sa} \tag{Eq. 9}$$

$$= \frac{10644371}{130}$$

= 81879.77 mm^2.

Total required cross-sectional area of bolts,

$$Am = Greater\ of\ Am1\ and\ Am2$$

Here, $Am1 > Am2$

Hence, $Am = 230209.53$ mm^2.

According to TEMA Section 9- Table D-5M, for the assumed diameter of bolts, Root Area of single bolt, $Ab = 3392.8964$ mm^2.

Flange design Bolt load, for Operating Conditions,

$$W = \frac{Sa}{2} * (Am1 + Ab) \tag{Eq. 170}$$

$$= \left(\frac{130}{2}\right) * (230209.53 + 352861.225)$$

= 37899600 N

- **Total Flange Loads and Flange Moments:**

Flange Loads:

Hydrostatic End Force on Area Inside of Flange,

$$Hd = 0.785B^2P \quad \text{(Eq. 181)}$$

$$= 0.785 * (4956.2)^2 * 0.448159$$

$$= 8645663 \text{ N}$$

Gasket Load [2],

$$Hg = Wm1 - H \quad \text{(Eq. 192)}$$

$$= 9339601 - 8924471$$

$$= 415130.16 \text{ N}$$

Difference between Total Hydrostatic End Force and the Hydrostatic End Force on Area Inside of Flange [11],

$$Ht = H - Hd \quad \text{(Eq. 203)}$$

$$= 8924471 - 8645663$$

$$= 278808.406 \text{ N}$$

Lever Arms,

Radial distance from the bolt circle, to the circle on which Hd acts [11],

$$hd = R + 0.5g1 \quad \text{(Eq. 214)}$$

where, $R = \left(\frac{C-B}{2}\right) - g1$ (Eq. 225)

$$= \left(\frac{5255-4953}{2}\right) - 65$$

= 86 mm.

$$hd = 86 + (0.5 * 63.4)$$

$$= 117.7 \text{ mm}$$

Radial distance from gasket load reaction to the bolt circle,

$$hg = \frac{C - G}{2} \quad \text{(Eq. 236)}$$

$$= \frac{5255-5035.315}{2}$$

= 109.7598 mm

Radial distance from the bolt circle to the circle on which Ht acts [18],

$$ht = \frac{R + g1 + hg}{2} \quad \text{(Eq. 247)}$$

$$= \frac{86+63.4+109.7598}{2}$$

= 129.5799 mm

Flange Moments [7]:

Component of moment due to Hd,

$$Md = Hd * hd \quad \text{(Eq. 258)}$$

$$= 8645663 * 117.7$$

= 1018006912 N.mm

Component of moment due to Ht,

$$Mt = Ht * ht \quad \text{(Eq. 269)}$$

$$= 278808.406 * 129.5799$$

= 36142608 N.mm

Component of moment due to Hg,

$$Mg = Hg * hg \quad \text{(Eq. 20)}$$

$$= 415130.16 \quad * 109.7598$$

= 45583068 N.mm

Total moment acting upon the flange under Operating conditions,

$$Mo = Md + Mt + Mg \quad \text{(Eq. 21)}$$

$$= 1060822850.1 \ + 37519247.19 + 47636185.85$$

= 1099732608 N.mm

Total moment acting upon the flange under Gasket Load,

$$Mog = W * hg \quad \text{(Eq. 22)}$$

$$= 37899600 * 114.76$$

= 3845832192 N.mm

Taking the higher value from the above-mentioned total moments,

$$Mo > Mog$$

Hence, the total moment for the flange considered is $Mo = 1099732608$ N.mm

➢ **Shape Constants used for calculations [13]:**

1) Shape Constant K:

$$K = \frac{A}{B} \tag{Eq. 23}$$

$$= \frac{5389}{4953}$$

$$= 1.087$$

2) Shape Constant T:

$$T = \frac{K^2(1 + 8.55246\, log_{10} K) - 1}{(1.04720 + 1.9448K^2)(K - 1)} \tag{Eq. 24}$$

$$= \frac{1.087^2(1+8.55246 \log_{10} 1.087)-1}{(1.04720+1.9448(1.087)^2)(1.087-1)}$$

$$= 1.882$$

3) Shape Constant U:

$$U = \frac{K^2(1 + 8.55246\, log_{10} K) - 1}{(1.36136(K^2 - 1))(K - 1)} \tag{Eq. 25}$$

$$= \frac{1.087^2(1+8.55246\, log_{10} 1.087)-1}{(1.36136(1.087^2-1))(1.087-1)}$$

$$= 25.378$$

4) Shape Constant Y:

$$Y = \frac{1}{k-1}\left[0.66845 + 5.71690\, \frac{k^2\, log_{10} k}{k^2 - 1}\right] \tag{Eq. 26}$$

$$= \frac{1}{(1.087-1)}\left[0.66845 + 5.71690\, \frac{1.087^2\, log_{10} 1.087}{1.087^2-1}\right]$$

$$= 23.094$$

5) Shape Constant Z:

$$Z = \frac{k^2 + 1}{k^2 - 1} \tag{Eq. 27}$$

$$= \frac{1.087^2+1}{1.087^2-1}$$

$$= 11.972$$

6) g1/g0:

$$\frac{g1}{g0} = \frac{63.4}{47.4} = 1.338 \qquad \text{(Eq. 28)}$$

7) h0 :

$$h0 = \sqrt{Bg0} = \sqrt{(4956.2 * 47.4)} = 484.69 \qquad \text{(Eq. 29)}$$

8) h / h0 :

$$\frac{h}{h0} = \frac{120}{484.69} = 0.248 \qquad \text{(Eq. 30)}$$

9) From Fig. 2-7.2, ASME Section VIII Div.1 Mandatory App.2, F = 0.891

From Fig. 2-7.3, ASME Section VIII Div.1 Mandatory App.2, V = 0.425

From Fig. 2-7.6, ASME Section VIII Div.1 Mandatory App.2, f = 1.042

10) Factors e and d:

$$e = \frac{F}{h0} = \frac{0.891}{484.69} = 0.0018 \qquad \text{(Eq. 31)}$$

$$d = \frac{U}{V} * h0 * g0^2 = \frac{25.378}{0.425} * 484.69 * 47.4^2 \qquad \text{(Eq. 32)}$$

$$d = 65024982.73$$

- **Stresses acting on the flanges:**

Assuming the flange thickness to be t=392 mm, calculating factor L,

$$L = \frac{te+1}{T} + \frac{t^3}{d} \qquad \text{(Eq. 33)}$$

$$= \frac{(364.4*0.0018)+1}{1.882} + \frac{364.4^3}{65024982.73}$$

$= 1.632$

Longitudinal Hub Stress:

$$SH = \frac{fM0}{Lg1^2B} \quad \text{(Eq. 34)}$$

$$= \frac{1.042*1099732608}{1.632*65^2*4956.2}$$

$$= 39.945 \text{ MPa}$$

Radial Flange Stress:

$$SR = \frac{(1.333te + 1)M0}{Lt^2B} \quad \text{(Eq. 35)}$$

$$= \frac{(1.333 * 364.4 * 0.0018 + 1) * 1099732608}{1.632 * 364.4^2 * 4956.2}$$

$$= 1.459 \text{ Mpa}$$

Tangential Flange Stress:

$$ST = \left(\frac{YM0}{t^2B}\right) - Z * SR \quad \text{(Eq. 36)}$$

$$= \frac{(23.094*1099732608)}{366.4^2*4956.2} - (11.972 * 1.627)$$

$$= 10.182 \text{ Mpa}$$

- **Check for Allowable Stresses [17]:**

Conditions required to be satisfied:

1) $SH \leq 1.5Sf$ (Eq. 37)

$1.5 * (34.26928) = 51.4 > 39.945$

Hence, the condition is satisfied.

2) $SR \leq Sf$ (Eq. 38)

$1.459 < 34.26928$

Hence, the condition is satisfied.

2) $ST \leq Sf$ (Eq. 39)

$10.182 < 34.26928$

Hence, the condition is satisfied.

4) Check for Combined Stress:

Greater of $\left(\frac{SH+SR}{2}\right), \left(\frac{SH+ST}{2}\right) < Sf$ (Eq. 40)

Greater of $\left(\frac{39.945+1.459}{2}\right), \left(\frac{39.945+10.182}{2}\right) < 34.26928$

$25.062 < 34.26928$