

University of Southern Queensland.  
Faculty of Engineering and Surveying.

**Assessment of Appropriate Pressure Vessel Flange Bolt  
Tension by Finite Element Modelling.**

A dissertation submitted by

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in fulfillment of the requirements of

**Courses ENG4111 and ENG4112 Research Project**

towards the degree of

**Bachelor of Engineering (Mechanical)**

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## **Abstract**

Flanged joints on large diameter flanges can prove problematic to seal successfully with many factors contributing to ensuring a successful operation. One such factor is stud bolt loading contributing to stress and deflection of the flanged joint.

This investigation involves the use of finite element analysis (F.E.A) to predict levels of stress and deflection of a particular flanged joint when the stud bolts are tightened and flange pressurised. The level of stud bolt force selected must ensure the joint is sufficiently tight to avoid leakage. However, the force must not be excessive causing damage.

The flanged joint is located on the channel head of a shell and tube heat exchanger.

For the purposes of this project, the educational version of ANSYS 5.5 was used thus a number of critical assumptions were made to operate within the restrictions of the software.

As a comparative check of the F.E.A method, a conventional method termed the target load bolt-up method was employed.

The analysis results using both methods, when interpreted, indicated the flange was not excessively stressed. Field monitoring by observation of the flanged joint for signs of leakage and other detrimental effects indicates the stud bolt load selected is acceptable.

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I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

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Brett Taylor.

*University of Southern Queensland.*

*October 2004.*

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## Nomenclature

$A$  = outside diameter of flange, in millimetres.

$A_b$  = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, in square millimetres.

$A_m$  = total required cross-sectional area of bolts, taken as the greater of  $A_{m1}$  and  $A_{m2}$ , in square millimetres.

$A_{m1}$  = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions, in square millimetres.

$$= \frac{W_{m1}}{S_b}$$

$A_{m2}$  = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, in square millimetres.

$$= \frac{W_{m2}}{S_a}$$

$A_R$  = area of ring used to calculate equivalent pressure  $P_e$ , in square millimetres.

$B$  = inside diameter of flange, in millimetres.

$B_1 = B + g_0$  for integral-type flanges when  $f$  is equal to or greater than 1.

$b =$  effective gasket or joint-contact-surface seating width, in millimetres.  
 $= 2.52\sqrt{b_o}$

$2b =$  effective gasket or joint-contact-surface pressure width, in millimetres.

$b_o =$  basic gasket seating width, in millimetres (from AS1210 Table 3.21.6.4(B)).  
 $= \frac{N}{2}$

$C =$  bolt circle diameter, in millimetres.

$D =$  diameter of bolt hole, in millimetres.

$D_b =$  bolt outside diameter, in millimetres.

$d =$  factor, in millimetres to the 3rd power, for integral-type flanges  
 $= \frac{U}{V} h_o g_o^2$

$E =$  modulus of elasticity of flange material at operating temperature in megapascals.

$e =$  factor, in millimetres to the power of minus 1 for integral flanges.  
 $= \frac{F}{h_o}$

$F =$  factor for integral-type flanges (from AS1210 Figure 3.21.6.6(B)).

$F_e =$  total equivalent force on flange, in millimeters.

$F_{rot} =$  calculated flange rotation, in degrees.

$f =$  hub stress-correction factor for integral flanges from AS1210 Figure 3.21.6.6(F)  
(when greater than 1, this is the ratio of the stress in the small end of hub to the stress  
in the large end), (for values below limit of figure use  $f = 1$ ).

$G =$  diameter at location of gasket-force, in millimetres; except as noted in AS1210 Figure  
3.21.6.2(a) it is defined as follows:  
when  $b_o > 6$  mm,  $G =$  outside diameter of gasket contact-face minus  $2b$ .

$g_0 =$  thickness of hub at small end, in millimetres.

$g_1 =$  thickness of hub at back of flange, in millimetres.

$H =$  total hydrostatic end-force, in newtons.  
 $= 0.785G^2P$

$H_D =$  hydrostatic end-force on area inside of flange, in newtons.  
 $= 0.785B^2P$

$H_G$  = for flanges covered by AS1210 Clause (3.21.6), gasket-force (difference between flange design bolt-force and total hydrostatic end-force), in newtons.

$$= W - H$$

$H_p$  = total joint-contact surface compression force, in newtons.

$$= 2b\pi GmP$$

$H_T$  = difference between total hydrostatic end-force and the hydrostatic end-force on area inside of flange, in newtons.

$$= H - H_D$$

$h$  = hub length, in millimetres.

$h_D$  = radial distance from the bolt circle to the circle on which  $H_D$  acts, as described in AS1210 Table 3.21.6.5, in millimetres.

$$= \frac{C - D - g_1}{2}$$

$h_G$  = radial distance from gasket-force reaction to the bolt circle as described in AS1210 Table 3.21.6.5, in millimetres.

$$= \frac{C - G}{2}$$

$h_o$  = a factor.

$$= \sqrt{Bg_o}$$



$h_T$  = radial distance from the bolt circle to the circle on which  $H_T$  acts as described in AS1210 Table 3.21.6.5, in millimetres.

$$= \frac{C-B}{4} + \frac{h_G}{2}$$

$J$  = flange rigidity index.

$K$  = ratio of outside diameter of flange to inside diameter of flange.

$$= \frac{A}{B}$$

$L$  = a factor.

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

$M_D$  = component of moment due to  $H_D$ , in newton millimetres.

$$= H_D h_D$$

$M_G$  = component of moment due to  $H_G$ , in newton millimetres.

$$= H_G h_G$$

$M_o$  = total moment acting upon the flange, for operating conditions or gasket seating as may apply, in newton millimetres (see AS1210 Clause 3.21.6.5).

$$= Wh_G$$

$M_T =$  component of moment due to  $H_T$ , in newton millimetres.  
 $= H_T h_T$

$m =$  gasket factor, obtained from AS1210 Table 3.21.6.4(A) (see Note, AS1210 Clause 3.21.6.4.1(a)).

$N =$  width used to determine the basic gasket seating-width  $b_o$ , based upon the possible contact width of the gasket (see AS1210 Table 3.21.6.4(B)), in millimetres.

$n =$  number of bolts.

$P_e =$  equivalent pressure on flange, in megapascals.

$S_a =$  design strength for bolt at atmospheric temperature (given in AS1210 Table 3.21.5 as  $f$ ), in megapascals.

$S_b =$  design strength for bolt at design temperature (given in AS1210 Table 3.21.5 as  $f$ ), in megapascals.

$S_f =$  design strength for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (given in AS1210 Clause 3.3.1 as  $f$ ), in megapascals.

$S_H =$  calculated longitudinal stress in hub, in megapascals.  
 $= \frac{fM_o}{Lg_1^2 B}$

$S_R$  = calculated radial stress in flange, in megapascals.

$$= \frac{(1.33te + 1)M_o}{Lt^2B}$$

$S_{stud}$  = calculated stud bolt stress, in megapascals.

$S_T$  = calculated tangential stress in flange, in megapascals.

$$= \frac{YM_o}{t^2B} - ZS_R$$

$S_Y$  = material yield stress of flange, in megapascals.

$S_{Y_{stud}}$  = yield stress of stud bolt, in megapascals

$T$  = factor involving  $K$  (from AS1210 Figure 3.21.6.6(A)).

$t$  = flange thickness, in millimetres.

$t_x$  = two times the thickness  $g_o$ , when the design is calculated as an integral flange, but not less than 6 mm, in millimetres.

$U$  = factor involving  $K$  (from AS1210 Figure 3.21.6.6(A)).

$V$  = factor for integral-type flanges (from AS1210 Figure 3.21.6.6(C)).

$W$  = flange design bolt-force, for the operating conditions or gasket seating, as may apply, in newtons (see AS1210 Clause 3.21.6.4.4).

$W_F =$  imparted load on flange, in kilonewtons.

$W_{m1} =$  minimum required bolt-force for operating conditions (see AS1210 Clause 3.21.6.4),  
in newtons.

$W_{m2} =$  minimum required bolt-force for gasket seating (see AS1210 Clause 3.21.6.4), in  
newtons.

$w =$  width used to determine the basic gasket seating width  $b_o$ , based upon the contact  
width between the flange facing and the gasket (see AS1210 Table 3.21.6.4(B)), in  
millimetres.

$Y =$  factor involving  $K$  (from AS1210 Figure 3.21.6.6(A)).

$y =$  gasket or joint-contact-surface seating stress (see Note in AS1210 Clause 3.21.6.4.1),  
in megapascals.

$Z =$  factors involving  $K$  (from AS1210 Figure 3.21.6.6(A)).

$\Delta z =$  flange deflection, in millimetres.

## **Glossary.**

Automatic Mesh Generation:	Whereby the computer program generates the input geometry for a Finite Element Analysis.
Axisymmetric:	A model that is symmetric about a central axis.
Boundary Conditions:	Loading and restraint conditions imposed at nodes in a model to mimic the conditions of a real system.
Degrees of Freedom:	The total number of displacement values needed to describe the deformation at a point, element or structure.
Element:	The smallest discrete component that a structure is divided into for a finite element analysis.
Element type:	A group of finite elements that have related derivations and geometry.

Hoop Stress:	Tensile stress in the direction of the tangent to the circumference. Can also be referred to as the tangential or circumferential stress.
Material Properties:	Typical properties that define the behaviour of a material such as Young's modulus and yield strength.
Membrane Stress:	Stresses developed in an axisymmetric vessel section where wall thickness is relatively thin, that is less than 10% of radius. Stress value is considered uniform across the wall thickness.
Mesh:	The grid or array of nodes or elements that make up a finite element analysis.
Node:	A point in 2D or 3D space used to describe the position of one point on an element.
Yield Stress:	The maximum stress that can be applied without permanent deformation. This is the value of the stress at the elastic limit for materials for which there is an elastic limit.
von Mises Stresses	States that failure occurs when the energy of distortion reaches the same energy for failure in tension.

# **Chapter 1**

## **Introduction**

### **1.1 Project Aims**

This research project aims to establish an appropriate bolt tension specific to a particular flanged joint on a shell and tube heat exchanger in order to successfully seal the flanged joint. The stud bolt tension specified needs to take into consideration critical factors such as the ability of the flange not to distort or deflect excessively.

The heat exchanger in question is a registered unfired pressure vessel, therefore Australian Standard AS1210 – Pressure Vessels is the applicable standard governing the design of the flanged joint and associated bolting.

## 1.2 Specific Objectives

Specific objectives relating to this project include:

- Research background information relating to the assembly stresses produced when bolting two flanges together.
- Construction of a model specific to the flange using finite element analysis techniques.
- Analysis of output from finite element analysis model.
- Comparison of output gained from model with a traditional calculation technique.
- Recommendation of required bolt tension to effectively seal flanged joint.
- Monitor via field observation, if flanged joint is successfully sealed.
- Comparison of allowable bolt tensions with those relevant to AS1210: Pressure Vessels.
- Monitoring and recording bolt tensions, establishing if tension has reduced during the time the flanged joint has been in service.

## 1.3 Layout of Dissertation

Chapter 2 Background presents history specific to the flanged joint including a general description of the heat exchanger process duty and general history of the flanged joint. Specific details of the flanged joint components are also presented and discussed whilst the method employed to tension the studs for the flanged joint, hydraulic tensioning is described.



Chapter 3 Literature Review is devoted to review of relevant literature. Presented in detail is background to the problem where both AS1210 and ASME VIII are not prescriptive in the maximum amount of tension that can be applied to a flanged joint for assembly purposes. Various methods of calculation specific to a flanged joint are discussed such as the Taylor-Forge design method, the Target Load Bolt-up method and a recently developed method, the European Method EN1591. The literature review also researches finite element methods specific to a flanged joint.

Chapter 4 F.E.A. Method of Design and Analysis discusses the type of software used, ANSYS 5.5 student edition, the specific finite element analysis method used to model the flanged joint, the axisymmetric approach, and basic assumptions made during the construction the model. Also discussed are material properties, dimensional inputs, elements selected, boundary conditions used and loadings and pressures imposed on the model.

Chapter 5 Target Load Bolt-up Method, Analysis and Results presents a more traditional calculation method, the target load bolt-up method. AS1210 Section 3.21 is used as a basis to calculate the minimum bolt force after which additional allowances are made for assembly purposes of the flanged joint. The calculation method for flange moments, longitudinal hub stress, tangential stress and radial stress are presented. Results of the calculations are presented in a table format.

Chapter 6 Results of F.E.A. Analysis presents results of the finite element analysis. General remarks on the findings on the analysis are discussed. Nodal stress results are presented for the flange assembly, the gasket, blind flange and the flange. Deflection is also presented for the flange and blind flange.

Chapter 7 Conclusions and Recommendations presents discussion whereby conclusions are drawn about stress and deflection results obtained from both the F.E.A and target load bolt-up method. Field trial observations are also described and recommendations are proposed for future observations.

## Chapter 2

### Background

#### 2.1 General Description

The spent liquor heaters were commissioned in the early 1970's and are used to heat 'spent liquor' on the tube-side of the heater whilst the shell-side contains process steam obtained from a flash tank as illustrated by Figure 2.1 below.

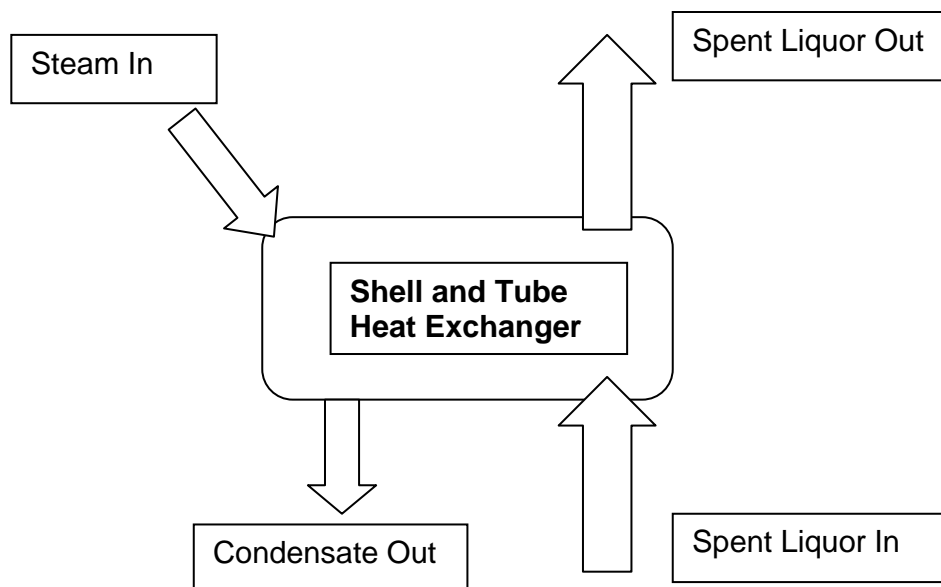


Figure 2.1 Heater Process Flow Diagram

Spent liquor is more commonly known as Sodium Hydroxide, NaOH and in accordance with AS4343 Pressure Equipment – Hazard Levels, is rated as VHL i.e. very harmful liquid.

During the operational life of the Spent Liquor Heaters, flanged joints have always been an area of concern due to leakage. In the early stages of operation it was not uncommon for the flanges to leak considerably, resulting in spray guards being fitted to each flanged joint affording some protection to personnel working in the vicinity.

Leaking joints have also, on occasions, caused failure of a number of bolts due to caustic embrittlement. With the flanged joint weeping at some point, sodium hydroxide dribbles out and downward (gravity) whilst following a path adjacent to the gasket (surface tension). The bolts towards the bottom of the flange tend to get coated in a crusty like material containing sodium hydroxide.

Due to the scaling nature of the process, heaters also require acid cleaning with a 13% sulphuric acid solution on a 6 to 8 day cycle.

Refinements over the years in equipment such as gasket configurations and work practices such as attention to bolt tensioning sequence have brought about considerable improvements resulting in flanged joint discharges being reduced from a spray to a weep. The current situation is still unacceptable by today's standards due to requirements of health, safety, environmental and plant efficiency.

Previous work practices in assembly of the flanged joint involved tensioning the flange bolts by means of long handle spanners and air driven spanners or 'rattle guns'. This

resulted in bolt tension variability that contributes to a less than optimal joint tension. In order to further improve the flange joint integrity, it was decided to use a hydraulic tensioning tool to enable repeatable and consistent tensions to be applied to the stud bolts. The hydraulic unit also provides the option of greater stud bolt tensions. This will provide a tighter potentially leak free joint, however, there are risks in overstressing the flange, stud bolt and gasket crushing.

Correct assembly of the joint requires the flange to be analysed and the correct bolt load established to seal the joint.

## 2.2 Heat Exchanger Details

The location of the flanged joint to be analysed is on the main body of the shell and tube heat exchanger as illustrated by Figure 2.2 below.

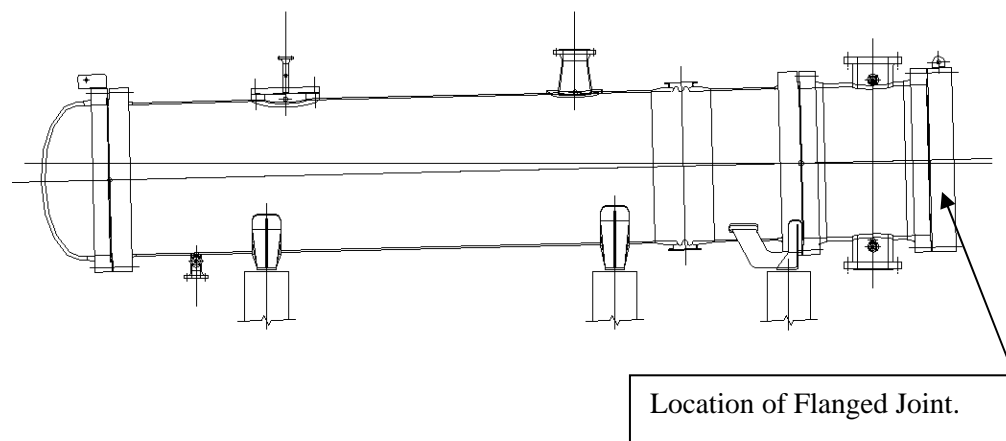


Figure 2.2 General Arrangement of Heater.

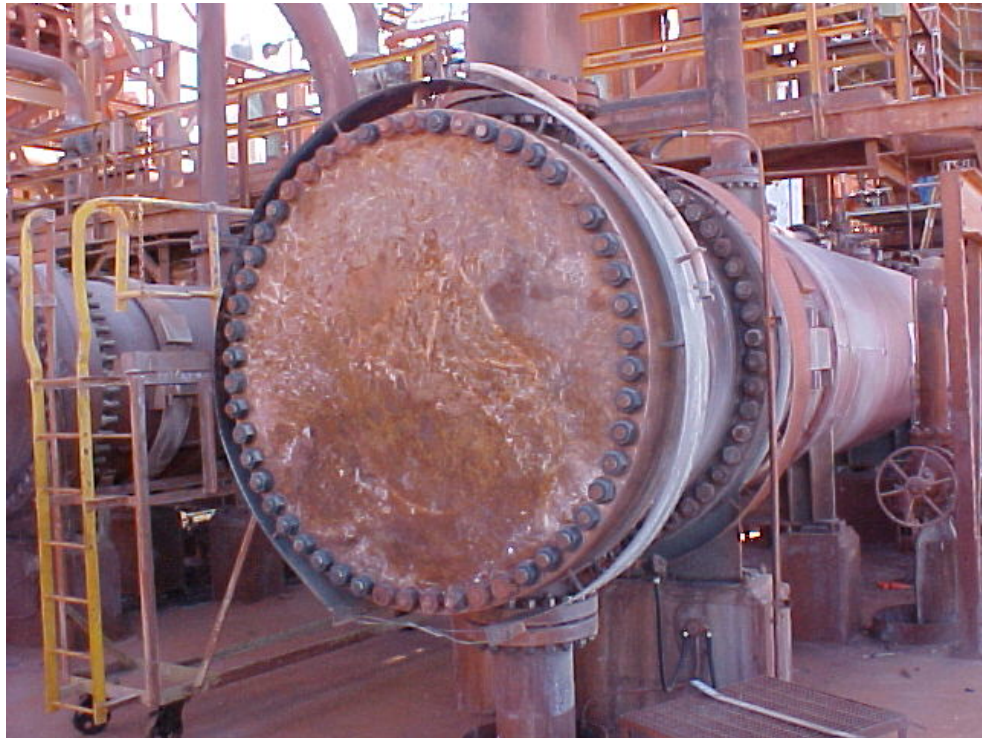


Figure 2.3 Heat Exchanger Flange Assembly [View on Blind Flange]



Figure 2.4 Heat Exchanger Flange Assembly [View on Flange]

### 2.2.1 Design Data for Vessel.

The following table lists relevant design parameters of the heat exchanger.

Table 2.1 Vessel Design Data.

Design Code	ASME VIII	
Hazard Level	B	
Contents Type	Very Harmful	
	Tube-side	Shell-side
Design Pressure	4434 kPa	3206 kPa
Design Temperature	216 °C	247 °C
Contents	Sodium Hydroxide	Steam
Volume	6760 litres	10420 litres

## 2.3 Flanged Joint Data

Figure 2.5 below illustrates the configuration of the flange under consideration.

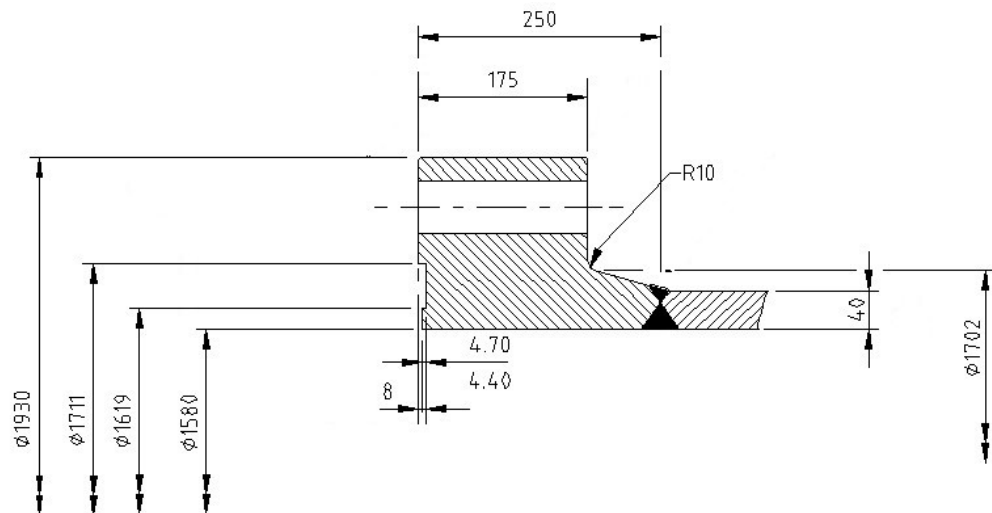


Figure 2.5 Detail of Flange Section.

The flange is shown in sectional view and is basically a hubbed flange, the hub being the section directly behind the flange ring which is 175 mm wide and the welded joint where the flange is attached to the shell.

The left hand face of the flange has a recess machined into its face. The recess is for locating the gasket.



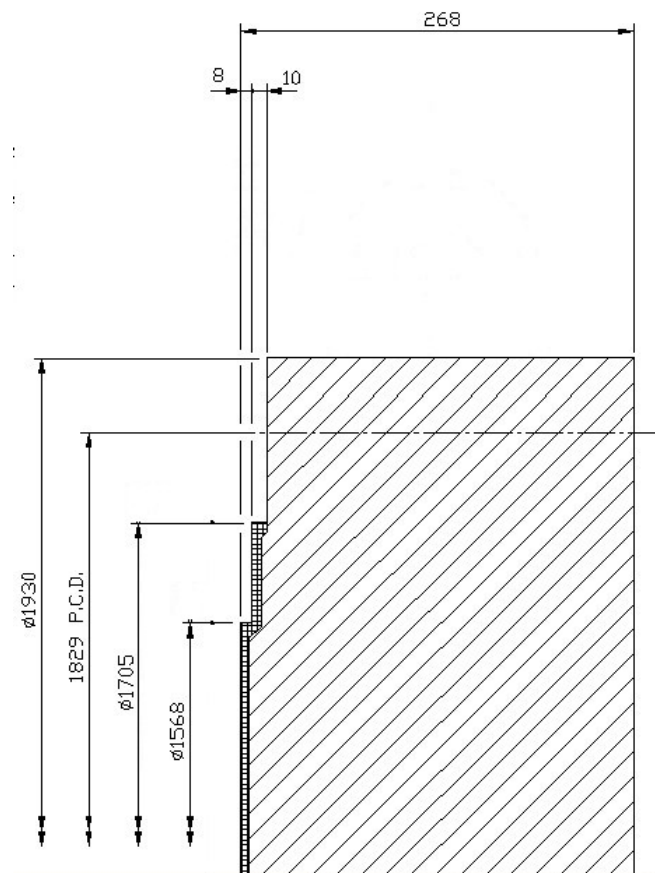


Figure 2.6 Mating Blind Flange

The blind flange is shown in sectional view being 268 mm wide. The hatching on the left hand face is nickel lining used for corrosion resistance.

### 2.3.1 Flanged Joint Parameters

Table 2.2 below lists design parameters of the flanged joint.

Table 2.2 Flanged Joint Data

Channel Head Material	ASTM A516 Grade 70
Blind Flange Material	ASTM A266
Stud Material	ASTM A193 – B7.
Hexagon Nut Material	ASTM A194 – 2H.
Outside / Inside Diameter	1930 mm / 1580 mm.
Pitch Circle Diameter	1829 mm
No. of Studs / Hole Dia.	52 / 54mm
Stud Data	2" x 555mm long, UN8 –2A.
Nut Data	2" Hexagon, UN8

## 2.4 Gasket Data

The gasket used is of spiral wound construction with an outer compression ring. Outside diameter of compression ring is 1705 mm whilst the inside diameter of the ring gasket is 1624 mm. Spirals are constructed using grade 304 stainless steel spiral winding with a soft flexible graphite filler material. The compression ring is constructed using carbon steel to AS1443 / grade CS1010 material.

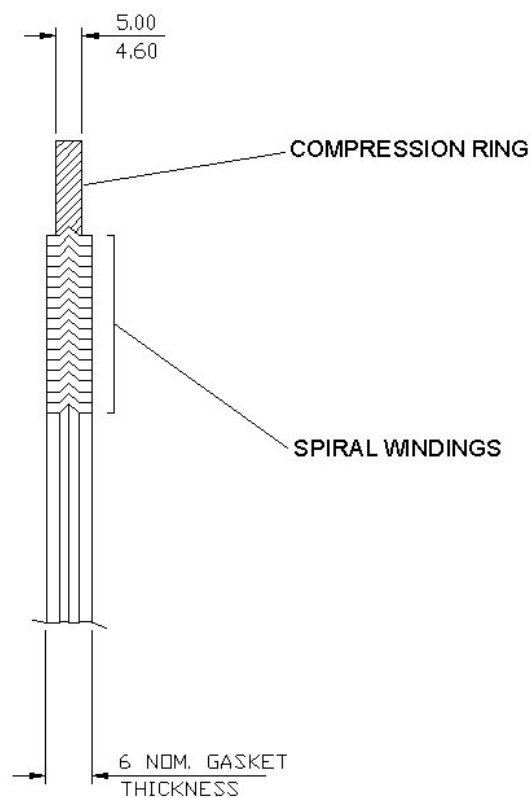


Figure 2.7 Gasket Detail.

## 2.5 Discussion

The subject for this project eventuated out of a situation whereby a change in work practice led to the introduction of a hydraulic tensioning tool to bolt-up flanged joints on a spent liquor heater. The tensioning tool supplier also supplied data suggesting pressures of the hydraulic tool to induce the correct bolt stress.

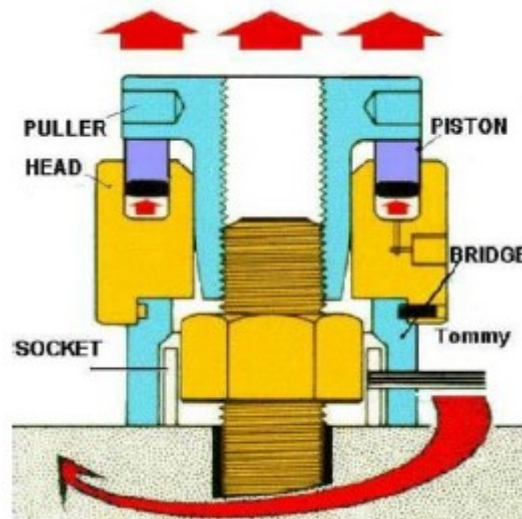


Figure 2.8 Typical Hydraulic Bolt Tensioner

Hydraulic bolt tensioners offer a number of advantages over other method of tightening bolts. These include:

- Accuracy: The method of tightening is independent of the frictional conditions of the bolted assembly, thereby giving accurate and consistent bolt loads.
- Uniformity: Any number of bolt tensioners can be linked together for simultaneous bolt tightening. This is particularly beneficial on flange applications where uniform loading on the gasket is essential in ensuring leak-free connections.

- Time Saving: By tightening many bolts simultaneously the time to bolt up flanges with large numbers of bolts is significantly reduced.
- Compact and Light Weight: Careful design has enabled the development of an effective yet lightweight and compact tool.
- Labour Saving: Bolt tensioners can be used easily by one operator with a minimum of effort.
- Safety: Bolt tensioners are safe in both design and use.
- Simple: Simplicity leads to trouble free, simple and maintenance free operation.

It is understood the bolt tensioners for the heat exchanger flange matches the following parameters:

Table 2.3 Bolt Tensioner Parameters

TITLE	PARAMETER
Stud Diameter	2"
Number of Studs	52
Tensioner Pressure Area	9179 mm <sup>2</sup>
Total Targeted Load/Bolt	500 kN

For safety reasons it was decided to check if the data supplied was correct. As the heat exchanger in question is a registered unfired pressure vessel, Australian Standard AS1210 – Pressure Vessels, Section 3.21 is the applicable standard governing the design of the flanged joint and associated bolting. A calculation was carried out using AS1210 as a basis.

---

The supplied stud tensions appeared to exceed the calculated design values according to AS1210.

Thus it was intended to investigate if stud tensions greater than those prescribed by AS1210 can safely be used and if so the implications of using these higher tensions. i.e. how does the flange react, does it have sufficient strength, is it rigid enough? This shall be achieved by finite element analysis and compared with another method called the Target Load Bolt-up Method.

# Chapter 3

## Literature Review

### 3.1 Background

Consultation with AS1210 revealed no definitive direction on assembly stresses for flange connections, only guidance on flange design limits.

Review of ASME VIII Division 1, Appendix S, Design Considerations for Bolted Flange Connections revealed “it is evident that an initial bolt stress higher than the design value may and in some cases, must be developed in the tightening operation, and it is the intent of this Division that such a practice is permissible, provided it includes necessary and appropriate provision to insure against excessive flange distortion and gross crushing of the gasket.” Appendix S indicates the maximum allowable stress values for bolting are design values and as such are minimum values only. The Appendix does caution against excessive bolt stress resulting in yielding of the bolt, excessive flange deflection and / or gasket crushing however Appendix S does not set upper limits. Gratton & Kempster (2002) concluded there are no guidelines for determination of flange bolting make-up loads in AS2885 and ASME 31.3 or AS1210 whilst Bickford (1995, p.706) informs that,

in his opinion, the ASME Code is intended to be a designers document and not an assemblers document asserting that nowhere does the Code specify or recommend assembly preloads. Bickford (1995, p.705) also comments Appendix S indicates that if one and a half times allowable bolt stress is not enough, and the joint leaks, you should feel free to go to higher levels of stress. He contends the closest Appendix S comes to quantifying an assembly stress in bolts is the amount of stress you might expect to produce is:

$$S_a = 45,000 / \sqrt{D} \quad (3.1)$$

where  $S_a$  = stress created in bolt on assembly [psi.];

$D$  = nominal diameter of the fastener [ in.]

Bowman (2003) states both AS1210 and ASME VIII design rules may not provide sufficient closing force to seal a joint and that engineering judgment may be required to determine what bolt / flange loads are needed whilst Sears and King (2003) suggests a target bolt load is required greater than the minimum bolt loads for operating and gasket seating as prescribed in AS1210.

### **3.2 Taylor Forge Design Method**

As stated above, one of the most common methods used for flange design is found in ASME VIII Division 1, Appendix S, Design Considerations for Bolted Flange Connections. Australian Standard AS1210 also follows this approach. These methods is adapted from of the Taylor-Forge method developed by Waters, Westrom, Rossheim and



Williams of the Taylor-Forge Company in Chicago in the 1930's and subsequently formed the basis of the ASME code for flanged joint design. Singh (1984, p 81-125) explains the Taylor Forge analysis in detail if the reader wishes to follow up. The assumptions made by this method are now generally regarded as simplistic. This method gave rise to the 'm' and 'y' gasket factors in AS1210 and ASME VIII as well as other codes. Some of the principal assumptions and simplifications involved in this method are summarised by Singh as follows:

- Materials of all of the elements are assumed to be homogenous and remain elastic under the loading conditions assumed in the design.
- The effect of the bolt holes in the flanges is neglected.
- Axial symmetry is used to reduce the problem to consideration of the conditions on a single flange, hub and shell cross section, neglecting variations due to location of bolts.
- All loading applied to the flange is reduced to a 'couple' involving a pair of equivalent loads located at the extremities of the flange.
- Stretching of the middle surface of the flange ring due to the applied couple is negligible.
- Displacements of the joint are small such that the theorems of superposition are valid.
- When a ring moment is applied to the flange, the point of connection between the flange and the hub is assumed to have zero radial displacement.
- Hub and shell are assumed to act as thin shells.
- The inside bore of the hub and shell is used in the shell theory analysis instead of the mean thickness diameter.
- Effects due to interaction of elements are neglected.

### **3.3 EN 1591 European Method**

In recent years a European Standard, EN 1591-1 Flanges and their Joints – Design Rules for Gasketed Circular Flanged Connections - Part 1: Calculation Method. This method attempts to address many of the shortcomings of the Taylor-Forge method whilst also giving guidance and setting limits on bolt up loads.

The reader is encouraged to seek further information on this method if desired, as it will not be discussed further during this dissertation.

### **3.4 Target Load Bolt-up Method**

Bickford (1995, p.740 - 746) has described a method to calculate the target bolt load based on the ASME VIII design calculation and taking into account such factors as bolt pre-load scatter, embedment, elastic interaction losses, hydrostatic end load, gasket creep loss for assembly purposes. Sears and King (2003) recommend a similar approach to calculating the target assembly load.

The target load bolt-up method has been employed to calculate the proposed bolt-up load, the output of which is documented in Appendix B in this document. This load will be used as the initial input load into the finite element analysis (F.E.A) model.

### 3.5 Finite Element Analysis Methods

A method to establish and / or review targeted bolt loads of a flanged joint is finite element analysis. Deininger and Strohmeier (1999) used the finite element approach to produce an axisymmetric model of a flanged ring joint and concluded F.E.A. was an acceptable tool for the analysis of flanged joints offering that for convergence of solution a fine mesh and small load steps were required. Welding Research Council Bulletin 341 (1989) also describes using the axisymmetric approach but indicated care should be taken on the non symmetric parts of the joint and non-linear gasket component.

Yasumasa & Satoshi (2000) discuss analyzing a gasketed flange joint using ANSYS F.E.A. software and indicate they have developed a method to model non-linear gasket material using elements available in ANSYS 5.5 when using axisymmetric analysis. They go on to suggest other F.E.A. modelling software such as ABAQUS supports the use of gasket elements.

Raub (2002) discusses a method to accommodate non linear response in gaskets whereby the response of the gasket material must be quantified experimentally.

Reference to AS1210 Appendix B, Finite Element Analysis, insists F.E.A. should only be used alongside conventional analytical techniques and not to use F.E.A. as a primary design tool. In short, F.E.A. should never be done in isolation but in conjunction with other methods. AS1210 Appendix B also gives guidance on calculation methods, result evaluation and reporting of results.

## **Chapter 4**

### **F.E.A. Method of Design and Analysis**

#### **4.1 General Remarks**

The flanged joint analysis was carried out using the student edition of ANSYS, Release 5.5.2. The student edition is limited in capacity to handle up to 1000 nodes only. As such the model was developed to work in with this restriction.

#### **4.2 Basic Assumptions**

In order to simplify the analysis of the flanged joint, a number of assumptions were made.

These basic assumptions are:

- Gasket material was assumed to have linear properties with the non-linear behaviour of the spiral wound gasket section ignored. When the gasket is loaded the spiral windings compress until the flange comes in contact with the outer steel compression ring that is solid steel. As such, when establishing the maximum

stress allowable on the flange joint, the spiral wound gasket, when under this load, is assumed to be acting in a linear fashion.

- All materials for the model, blind, gasket and hub flange are assumed isotropic, i.e. materials have the same elastic properties in all directions, which is a valid approximation for steel.
- Modelling will be via linear static analysis.
- Temperature effects will not be considered.
- Stud loads will be averaged over the area where the studs are located in the circular ring.

### **4.3 Modelling of Joint**

The joint was modeled in a two dimensional area by axisymmetric methods. As the name suggests axisymmetric modelling is symmetrical about an axis. This can best be explained as imagining a cross section of an object. The sectional view is rotated through 360 degrees about an axis. In the case of ANSYS, the symmetrical axis must be the y-axis (vertical axis).

## 4.4 Material Properties

Material properties input into ANSYS were as follows:

Table 4.1 Material Properties.

TITLE	MATERIAL	YOUNG'S MODULUS	PROPERTY DIRECTION
Flange	ASTM A516 Gr 70.	200 GPa	Isotropic
Blind Flange	ASTM A266.	200 GPa	Isotropic
Gasket	Carbon Steel.	200 GPa	Isotropic

## 4.5 Dimensional Inputs.

The flanged joint was modeled by use of the ANSYS GUI input making reference to Figure 4.1 re-presented for continuity, Figure 2.6 Mating Blind Flange and Figure 2.7 Gasket Detail.

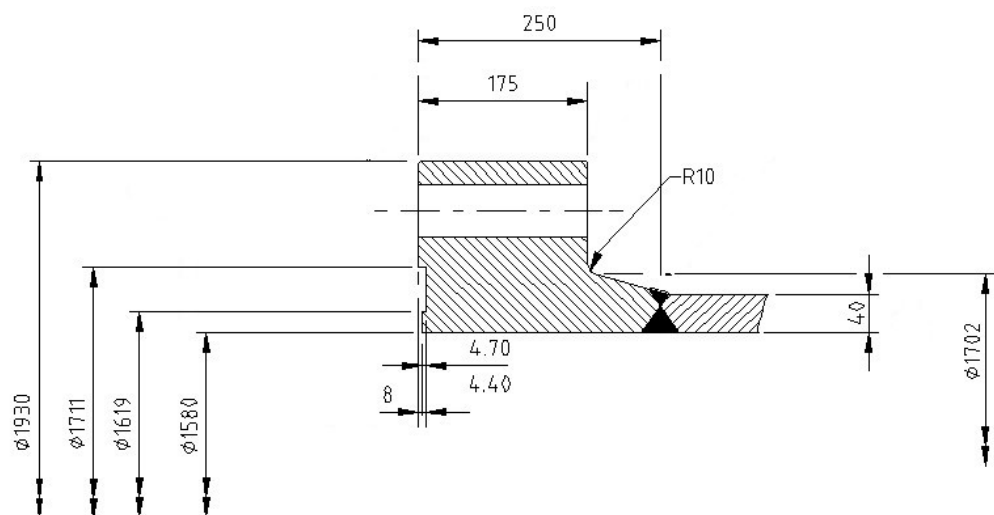


Figure 4.1 Flange Details

Using the dimensions from these details, keypoints were first input as listed in Table 4.2 below.

Table 4.2 Keypoint Input Data.

LIST ALL SELECTED KEYPOINTS. DSYS= 0

NO.	X, Y, Z LOCATION			THXY, THYZ, THZX ANGLES		
1	790.0000	0.000000	0.000000	0.0000	0.0000	0.0000
2	790.0000	695.5000	0.000000	0.0000	0.0000	0.0000
3	809.5000	695.5000	0.000000	0.0000	0.0000	0.0000
4	809.5000	692.0000	0.000000	0.0000	0.0000	0.0000
5	855.5000	692.0000	0.000000	0.0000	0.0000	0.0000
6	855.5000	700.0000	0.000000	0.0000	0.0000	0.0000
7	965.0000	700.0000	0.000000	0.0000	0.0000	0.0000
8	965.0000	525.0000	0.000000	0.0000	0.0000	0.0000
9	941.5000	525.0000	0.000000	0.0000	0.0000	0.0000
10	887.5000	525.0000	0.000000	0.0000	0.0000	0.0000
11	851.0000	525.0000	0.000000	0.0000	0.0000	0.0000
12	830.0000	450.0000	0.000000	0.0000	0.0000	0.0000
13	830.0000	0.000000	0.000000	0.0000	0.0000	0.0000
14	812.0000	692.0000	0.000000	0.0000	0.0000	0.0000
15	812.0000	697.0000	0.000000	0.0000	0.0000	0.0000
16	852.5000	697.0000	0.000000	0.0000	0.0000	0.0000
17	852.5000	692.0000	0.000000	0.0000	0.0000	0.0000
18	0.000000	689.0000	0.000000	0.0000	0.0000	0.0000
19	0.000000	957.0000	0.000000	0.0000	0.0000	0.0000
20	887.5000	957.0000	0.000000	0.0000	0.0000	0.0000
NO.	X, Y, Z LOCATION			THXY, THYZ, THZX ANGLES		
21	941.5000	957.0000	0.000000	0.0000	0.0000	0.0000
22	965.0000	957.0000	0.000000	0.0000	0.0000	0.0000
23	965.0000	707.0000	0.000000	0.0000	0.0000	0.0000
24	852.5000	707.0000	0.000000	0.0000	0.0000	0.0000
25	852.5000	697.0000	0.000000	0.0000	0.0000	0.0000
26	784.0000	697.0000	0.000000	0.0000	0.0000	0.0000

27	784.0000	689.0000	0.000000	0.0000	0.0000	0.0000
28	858.5846	525.0000	0.000000	0.0000	0.0000	0.0000
29	848.9550	517.6963	0.000000	0.0000	0.0000	0.0000
30	830.3704	451.3227	0.000000	0.0000	0.0000	0.0000
31	830.0000	448.6264	0.000000	0.0000	0.0000	0.0000

Once the keypoints were generated, lines were created based on the keypoints. Figure 4.2 illustrates the flanged joint assembly centre line or the y-axis. This is the axis used to revolve the flanged joint about to produce axisymmetry mentioned previously.

In reality, the flange joint has a number of minor fillets and chamfers machined into each respective component. Only major fillets or chamfers were considered necessary to be reproduced in the model such as those at the flange hub region.

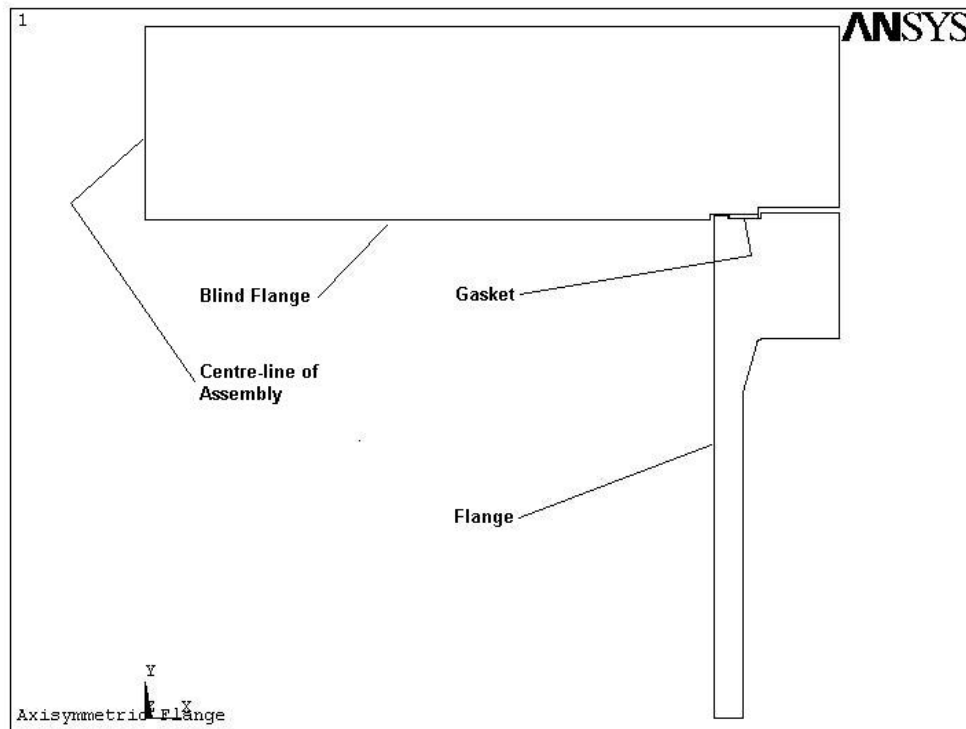


Figure 4.2 Lines generated from Keypoints



The lines were then used to form areas A1, blind flange A2 flange and A3 gasket as illustrated below:

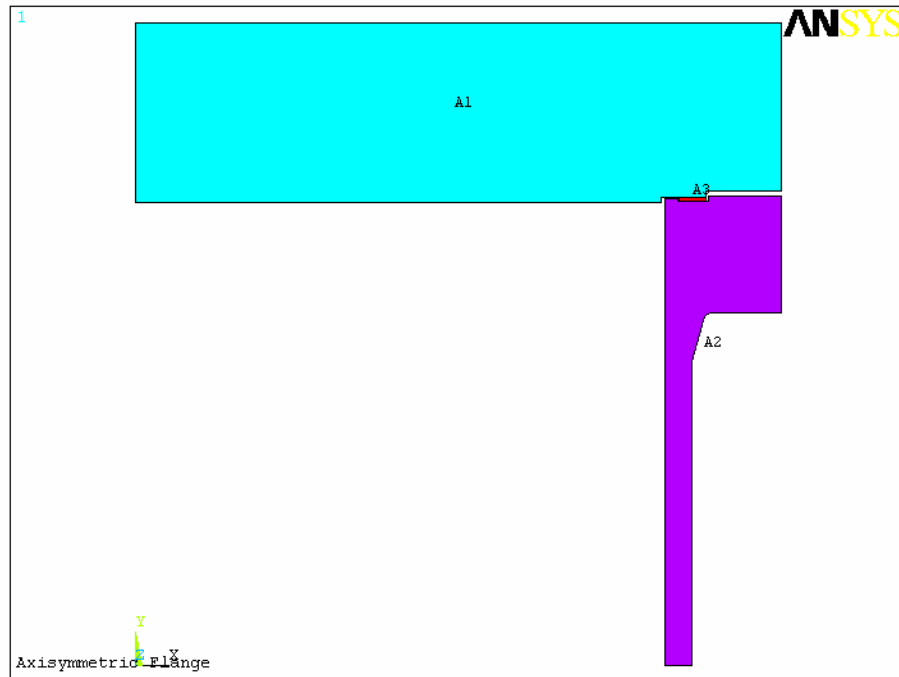


Figure 4.3 Areas generated from Lines

In Figure 4.3 above, A1 represents the blind flange, A2 the flange and A3 the gasket.

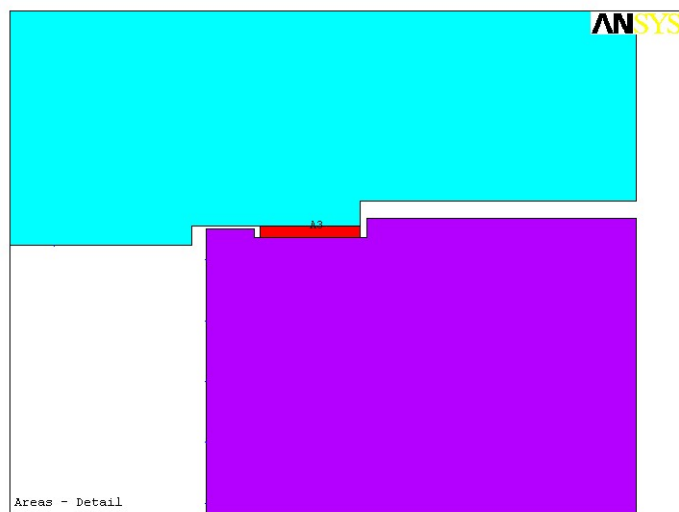


Figure 4.4 Close-up view of Area Assembly

## 4.6 Model Elements.

The element types selected are listed in Table 4.3 below:

Table 4.3 Element Listing

Area Number	Element Type.
A1	PLANE 2
A2	PLANE 2
A3	PLANE 2
A1 / A3 Contact	TARGE169 / CONTA172
A2 / A3 Contact	TARGE169 / CONTA172

According to the ANSYS help files, the PLANE2 element is a six-node triangular element and is suited to model irregular meshes and allows axisymmetric modelling. The PLANE 2 element has 2 degrees of freedom with translation along the x and y axis.

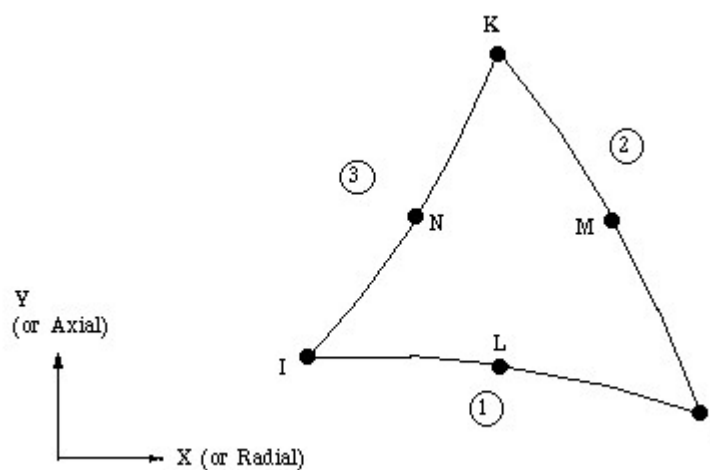


Figure 4.5 Plane2 Element.

TARGE169 is used to represent various 2-D "target" surfaces for the associated contact with CONTA172 elements is used to represent contact between 2-D "target" surfaces. Both of these elements are suitable for use with the PLANE2 element.

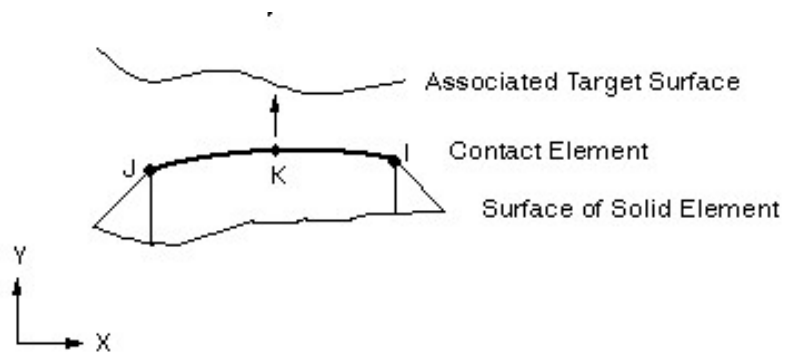


Figure 4.6 Targe169 / Conta172 Elements

## 4.7 Boundary Conditions and Meshing.

Boundary conditions were applied to the axisymmetric model as illustrated in Figure 4.7.

Boundary condition 1 was applied to the vertical centre-line of the blind flange. This constrains the blind in the x-axis direction but leaves the blind free to move along the y-axis. This approach is realistic because of axisymmetry the vertical centre-line axis will not move in the x-axis direction.

Boundary condition 2 was applied to the flange as illustrated. This constrains the flange in the y-axis direction but leaves the flange free to move along the x-axis. Once again this approach is realistic.

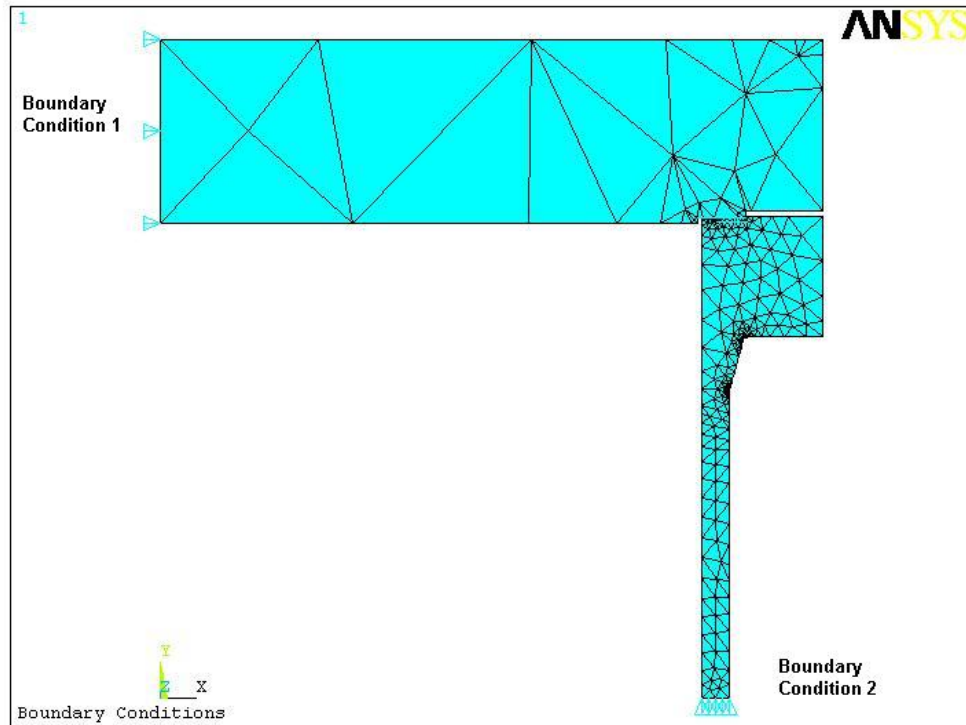


Figure 4.7 Boundary Conditions and Meshing

Meshing was performed with results as illustrated. A coarse mesh was first put in place followed by mesh refinement in regions where highest stresses were thought to exist in the flange. Particular attention had to be paid to mesh density as trouble was encountered numerous times where the number of nodes exceeded the limitations of the ANSYS student edition.

#### 4.8 Loadings / Pressures.

As an axisymmetric approach has been used to model this flange, a method had to be used whereby the load imparted on the flanged joint by the fifty-two studs had to be converted into an equivalent pressure.

For reasons stated in Section 5.1 it was assumed that a 500 kN stud load corresponds to a 440 kN load being transferred to the flanged joint. Therefore the total load transferred to the flanged joint is 22880 kN.

Table 4.4 Loading Parameters

TITLE	PARAMETER	
Stud Size	50.8 (2" UN8 )	mm
Total Number of Studs	54	-
Stud Load Transferred	440	kN
Total Stud Load Transferred	22880	kN
Pitch Circle Diameter	1829	mm

As axisymmetric modelling was used, the total load was represented as an equivalent pressure such that the hole size diameter was chosen as the area of contact where the load was applied. Thus a calculation was performed based on the ringed area with the midpoint being the pitch circle diameter. The area hatched in Figure 4.8 illustrates the region where the equivalent pressure was applied. Of course this pressure was applied over a full 360 degrees of the area on the flange.

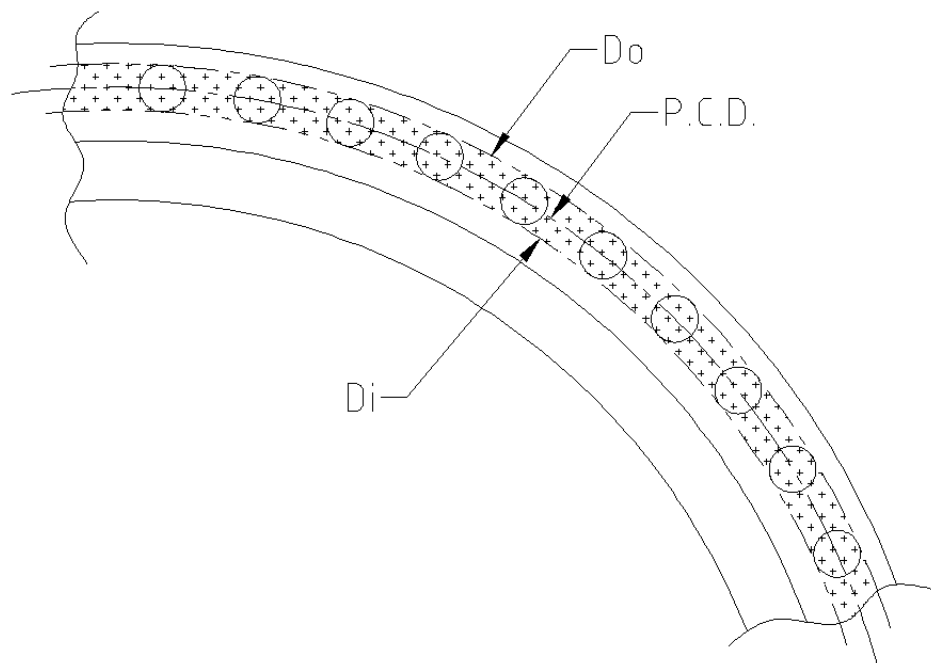


Figure 4.8 Pressure Area

The method of calculation is set out below.

Calculation of diameters:  $D_o$  and  $D_i$ :

$$D_o = PCD + L_c \quad (4.2)$$

$$D_i = PCD - L_c \quad (4.3)$$

where:  $D_o$  is outside diameter [mm];  
 $D_i$  is inside diameter [mm];  
 $L_c$  is line of contact length [mm]; and  
 $PCD$  is pitch circle diameter [mm];

thus:

$$D_o = 1829 + 54$$

$$D_o = 1883 \text{ mm}^2$$

and

$$D_i = 1829 - 54$$

$$D_i = 1775 \text{ mm}^2$$

The area of the ring was calculated as below:

$$A_R = \frac{\pi}{4} (D_o^2 - D_i^2) \quad (4.4)$$

where:  $A_R$  is area of ring [mm<sup>2</sup>];

thus:

$$A_R = \frac{\pi}{4}(1883^2 - 1775^2)$$

$$A_R = 310.28 \times 10^3 \text{ mm}^2$$

Whilst equivalent pressure was then calculated using:

$$P_e = \frac{F_e}{A_R} \tag{4.5}$$

where:  $P_e$  is equivalent pressure [ $\text{N}\cdot\text{mm}^{-2}$ ]; and

$F_e$  is total force applied to flange [N];

thus:

$$P_e = \frac{22880 \times 10^3 \text{ N}}{310.28 \times 10^3 \text{ mm}^2}$$

$$P_e = 73.74 \text{ MPa}$$



The pressure  $P_e$  is then the 'Pressure 2' region as illustrated by Figure 4.9 below.

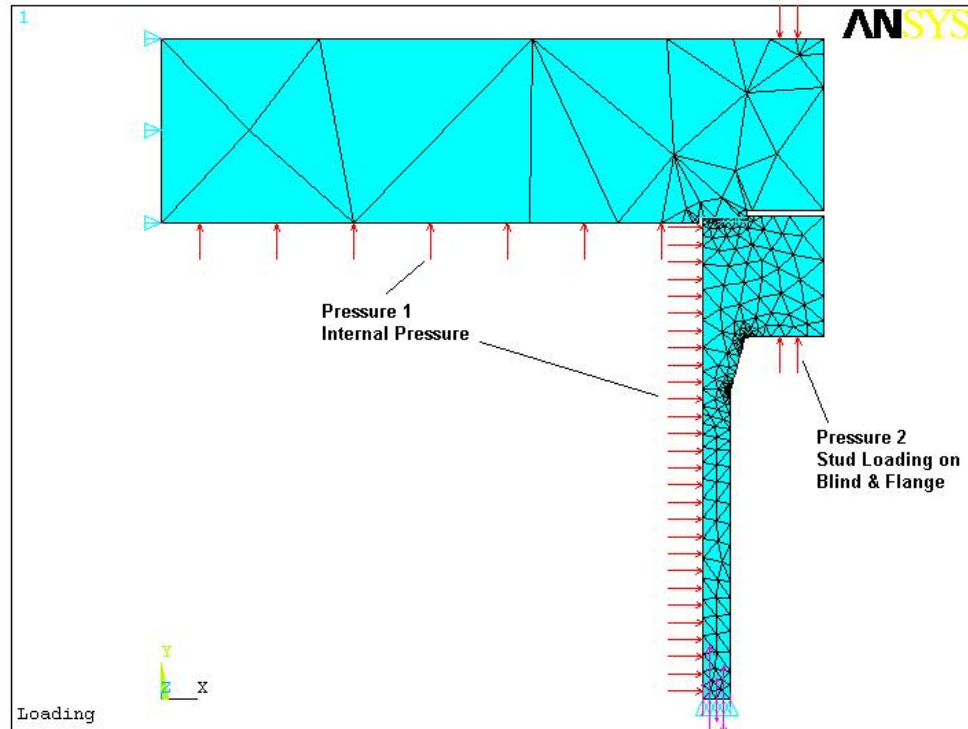


Figure 4.9 Illustration of Pressure Loads

The final pressure applied to the model was that of design internal pressure as illustrated above as 'Pressure 1' region. This internal pressure is simply the pressure that the flanged joint is designed to retain.

With all necessary data input into the model, the solve routine was invoked.

## **Chapter 5**

### **Target Load Bolt-up Method, Analysis and Results**

#### **5.1 General Remarks**

The target load bolt-up method mentioned previously was employed to calculate target bolt-up forces and subsequent flange stresses. The remainder of this chapter presents the calculation method and equations based AS1210 Section 3.21 and additional imposed loads as described in Bickford. Appendix B of this document then provides outputs of such a calculation.

Recalling Section 2.5, Table 2.3 Bolt Tensioner Parameters stated the targeted load per stud, as suggested by the supplier of the bolt tensioner was 500 kN per stud. It is logical that not all the load is transferred to the flange faces. Use of the hydraulic tensioning tool removes most of the variables out of the bolt up process as the stud is simply stretched and nut rotated until the stretch is taken up. However, even though the stud is stretched to an equivalent 500 kN, when the nut is done up, such factors as embedment and thread engagement contribute to reduce the applied load to the flange faces.

Bickford has estimated this reduction to be in the order of 10 to 15%. For the purpose of this investigation, the reduction was estimated to be in the order of 60 kN. Therefore as a basis for all stress calculations involving the flanged joint, but not the stud bolt, a residual load of 440 kN was adopted.

## 5.2 AS1210 Flange Design Bolt Forces

As per AS1210 Section 3.21, the following section details the procedure required to calculate the minimum required bolt force for a flanged joint.

The maximum of the two calculated forces ,  $W_{m1}$  and  $W_{m2}$  is used to set the minimum required bolt force as set out below.

Minimum required gasket seating force  $W_{m2}$  [N] is given by:

$$W_{m2} = \pi b G y \quad (5.6)$$

and the minimum required bolt-force for operating conditions  $W_{m1}$  [N] is given by:

$$W_{m1} = 0.785 G^2 P + 2 b \pi G m P \quad (5.7)$$

where  $b$  is the effective gasket seating width [mm];

$G$  is the diameter at location of gasket force [mm];

$y$  is the gasket seating stress [MPa];

$P$  is the calculation pressure [MPa] and

$m$  is a gasket factor.

Flange design bolt force bolt force,  $W$  [N] is the maximum of  $W_{m1}$  and  $W_{m2}$  above.

### 5.3 Additional Allowances for Bolt-up

Additional loads are now applied to go above the minimum load calculated as per Section 5.2. Bickford (1995, p706-710) describes assembly preload allowances to cope with potential losses in clamping force either during tightening or when the joint is put into service. These allowances are listed in Table 5.1 with allowance values specified:

Table 5.1 Bolt-up Allowance Values.

<b>BOLT-UP ALLOWANCES</b>	<b>%</b>
Preload Scatter	10
Embedment	10
Elastic Interaction Losses	48
Gasket Creep Losses	30

Thus the flange design bolt force  $W$  was increased by applying the above factors. The factored up bolt-up load is given by  $W_F$  [N] with the aim of locating this force between the lower load range and upper load range as illustrated in Figure 5.1.

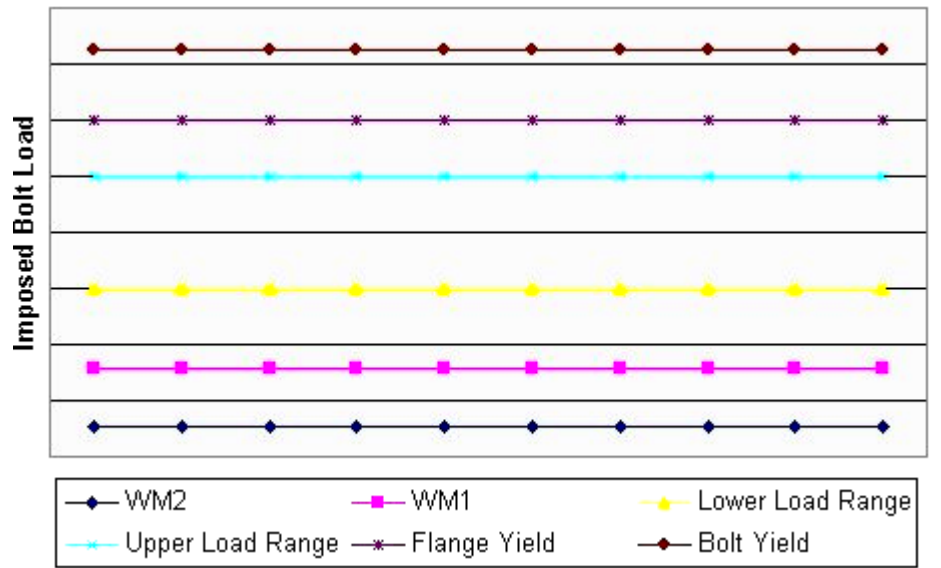


Figure 5.1 Bolt Load Comparative Range

Figure 5.1 demonstrates the relative bolt loads imposed where:

$W_{m1}$  is minimum required bolt force for operating condition,  $W_{m2}$  is minimum required bolt force for gasket seating condition are Code calculated minimum loads whilst upper load range and lower load range indicate target bolt up load range and flange yield and bolt yield indicate loads at which respective yield stresses are reached.

### 5.4 Flange Moments

Total flange moment acting on the flange, for the operating conditions  $M_o$  [N mm] is given by:

$$M_o = M_D + M_T + M_G \tag{5.8}$$

or for gasket seating condition  $M_o$  [N mm] is given by:

$$M_o = W_F h_G \quad (5.9)$$

where  $M_D$  is  $H_D h_D$ , the component of moment due to  $H_D$  [N mm];

$M_T$  is  $H_T h_T$ , the component of moment due to  $H_T$  [N mm];

$M_G$  is  $H_G h_G$ , the component of moment due to  $H_G$  [N mm];

$h_G$  is the radial distance from gasket force reaction to the bolt circle;

$h_T$  is the radial distance from the bolt circle to circle on which  $H_T$  acts and

$h_D$  is the radial distance from the bolt circle to circle on which  $H_D$  acts.

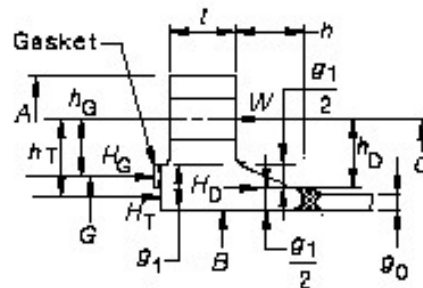


Figure 5.2 Typical Hubbed Flange Diagram

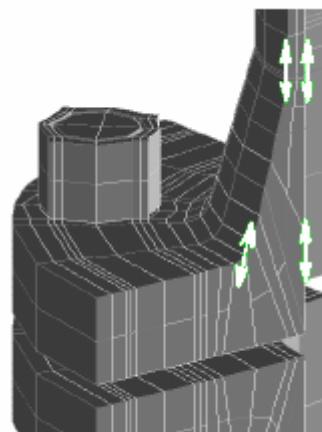
## 5.5 Flange Stresses

Three flange stresses are calculated in the AS1210 method as follows:

- Longitudinal hub stress,
- Radial stress, and
- Tangential stress.

### 5.5.1 Longitudinal Hub Stress

Longitudinal Hub Stress  $S_H$  [MPa] is the bending stress that varies through the hub thickness the location of which is illustrated in Figure 5.3. Singh and Soler (p125) described this stress as essentially a bending stress with the maximum stress being nearly always at either extremity of the hub. Paulin (2003) indicated that the maximum longitudinal hub stress could be up to is 2 times the material yield stress in this region.



[Paulin (2003)]

Figure 5.3 Longitudinal Hub Stress Region

$$S_H = \frac{fM_o}{Lg_1^2 B} \quad (5.10)$$

where  $S_H$  is the longitudinal hub stress [MPa];  
 $f$  is a hub stress-correction factor;  
 $B$  is the inside diameter of flange [mm];  
 $L$  is a factor and  
 $g_1$  is the thickness of the hub at back of flange [mm];

### 5.5.2 Radial and Tangential Stress

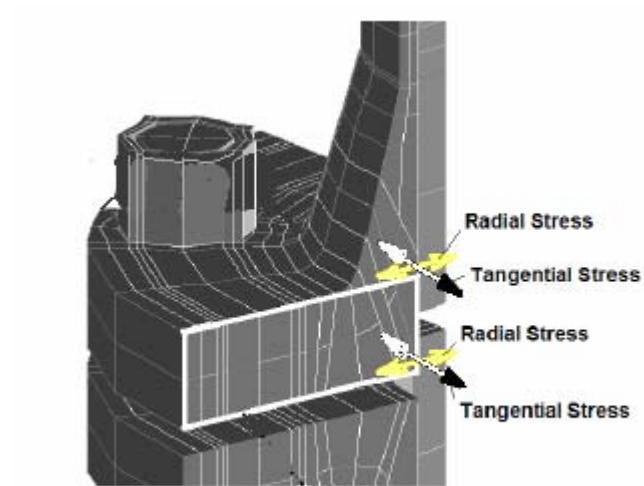
Radial Stress  $S_R$  [MPa] and tangential stress  $S_T$  [MPa] are stresses located in the region as illustrated in Figure 5.4.

Singh and Soler (p 125) describe the radial stress in the flange ring consists of two components, the bending stress caused by the radial bending moment and the membrane stress caused by in-plane surface loads on the inside diameter. Waters et al. demonstrated the maximum stress always occurs at the inside diameter of the ring.

Singh and Soler (p125-126) also indicated the tangential stress in the ring is made up of two parts, the bending stress caused by the circumferential bending moment and the circumferential stress due to membrane stress caused by in-plane surface loads on the inside diameter. Waters et al.



demonstrated the maximum stress always occurs at the inside diameter of the ring. Maximum radial and tangential stresses allowable are 1.0 times the material yield stress.



[Paulin (2003)]

Figure 5.4 Radial & Tangential Stress Regions

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2B} \quad (5.11)$$

and

$$S_T = \frac{YM_o}{t^2B} - ZS_R \quad (5.12)$$

where  $S_R$  is the radial hub stress [MPa];

$S_T$  is the tangential hub stress [MPa];

$t$  is the flange thickness [mm];

$e$  is a factor [ $\text{mm}^{-1}$ ];

$Y$  is a factor involving  $K$ ;

$Z$  is a factor involving  $K$  and;

$K$  is the ratio of outside to inside flange diameter.

## 5.6 Results of Analysis

Inputs and results of the AS1210 / target load bolt-up method are presented in Appendix B of this document. The computed stress values are re-presented in Table 5.2 for continuity of reading.

Table 5.2 Calculated Flange Stresses

<b>INPUTS</b>			
Internal Pressure	$P$	4.34	MPa
Imparted Flange Load	$W_F$	440	kN
Yield Stress	$S_Y$	262	MPa
<b>OUTPUT STRESSES</b>			
Longitudinal Hub Stress	$S_H$	317	MPa
Radial Flange Stress	$S_R$	58	MPa
Tangential Flange Stress	$S_T$	125	MPa
Combined Stresses	$0.5(S_H + S_R)$	188	MPa
Combined Stresses	$0.5(S_H + S_T)$	221	MPa

The limits of stress set by AS1210 Clause 3.21.6.7 are as follows:

- $S_H: 1.5 \times S_f$  ;
- $S_R: 1.0 \times S_f$  ;
- $S_T: 1.0 \times S_f$  ;
- $0.5(S_H + S_R): 1.0 \times S_f$  and;
- $0.5(S_H + S_T): 1.0 \times S_f$  .

Note that these stress multiples are referenced to  $S_f$  , the design strength of the flange material, in this case 135 MPa.. Reviewing Table 5.2 it is evident that in some cases, stresses have been exceeded. However, remember,  $S_f$  is referring to design stress limits and not bolt-up stress limits. Limits for bolt-up can be set at  $S_Y$  the material yield strength.

Therefore bolt-up stress limits are a follows:

- $S_H: 1.5 \times S_Y = 393 \text{ MPa}$ ;
- $S_R: 1.0 \times S_Y = 262 \text{ MPa}$ ;
- $S_T: 1.0 \times S_Y = 262 \text{ MPa}$ ;
- $0.5(S_H + S_R): 1.0 \times S_Y = 262 \text{ MPa}$  and;
- $0.5(S_H + S_T): 1.0 \times S_Y = 262 \text{ MPa}$ .

Explanation regards the bolt-up stress limit for  $S_H$  exceeding the material yield stress was explained by Paulin (2003) in that the stress is a bending stress. Also present in the hub region is a membrane stress component acting opposite to the longitudinal hub

(bending) stress. Hoop direction stresses are also present due to internal pressure as illustrated in Figure 5.5.

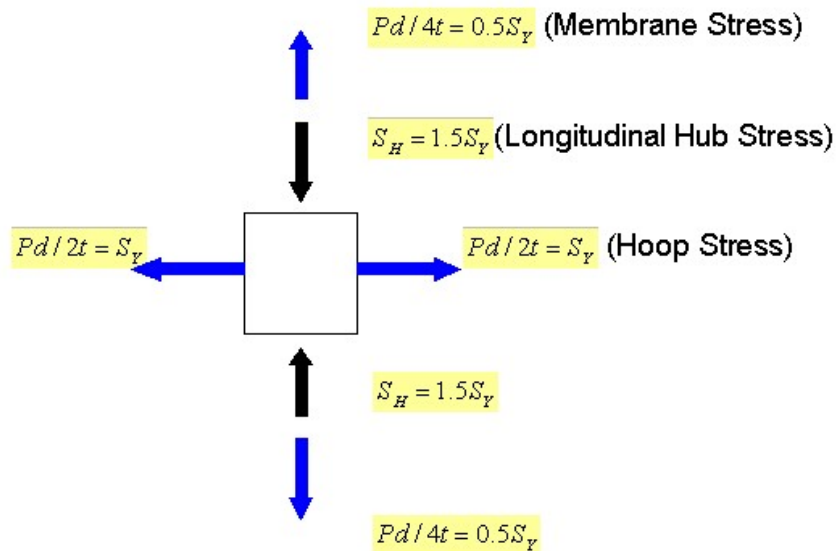


Figure 5.5 Stress Element at Hub Region.

The longitudinal hub stresses are compressive whilst the membrane stresses are tensile thus:

$$(1.5S_y - 0.5S_y) - (-(1.5S_y - 0.5S_y)) = 2S_y \quad (5.13)$$

Therefore the stresses in the hub region could be twice the yield stress in the longitudinal direction. Paulin (2003) concludes that this situation appears safe in that the bending stresses are self relieving and the bending component is non-cyclical.

For the purposes of this investigation a limit of  $1.5S_y$  was placed as the maximum longitudinal hub stress allowable.

Table 5.3 presents calculated flange stresses and compares them to allowable stresses. It can be seen that, according to these calculations, the flange is within allowable stress limits.

Table 5.3 Calculated vs. Allowable Stresses.

STRESS	SYMBOL	CALCULATED	ALLOWABLE	UNITS
Longitudinal Hub Stress	$S_H$	317	393	MPa
Radial Stress	$S_R$	58	262	MPa
Tangential Stress	$S_T$	125	262	MPa
Combined Stresses	$0.5(S_H + S_R)$	188	262	MPa
Combined Stresses	$0.5(S_H + S_T)$	221	262	MPa

## 5.7 Flange Rotation.

As a check on flange rotation or rigidity, the following calculation from ASME VIII Division 1, Appendix S-2 was performed. The flange is deemed sufficiently rigid when the calculated value of the flange rigidity index  $J$  is  $\leq 1$  where  $J$  is given by;

$$J = \frac{52.14M_oV}{K_1LEg_o^2h_o} \quad (5.14)$$

where  $J$  is the index of rigidity;  
 $M_o$  is the total flange moment [N mm];  
 $V$  is a factor relating to an integral flange;  
 $L$  is a factor;  
 $E$  is the modulus of elasticity [kPa];  
 $K_l$  is a factor equal to 0.3 for an integral flange.  
 $g_o$  is the hub thickness at small end and  
 $h_o$  is a factor;

thus,

$$J = \frac{52.14 \times 2074 \times 10^3 \times 0.366}{0.3 \times 1.33 \times 207000 \times 10^3 \times (40 \times 10^{-3})^2 \times 251.4} = 1.23$$

where  $M_o = 2074 \times 10^3$  N.m.  
 $V = 0.366$   
 $L = 1.333$   
 $E = 207000 \times 10^3$  kPa  
 $K_l = 0.3$   
 $g_o = 40 \times 10^{-3}$  m  
 $h_o = 251.4$

Thus  $J = 1.23$ , and exceeds the suggested index value of 1. This indicates the flange may not be rigid enough and thus allow leakage at the joint. It does not however suggest the configuration does not meet the requirements of the Code as Appendix S-2 is classed non-mandatory.

## 5.8 Stud Bolt Stresses

Another important issue to consider is the level of stress imposed on the stud bolts. Table 5.4 illustrates output calculations of stud bolt stresses as per Appendix B in this document. The stud bolt stress is 40% of the yield stress. According to Bickford, 40% to 50% is the recommended limit for stud bolt stress with a limit of 40% being recommended in situations where stress corrosion cracking may be a problem. This is the case in this particular situation where sodium hydroxide is known to promote cracking at high levels of stress.

Table 5.4 Stud Bolt Stresses.

<b>INPUTS</b>			
Imparted load per stud	$W_F$	500	kN
Stud bolt effective area	$A_b$	1729	mm <sup>2</sup>
Yield Stress	$S_{Y\_stud}$	720	MPa
<b>CALCULATED STRESS</b>			
Stud Bolt Stress	$S_{stud}$	289	MPa

## 5.9 Summary of Results – Target Bolt-up Method.

Initially AS1210 Section 3.21.6 was used to calculate the minimum required stud load for gasket seating and operational cases. The maximum of these two values was used as a starting point to apply extra load to the stud to provide a margin above the minimum value.

The target bolt-up method was used to provide guidance as to how much extra load should be applied. Before using the suggested value it must be checked that maximum stress values are not exceeded in the flanged joint.

A stud load of 500kN was suggested however it is thought only approximately 440 kN is actually imposed or transferred to the flanged joint. This is the value used for stress calculations on the flange.

The methodology used to calculate flange stresses was taken from AS1210 Section 3.21.6.6. whilst Appendix B of this document presents the output of such calculation.

The calculated stresses were then compared with allowable stresses as presented in Table 5.3 with results suggesting the flange is not overstressed.

Flange rigidity was then calculated and suggested the flange may be prone to over rotation.



Finally, the stud bolt stress was calculated and found to be 40% of the yield stress which is an acceptable level of stress.

# Chapter 6

## Results of F.E.A. Analysis

### 6.1 General Remarks

It is believed that accuracy of results in areas of the model were limited to a degree by limitation on mesh density. However in an effort to get the best result whilst operating within the constraints of the educational version, increased density was chosen in areas of interest thought to contain highest stresses about the hubbed region of the flange.

The general results output from ANSYS appeared to be consistent in what was expected to eventuate. These general results and observations include:

- Flange ring outside diameter region deflecting generally in the positive y-axis direction,
- Blind flange outside diameter region deflecting generally in the negative y-axis direction,
- Gasket region being the point of zero rotation, i.e. both the flange and the blind flange rotated about the gasket region,

- A portion of the gasket exceeded the material yield stress and
- Flange stresses are significant in the hubbed region of the flange.

## 6.2 Nodal Stress Results – Joint Assembly

Presented in Figure 6.1 are the von Mises stresses for the nodal solution of the assembled joint.

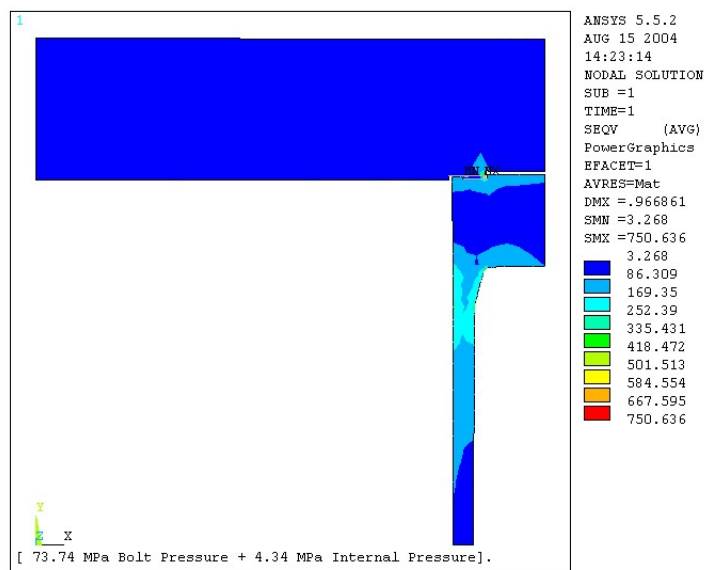


Figure 6.1 Nodal Stress Solution – Assembly

Figure 6.2 gives a little more clarity as to where these stresses are located with maximum stress of 750 MPa occurring at the outside diameter of the gasket where the gasket contacts the flange face with minimum stress occurring at the inside diameter of the gasket / flange interface.

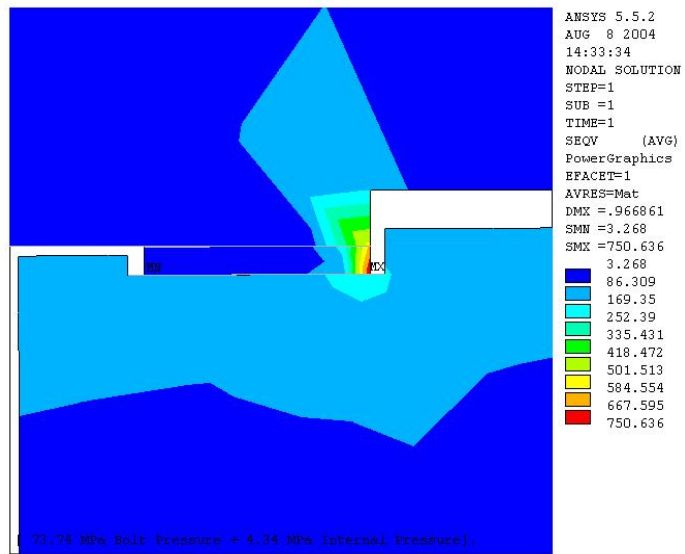


Figure 6.2 Nodal Stress Solution – Assembly Detail

Apart from indicating maximum stresses occur at the outside diameter of the gasket, it also gives an indication to the region that is the 'pivot point' of the assembled flange under load. That is the point of zero rotation in the x-y plane. This will be further discussed in Section 6.5 on deflection.

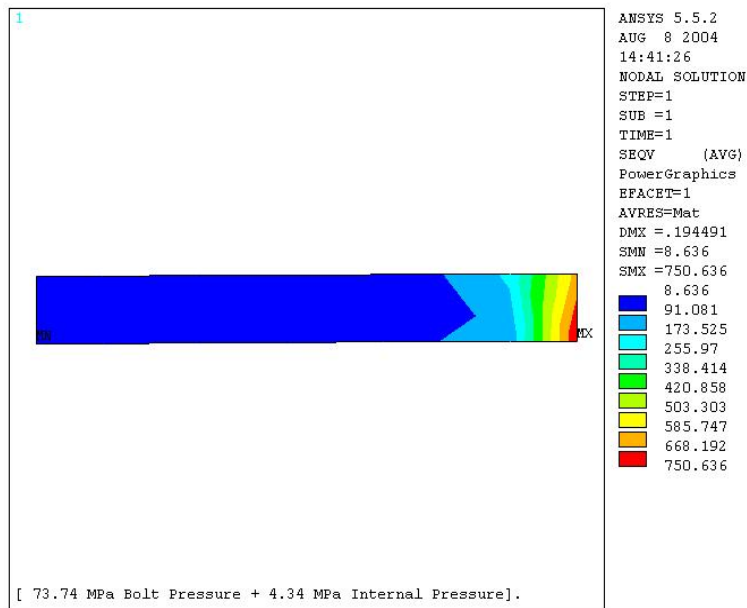


Figure 6.3 Nodal Solution – Gasket.

### 6.3 Nodal Stress Results – Flange

Stress plots for the flange are presented in Figure 6.2, Figure 6.4 and Figure 6.5 with maximum stresses occurring in the outside diameter of the hubbed region. This is as expected with the stud load tending to rotate the flange ring in the positive y-axis direction. This in turn produces bending stresses in the hub region through the section of minimum area.

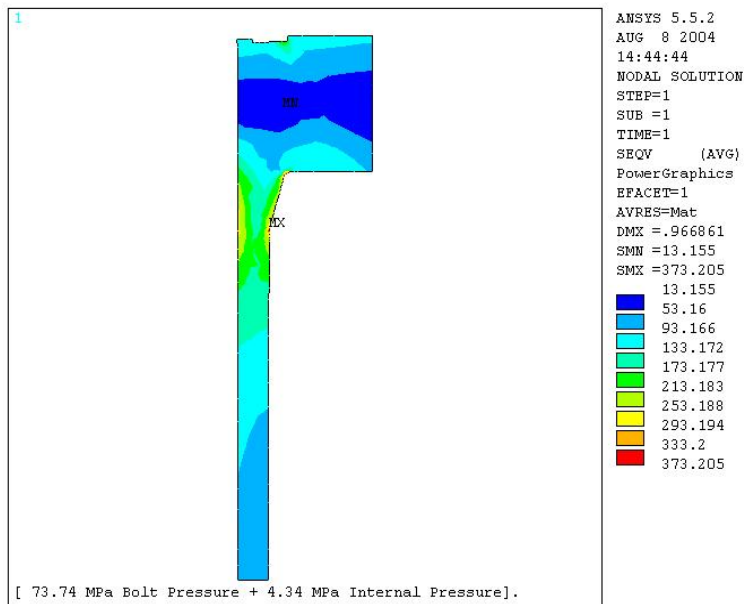


Figure 6.4 Nodal Stress Solution – Flange

A close up view of this region indicates three areas of interest. Point 2, once again is where the maximum stress occurs, whilst point 3 on the opposite side of the flange indicates a region of high stress, although not as high.

Recalling that the yield stress for the flange material is 262 MPa, it is evident that some small regions are overstressed, however this stress state does not exist through the entire cross section of this area. It is envisaged point 1, also an area of high stress occurs mainly due to the sharp change in direction, a stress raiser.

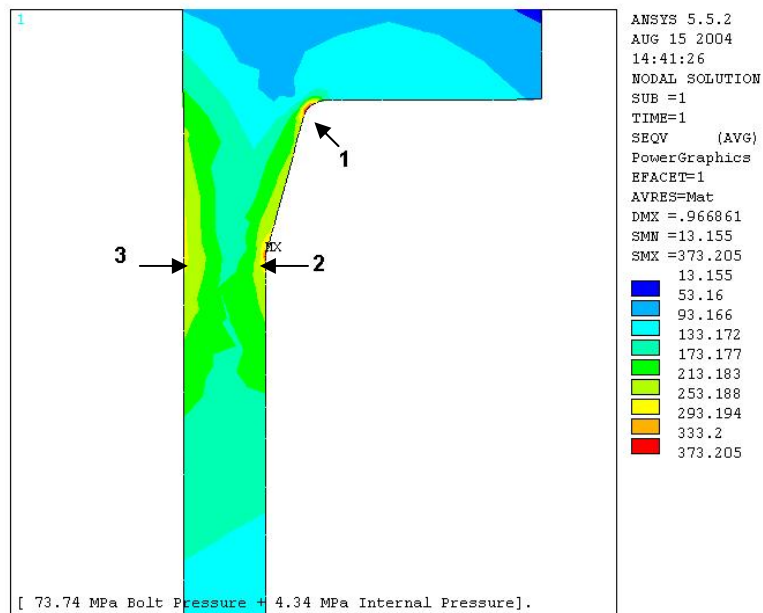


Figure 6.5 Nodal Solution – High Stress Area on Flange.

Figure 6.6 is presented to give a detailed view in the region of maximum stresses at the base of the flange hub. Taking a cross-section through this region it is evident approximately 90% of the cross-section is below the yield stress of the material. In those areas the stress would tend to be redistributed.

Smith & van Laan (p63) reviewed the various failure modes of piping systems and divided types of failure modes into the following categories:

- Primary stress; plastic deformation.
- Secondary stress; plastic instability leading to incremental collapse and
- Peak stress; fatigue failure resulting from cyclic loading.

They contend primary stresses are developed when mechanical loads are applied and are not self-limiting. Thus if the yield stress is exceeded through the entire cross section failure will occur. Local primary stresses that exceed yield will redistribute themselves as the local distortion occurs.

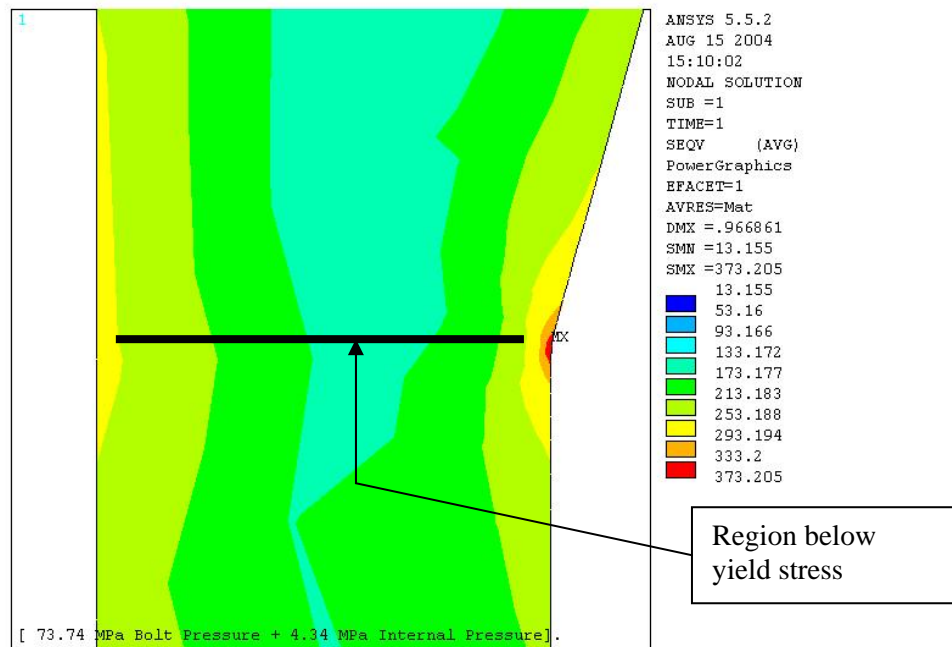


Figure 6.6 Nodal Solution – High Stress Area on Flange (Zoom)

## 6.4 Stress Results – Blind Flange

Results for the blind flange are somewhat limited due to the coarse meshing employed in this area.

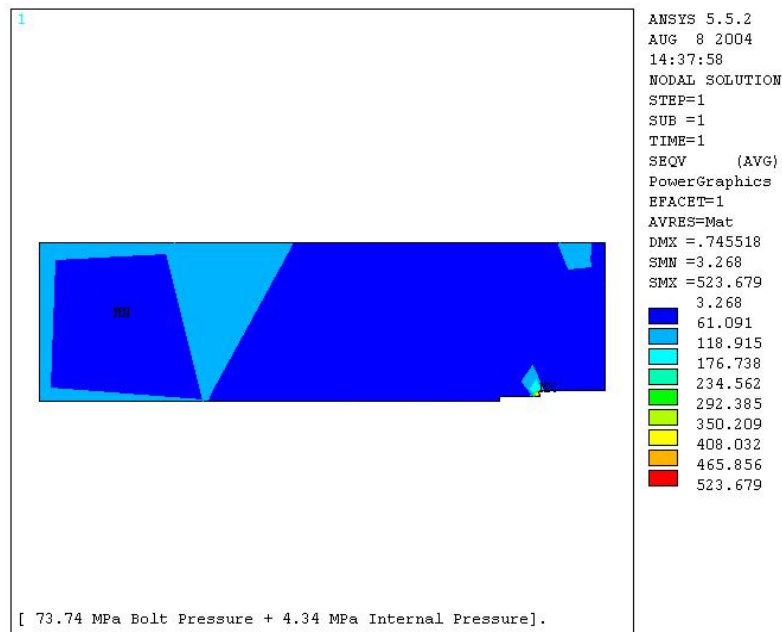


Figure 6.7 Nodal Solution – Blind Flange

The maximum stress reported was 523 MPa and occurred at a discontinuity where there is a section change in thickness. This was not considered an issue due to the localised nature of the region in which the yield stress was exceeded.

Generally, apart from the region discussed above, the blind flange exhibited stresses well below that of yield.



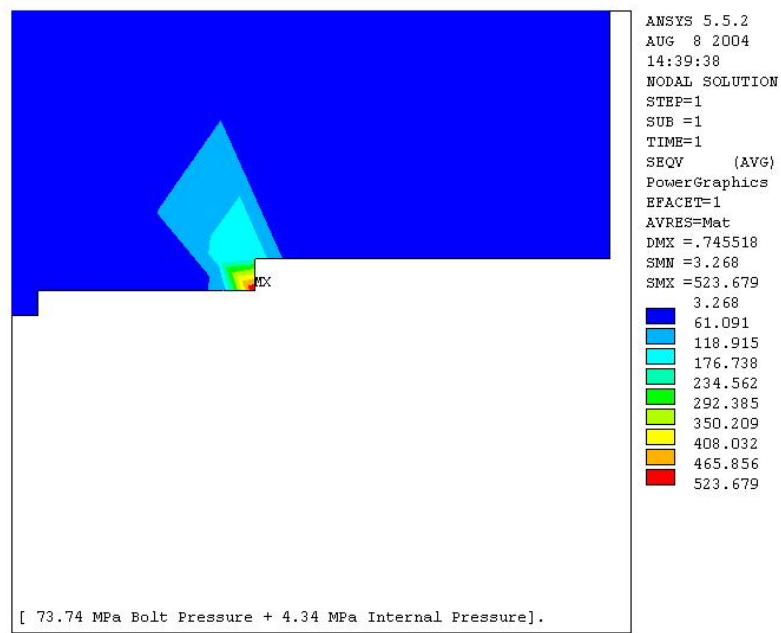


Figure 6.8 Nodal Solution – Blind Flange (Zoom)

## 6.5 Deflection Results.

Figure 6.9 presents general deflection results. Maximum deflection of the flanged joint is 0.96 mm and is located at the outside diameter of the flange.

The deflection of the blind flange is understandable with a combination of internal pressure and to a lesser extent the stud bolt force causing the centre of the blind flange to move in the positive y-axis direction as indicated by the left hand arrow in Figure 6.9. At the same time the outside diameter of the blind flange moves in the negative y-axis direction.

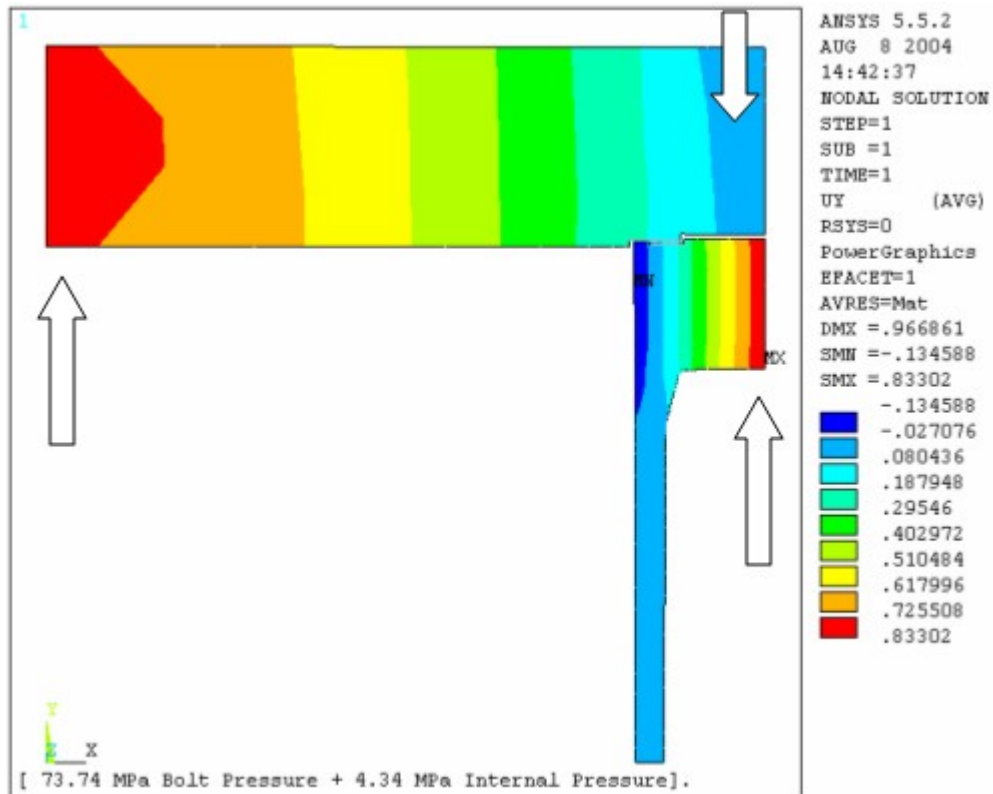


Figure 6.9 Deflection – Assembly

In regard to specific results for the flange, Figure 6.10 demonstrates that the outside diameter region moves in the positive y-axis direction, 0.83 mm whilst the inside diameter region moves in the negative y-axis direction by 0.13 mm. Once again, this outcome appears reasonable suggesting the flange is actually rotating about some point in the gasket contact region.

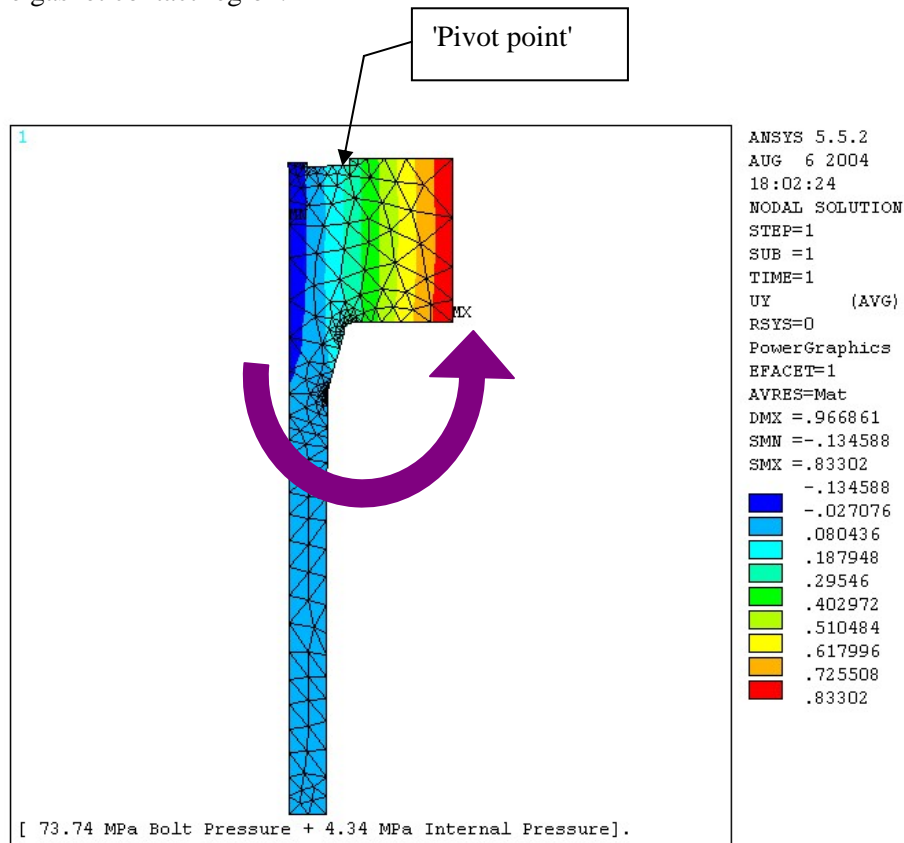


Figure 6.10 Deflection – Flange.

## 6.6 Flange Rotation.

ASME VIII Division 1 sets a non-mandatory value of ring rotation or flange rotation for an integral hub flange as 0.3 degrees.

The reported deflection from the F.E.A. analysis was used to calculate the ring rotation and comparing this value with the 0.3 degree limit.

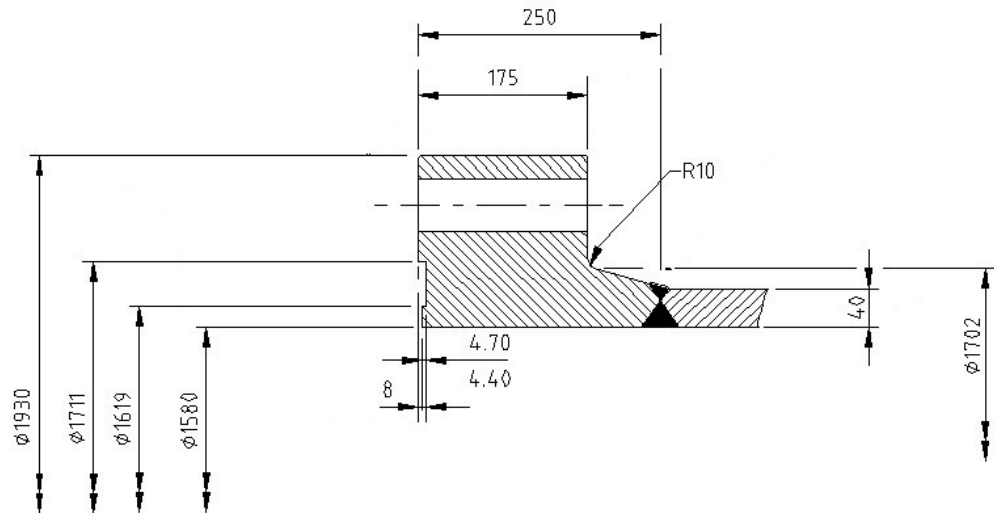


Figure 6.11 Flange Dimensions

This rotation calculation uses the outside diameter (1930 mm) and inside diameter (1580 mm) as thus:

$$F_{rot} = \tan^{-1} \left[ \frac{\Delta z}{0.5(D_o - D_i)} \right] \quad (6.1)$$

where:  $D_o$  is outside diameter [mm];  
 $D_i$  is inside diameter [mm] and  
 $\Delta z$  is flange deflection [mm];

thus:

$$F_{rot} = \tan^{-1} \left[ \frac{0.96}{0.5(1930 - 1580)} \right] = 0.349^\circ$$

where  $D_o = 1930$  mm.

$D_i = 1580$  mm and

$\Delta z = 0.96$  mm

The calculated value of 0.349 degrees, in this case marginally exceeds the ASME value of 0.3 degrees. As the stresses in the flange are acceptable, it was considered reasonable to adopt the stud bolt load of 500 kN even though the flange ring rotation was marginally exceeded.

## 6.7 Summary of Results – F.E.A Method.

A summary of the results of this chapter is as follows. Firstly, stress results of the flanged joint were presented. The maximum stress reported was at the outside diameter of the gasket.

Maximum stress results of the flange occurred at the outside diameter lower end of the flange hub. Whilst exceeding yield, the stress was generally localized and occurred at a structural discontinuity. It was not considered an issue and the levels of stress in the flange were considered acceptable.

Stresses in the blind flange were of a similar nature exceeding yield in an extremely localized area at a structural discontinuity. One again this was not considered an issue.

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A maximum deflection of 0.96 mm at the outside diameter of the flange was reported. This value corresponded to a ring rotation of 0.349 degrees which was marginally greater than 0.3 degrees as suggested by ASME VIII Division 1.

## **Chapter 7**

### **Conclusions and Recommendations**

#### **7.1 General Remarks**

The outcome of this investigation concludes that a stud bolt load of 500 kN per stud is sufficient to successfully seal the flange joint whilst not overstressing any of its component members. In conjunction with this exercise it was also established stud bolt stress levels are not excessive.

AS1210 Appendix B: Finite Element Analysis, states that F.E.A. should not be performed in isolation and should be conducted with other established methods. The target load bolt-up method was chosen to fulfill this requirement as a comparative cross-check of F.E.A. results.

## 7.2 Stress Results

Stress results obtained from the F.E.A. analysis indicated the flanged joint is within acceptable levels. It was attempted to compare the F.E.A. results with that of the target load bolt-up method. It was concluded the results were not directly comparable, however it is evident both sets of results produce a similar outcome, that is, the flanged joint is not overstressed and fit for purpose.

## 7.3 Deflection Results

Results due to deflection produced a variable outcome with the finite element method predicting a maximum flange rotation of 0.349 degrees. This value is marginally in excess of a suggested limit of 0.3 degrees taken from ASME VIII Division 1. In comparison, using calculated values from the target load bolt-up method as inputs, the rigidity index equation found in ASME VIII Division 1 also indicates the flange may be marginally in excess of the suggested limit. Therefore both methods appear to be in general agreement where flange rigidity / rotation is concerned.

As stated previously it was decided to progress with tensioning the flange at a value of 500 kN per stud bolt as compliance with the Code regarding rigidity / rotation is not mandatory but suggested.



## 7.4 Field Trial Observations and Results

After imposing a load of 500 kN per stud to the flange joint, visual monitoring of the joint over a period of two months has indicated no detectable leakage.

Monitoring has taken place at varying modes of heater operation. The operational modes include:

- Startup mode:- where heater pressure and temperature increase up to operational conditions,
- Normal operation mode:- where heater is operated normally heating sodium hydroxide and,
- Acid wash mode:- where heater is operated at a lower pressure whilst circulating sulphuric acid.

This result is in contrast to previous efforts where stud bolt tensioning has taken place with pneumatic spanners. It is apparent the higher loads imparted by the bolt tensioner and uniformity of loading has contributed to successfully sealing the joint.

## 7.5 Recommendations

It is recommended to carry out the following future actions:

- Monitor flanged joint for leakage over a period of six months. To date, monitoring has spanned two months in total. Six months is the usual period of

time before the flange joint is opened allowing inspection of internal components within the heater.

- It is the intention to also accumulate data when the flanged joint is disassembled during a routine heat exchanger outage for maintenance and inspection. To date this outage has not occurred. It is envisaged the data will take the form of recording pressure readings on the bolt tensioner as the bolt tensioner can also be used during the disassembly of the flanged joint as well. The bolt tensioner pressure will be progressively increased, stretching the stud to the point where the stud nut can be turned by hand. Conversion of this pressure reading to a stud load will indicate what load the flanged joint has retained after having been in service for a period of months.

## References

Australian Standard AS 1210, *Pressure Vessels, 1997*. Appendix B, Standards Association of Australia.

Bickford, J.H., Hayashi K, Chang A.T., Winter, J.R., February 1989. 'A preliminary evaluation of the elevated temperature behaviour of a bolted flanged connection', Bulletin No.341. Welding Research Council.

Bickford, J.H., 1995, *An introduction to the design and behaviour of bolted joints*, Dekker, New York.

Boiler and Pressure Vessel Code 1992, Section 2, *Materials, Part D Properties*. American Society of Mechanical Engineers.

Boiler and Pressure Vessel Code 2000, Section VIII Division 1, *Appendix S*. American Society of Mechanical Engineers.

Bowman, A, 2003, 'Keep Flogging Until its Tight', *Operating Pressure Equipment 7*. Proceedings of Institute of Materials Engineering Australasia Ltd.

Carter, W. & Ball, B , 2000, Appendix 2 Rules for bolted flange connections with ring type gaskets. pages 194 – 208. *Guide to ASME Section VIII Division 1 Pressure Vessels*, Volume 4, 2nd edition. McGraw Hill, New York.

Deiningner, J & Strohmeier K.,1999 *Design of a ring –joint flange in accordance to the new European standard EN 1591 and by the use of the finite element approach*, University of Technology, Munich.

<http://www.lrz-muenchen.de/projekte/hlr-projects/1997-1999/cd/daten/pdf/t7231aa.pdf>, May 2004.

Gratton, D & Kempster, PL., August 2002, AS2885.1 Issue Paper 1.9–, *Flange Bolting*. Australian Industry Pipeline Association.

<http://www.apia.net.au/issues/issuespapers.html>, May 2004.

Paulin Research Group, September 2003, *Axipro 2.0 Program Manual*. pp. 2.4.2 – 2.4.4.

Raub J., 2002. ‘Modelling diesel engine head gaskets using the gasket material option’, *Proceedings of ANSYS User’s Conference 2002*.

[http://www.ansys.com/industry/nonlinear/published\\_papers.htm](http://www.ansys.com/industry/nonlinear/published_papers.htm), May 2004.

Sawa, T, Hirose, T., and Kumano, H., "Behavior of Pipe Flange Connection in Transient Temperature Field," *ASME Journal of pressure Vessel Technology*, Vol. 115, Number 2, 1993, pp. 142-146.

Sears, G & King, D, 2003, ‘Joint Integrity Management of Critical Flanges’, *Operating Pressure Equipment 7*. Proceedings of Institute of Materials Engineering Australasia Ltd.

Singh K P and Soler A I: '*Mechanical Design of Heat Exchangers and Pressure Vessel Components*', Arcturus , New Jersey, 1984, page 81 – 126.

Smith, P and van Laan, T: '*Piping and Pipe Support Systems, Design and Engineering*' McGraw Hill, New York, 1987, page 63.

Waters E O; Wesstrom D B; Rossheim D B and Williams F S G: 'Formulas for Stresses in Bolted Flanged Connections', *Transactions of American Society of Mechanical Engineers*, Vol 59, 1937, p 161.

Yasumasa S and Satoshi N., 2000. '*Analysis of Gasketed Flanges with ordinary elements using APDL control*', Proceedings of ANSYS User's Conference 2000. [http://www.ansys.com/industry/nonlinear/published\\_papers.htm](http://www.ansys.com/industry/nonlinear/published_papers.htm), May 2004.

Zerres, H, Lemauviel, L, Perez, M, Sciffet, L, 'Comparison between the Analysis of the Mechanical Behaviour of Bolted Joints by Finite Elements Method and by the European Approach', *American Society of Mechanical Engineers, Proceedings of Pressure Vessels and Piping Conference* , PVP, v 367, Analysis of Bolted Joints, 1998, p 69-73.

Zerres, H, Perez, M, Guerout, Y, 'Metal to metal contact for gasketed assemblies – Generalisation of EN 1591 European calculation code for circular bolted flanged connections', *American Society of Mechanical Engineers Proceedings of Pressure Vessels and Piping Conference*, Seattle, 2000.

## **Appendix A**

### **Project Specification**

University of Southern Queensland  
FACULTY OF ENGINEERING AND SURVEYING

ENG 4111 / 4112 Research Project

**PROJECT SPECIFICATION**

**FOR:** **BRETT C TAYLOR.**

**TOPIC:** Assessment of appropriate pressure vessel flange bolt tension by FE modelling.

**SUPERVISORS:** Chris Snook, USQ.  
Dr Wenyi Yan, USQ.

**ENROLEMENT:** ENG 4111 – S1, X, 2004.  
ENG 4112 – S2, X, 2004.

**PROJECT AIM:** The project aims to establish maximum bolt tension allowable to seal a specific flange on a shell and tube heat exchanger.

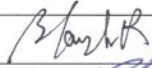

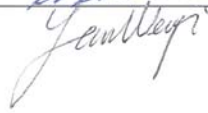
**PROGRAMME:** **Issue A, 12 March 2004.**

1. Research background information relating to assembly stresses produced when bolting two flanges together.
2. Construct model of specific flange using finite element analysis techniques.
3. Analyse output from finite element analysis model.
4. Compare output gained from model with a traditional calculation technique.
5. Recommend required bolt tension to effectively seal flanged joint.
6. Monitor via field observation, if flanged joint is successfully sealed.

As time permits

7. Compare allowable bolt tensions with those relevant to AS1210: Pressure Vessels.
8. Monitor and record bolt tension establishing if tension has reduced during the time the flanged joint has been in service.

Agreed:

Student		Date:	12/3/04
Supervisor	 	Date:	18/3/04

18.03.04

## **Appendix B**

### **Target Bolt-up Method Calculation Sheet Results**



<b>BOLTUP CALCULATION</b>	AS1210 Bickford	
<b>BOLT LOAD CALCULATIONS</b>		
<b>DESIGN PARAMETERS:</b>	<b>Operating</b>	
Pressure	4.34 Mpa	
Temperature	247 °C	
<b>GASKET DETAILS:</b>		
TYPE	=	Spiral Wound
O.D.	=	1671 mm
I.D.	=	1624 mm
t	=	5.0 mm
b <sub>o</sub>	=	11.8 mm
b	=	8.6 mm
G	=	1653.7 mm
<b>m</b>	=	3
<b>y</b>	=	69 MPa
H	=	9321.92 kN
H <sub>p</sub>	=	1168.62 kN
W <sub>m1</sub>	=	10491 kN
W <sub>m2</sub>	=	3097 kN
<b>W</b>	=	10517 kN
	where	$H = 0.785G^2P$ total hydro end force $H_p = 2b\pi GmP$ total joint contact surface compression $W_{m1} = H + H_p$ min required bolt force for operating cond. $W_{m2} = \pi bGy$ min required force for gasket seating  $(W = \max: W_{m1}, W_{m2})$
<b>STUD DETAILS:</b>		
Bolt Grade:	B7	
Size :	2.000 inch	
Number:	52	
	1729 mm <sup>2</sup>	Effective Stress Area / Bolt
	89,908 mm <sup>2</sup>	TOTAL Effective Stress Area
	172 MPa	Allowable Stress - Ambient :
	172 MPa	Allowable Stress - Operating :
	720 MPa	0.2% Proof Stress - Ambient :
	10%	Temperature Relaxation @ 1000 hours
<b>TARGETED BOLT LOAD</b>		
Min. Req'd bolt load for operating cond. =	202.3 kN	Percentages from Bickford
20% Preload Scatter =	40 kN	
20% Embedment Loss =	40 kN	
48% Elastic Interaction Loss =	97 kN	
35% Gasket creep loss =	71 kN	
Diff. thermal expansion =	50 kN	
<b>Bolt Load :</b>	<b>500 kN</b>	
<b>Bolt Stress :</b>	<b>289 Mpa</b>	
Bolt stress less than 40% Yield Yes/No	Yes	
Estimated bolt load losses at bolt-up	<b>60 kN</b>	
<b>ESTIMATED TRANSFERED LOAD : W<sub>F</sub></b>	<b>440 kN</b>	

*Below 40% yield stress corrosion cracking usually not a problem. (Bickford)*

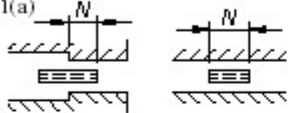
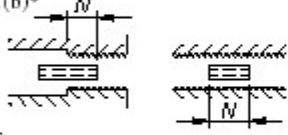
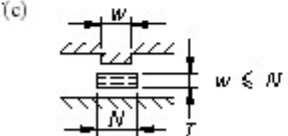
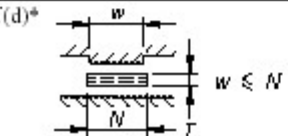
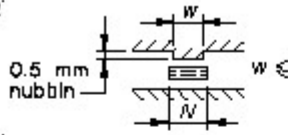
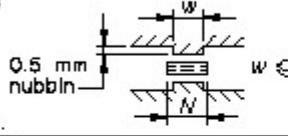


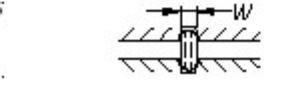
<b>BOLTUP CALCULATION</b>		AS1210 Bickford	
<b>FLANGE STRESS CALCULATIONS</b>			
<b>Design Data</b>		<b>Table 3.21.6.6 (A)</b>	
<b>Outside Diameter of Flange (A):</b>	1930 mm	$h_D =$	94.0
<b>Inside Diameter of Flange (B):</b>	1580 mm	$h_G =$	87.6
<b>Thickness of Flange (t):</b>	175 mm	$h_T =$	106.1
<b>Hub Thickness - Flange Side (<math>g_1</math>):</b>	61 mm	$h_0 =$	251.4
<b>Hub Thickness - Vessel Side (<math>g_0</math>):</b>	40 mm	$F =$	0.879
<b>Hub Length (h):</b>	75 mm	$V =$	0.366
<b>Bolt Circle Diameter (C):</b>	1829 mm	$f =$	1.198
<b>Flange Material:</b>	ASTM-A516 Gr70	$Z =$	5.064
<b>Allowable Stress - Ambient:</b>	138 Mpa	$e =$	0.003
<b>Allowable Stress - Operating:</b>	135 Mpa	$Y =$	9.818
<b>Yield Stress - Ambient:</b>	262 Mpa	$L =$	1.333
		$K =$	1.2215
		$T =$	1.8302
		$U =$	10.7885
		$d =$	11854548
<b>FLANGE STRESSES AFTER TENSIONING (Before Pressure)</b>			
Moment Component:	$M_D =$	0 kNm	
Moment Component:	$M_G =$	2005 kNm	
Moment Component:	$M_T =$	0 kNm	
Total Moment:	$M_o =$	2005 kNm	
Longitudinal Hub Stress:	$S_H =$	307 MPa	150% x Yield
Radial Flange Stress:	$S_R =$	56 MPa	100% of Yield
Tangential Flange Stress:	$S_T =$	121 MPa	100% of yield
0.5 x ( $L_{Hub} + R_{Flange}$ ):		181 MPa	100% of yield
0.5 x ( $L_{Hub} + T_{Flange}$ ):		214 MPa	100% of yield
			<b>Limits</b>
			<b>Mpa</b>
			<b>% of Limit</b>
			393
			262
			262
			262
			262
			78%
			22%
			46%
			69%
			82%
<b>FLANGE STRESSES - OPERATING (With Pressure)</b>			
Moment Component:	$M_D =$	800 kNm	
Moment Component:	$M_G =$	1188 kNm	
Moment Component:	$M_T =$	86 kNm	
Total Moment:	$M_o =$	2074 kNm	
Longitudinal Hub Stress:	$S_H =$	317 MPa	150% x Yield
Radial Flange Stress:	$S_R =$	58 MPa	100% of yield
Tangential Flange Stress:	$S_T =$	125 MPa	100% of yield
0.5 x ( $L_{Hub} + R_{Flange}$ ):		188 MPa	100% of yield
0.5 x ( $L_{Hub} + T_{Flange}$ ):		221 MPa	100% of yield
			<b>Limits</b>
			<b>Mpa</b>
			<b>% of Limit</b>
			393
			262
			262
			262
			262
			81%
			22%
			48%
			72%
			84%

## **Appendix C**

### **Selected Reference Data**




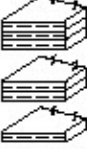




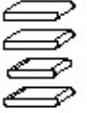
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**TABLE 3.21.6.4(B)**  
**EFFECTIVE GASKET WIDTH (See Clause 3.21.4.3)**

Facing sketch (exaggerated)	Basic gasket seating width $b$ ,	
	Column I †	Column II †
1(a) 		
1(b)* 	$\frac{N}{2}$	$\frac{N}{2}$
1(c) 	$\frac{w + T}{2}$ but not to exceed $\frac{w + N}{4}$	$\frac{w + T}{2}$ but not to exceed $\frac{w + N}{4}$
1(d)* 		
2 0.5 mm nubbln 	$\frac{w + N}{4}$	$\frac{w + 3N}{8}$
3 0.5 mm nubbln 	$\frac{N}{4}$	$\frac{3N}{8}$
4* 	$\frac{3N}{8}$	$\frac{7N}{16}$
5* 	$\frac{N}{4}$	$\frac{3N}{8}$
6 	$\frac{w}{8}$	

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**TABLE 3.21.6.4(A)**  
**GASKET MATERIALS AND CONTACT FACINGS**

Gasket factor ( $m$ ) for operating conditions and minimum design seating strength ( $\sigma$ ) (see Note 1)				Refer to Table 3.21.6.4(B)		
Gasket material (see Note 2)	Gasket factor $m$	Min. design seating stress $\sigma$ (MPa)	Sketches and notes	Use facing sketch	Use column	
<i>Self-energizing types:</i> O-rings, metallic, elastomer other gasket types considered as self-sealing	0	0	—	—	—	
<i>Elastomers without fabric or a high percent of asbestos fibre</i>				1 (a, b, c, d), 4, 5	II	
Below 75 Shore Durometer	0.50	0				
75 or higher Shore Durometer	1.00	1.4				
Asbestos with a suitable binder for the operating conditions or PTFE (see Note 4)	3 mm thick 1.5 mm thick 1 mm thick	2.00 2.75 3.50	11.0 25.5 45.0			
Elastomers with cotton fabric insertion	1.25	2.8				
Elastomers with asbestos fabric insertion, with or without wire reinforcement (see Note 3)	3-ply 2-ply 1-ply	2.25 2.50 2.75	15.2 20.0 25.5			
Vegetable fibre	1.75	7.6				
Spiral-wound metal, asbestos filled (see Note 4)	Carbon Stainless or monel	2.50 3.00	69.0 69.0			
Corrugated metal, asbestos inserted or Corrugated, metal, jacketed asbestos filled	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	20.0 25.5 31.0 38.0 45.0			
Corrugated metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	25.5 31.0 38.0 45.0 52.5			
Flat metal jacketed asbestos filled	Soft aluminium Soft copper or brass Iron or soft steel Monel 4-6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	38.0 45.0 52.5 55.5 62.0 62.0			

Flange Material: ASM A516 Grade 70:

<b>ASTM A516</b>				
<b>PRESSURE VESSEL QUALITY</b>				
<b>General Description</b> - Carbon-silicon steel plates four tensile strength ranges (Grades 55, 60, 65 and 70). Plates are made to a fine grade practice				
<b>Mechanical Properties - Tensile Requirements</b>				
	<b>Grade 55</b>	<b>Grade 60</b>	<b>Grade 65</b>	<b>Grade 70</b>
Tensile Strength, ksi	55/75	60/80	65/85	70/90
Yield Strength min, ksi	30	32	35	38
<b>Chemical Requirements - Composition Percent, %</b>				
<b>Elements</b>	<b>Grade 55</b>	<b>Grade 60</b>	<b>Grade 65</b>	<b>Grade 70</b>
Carbon, max. %	0.18	0.21	0.24	0.27
1 in. and under	0.20	0.23	0.26	0.28
Over 1 to 2 in. incl.	0.22	0.25	0.28	0.30
Over 2 to 4 in. incl.	0.24	0.27	0.29	0.31
Over 4 to 8 in. incl.	0.26	0.27	0.29	0.31
Over 8 to 12 in. incl.				
Manganese, max. %	0.60/0.90	0.60/0.90	0.85/1.20	0.85/1.20
1/2 in. and under	0.60/1.20	0.60/1.20	0.85/1.20	0.85/1.20
Over 1/2 in.				
Phosphorus, max. %	0.035	0.035	0.035	0.035
Sulfur, max. %	0.04	0.04	0.04	0.04
Silicon, %	0.15/0.30	0.15/0.30	0.15/0.30	0.15/0.30
Plates 1-1/2" Thick and Under are supplied in as-rolled condition unless ordered heat treated. Plates Over 1-1/2" Thick should be normalized.				

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**TABLE 3.3.7  
YOUNG MODULUS (MODULUS OF ELASTICITY) (E)**

Material		Young modulus, GPa															
		Temperature, °C															
Type or grade	Nominal composition	-200	-150	-100	-50	0	50	100	150	200	250	300	350	400	450	500	
Carbon and low alloy steels	C ≤ .3% C	217	213	210	207	204	201	198	195	192	189	186	179	171	162	150	
	C > .3% C	215	212	209	206	203	200	197	194	191	187	184	178	170	161	149	
	C-.5Mo, Mn-.5Mo, Mn-.25Mo, Mn-V	215	211	208	205	202	199	196	193	190	187	184	178	170	160	150	
	.5Ni-.5Mo-V, .5Ni-.5Cr-.25Mo-V,	204	201	198	196	193	190	187	184	181	178	175	171	167	163	159	
	.75Ni-.5Mo-Cr-V, .75Ni-1Mo-.75Cr,																
	.75Ni-.5Cu-Mo, 1Ni-.4Cr-.5Mo,																
	.75Cr-.5Ni-Cu, .75Cr-.75Ni-Cu-Al,																
	2Ni-1Cu, 2.5Ni, 3.5Ni,																
		.5Cr-.5Mo, 1Cr-.5Mo,	218	215	212	210	207	204	200	196	193	190	187	183	179	174	170
		1.25Cr-.5Mo(+Si), 2Cr-.5Mo															
	2.25Cr-1Mo, 3Cr-1Mo	225	222	218	215	212	209	206	203	199	196	192	188	184	179	175	
	5Cr-.5Mo(+Si, +Ti), 7Cr-.5Mo, 9Cr-Mo	—	—	—	219	215	211	207	204	201	198	194	190	190	176	168	
Stainless steels	12Cr-Al, 13Cr,	215	213	210	206	202	199	196	192	189	185	181	178	174	166	156	
	15Cr, 17Cr																
	18Cr-8Ni																
	16Cr-12Ni-2Mo, 18Cr-13Ni-3Mo																
	18Cr-10Ni-Ti																
	18Cr-10Ni-Nb																
	23Cr-12Ni, 25Cr-12Ni, 25Cr-20Ni																
	22Cr-.5Ni-.3Mo, 23Cr-4Ni																
	25Ni-20Cr-4.5Mo-1.5Cu																
	31Ni-27Cr-3.5Mo-1.0Cu																
Aluminium alloys	3003, 3004, 6061, 6063	77	75	73	72	70	68	66	63	60	—	—	—	—	—	—	
	5052, 5054	78	76	75	73	71	69	67	65	62	—	—	—	—	—	—	
	5083, 5086	79	77	76	74	72	70	68	65	62	—	—	—	—	—	—	

(continued)

## **Appendix D**

### **AS2528 Bolting Data**



**TABLE F3**  
**MECHANICAL PROPERTIES OF INCH STUDBOLTS**

Studbolt grade	Material type	Condition and recommended heat treatment	Mechanical properties							
			Tensile strength MPa	Yield stress (0.2% proof stress) MPa	Elongation percent	Impact toughness ft-lbf	HV		Hardness	
							min.	max.	min.	max.
B6	Martensitic 13% chromium steel	Hardened and tempered Hardening temperature 950°C to 1020°C Air or liquid quenched Temper at 600°C min.	760	585	15	—	195	290	185	272
B7	Chromium—molybdenum steel	Hardened and tempered	860	720	16	—	261	353	248	335
B8 (Note 4)	Austenitic chromium-nickel 18/8 steel	Solution treated 1000°C to 1100°C Liquid quenched	520	207	30	1.5 at -200°C (Note 3)	193	—	183	—
B16	Chromium—molybdenum-vanadium steel	Hardened and tempered Hardening temperature 930°C to 970°C Liquid quenched Temper at 650°C min.	860	725	18	—	261	353	248	335
B7	Chromium—molybdenum steel	Hardened and tempered Hardening temperature 850°C to 880°C Liquid quenched Temper at 600°C min.	860	725	16	20 at -100°C	261	353	248	335

**NOTES:**

1. For further information see Appendix A.
2. For corrosive service conditions, whether or not high temperatures prevail, the high chromium steel (B6) or chromium-nickel steel (B8) should be used.
3. The impact toughness requirement for strength grade B8 studbolts only applies to those used at temperatures of -200°C and below.
4. Where grade B8 is required in the solution-treated and strain-hardened condition, the mechanical properties should be agreed between the purchaser and the supplier.

## **Appendix E**

### **AS1210 Finite Element Guidance**

## FINITE ELEMENT ANALYSIS

(Normative)

**B1 GENERAL** This Appendix provides guidance for using and interpreting finite element analysis (FEA) stress results. Such guidance is necessary because the output from such an analysis, despite the sophistication of the software, is difficult to classify, that is total stresses are not readily separated into primary versus secondary, membrane versus bending and the like.

Finite element stress analysis should only be used—

- (a) alongside conventional analytical techniques, e.g.—
  - (i) nozzle local loads e.g. WRC 107 and 297;
  - (ii) standard results for plates and shells;
  - (iii) well established specialist code techniques e.g. ANSI B31.3 equations for mitre bends;
  - (iv) stress concentration factors as listed in Shigley\*; and
  - (v) other commonly employed results as listed in Roark† and Timoshenko's‡ analytical methods. (Where these analytical results exist, they should be preferred.)
- (b) to verify other calculations or analyse problems not amenable to any other technique rather than as a primary design tool; and
- (c) by experienced, competent stress analysts.

Finite element stress analysis should never be done in isolation, but should be conducted with other established methods.

**B2 CALCULATION METHODS** As a minimum, a structure shall be analysed assuming linear elastic behaviour. Nearly all necessary results can be obtained in this way. Other techniques may include for example dynamic eigenvalue, compressive buckling, heat transfer. However, this Appendix is primarily concerned with stress analysis and the interpretation of such from FEA. No further comment will be made on these other types of FEA.

Occasionally further, non-linear (e.g. plastic) analysis will be required, but such analysis should be used with caution; and only with sufficient supporting data to ensure convergence of forces and stresses.

In general, results should be reported as Tresca stresses, i.e. the difference between the maximum and minimum principal stress at any point, i.e. twice the maximum shear stress. It will be assumed that all stresses are Tresca stresses. Exceptions to this are as follows:

- (a) Shell and strut structures which could buckle, in which case the magnitude of the membrane compressive stress is important and a more elaborate buckling analysis will be necessary.
- (b) Brittle structures (e.g. some cast irons) whose compression/tension failure modes are not symmetric.

In all cases, the meshing technique should ensure the following:

- (i) Large elements are not adjacent to small elements; rather, element size varies through the structure smoothly. (The ratio of adjacent element size should not exceed 2:1).
- (ii) The aspect ratio of elements falls between 0.33 and 3.
- (iii) Four sided elements are preferable to three sided elements and higher order elements are preferable to lower order elements.
- (iv) Structural discontinuities have sufficient elements to capture the local behaviour; e.g. a cylindrical shell has a characteristic length  $L = 0.55\sqrt{Dt}$ , a hole in a plate has a characteristic length equal to its radius. In such cases, at least two quadratic elements or six linear elements within this length are required to capture local behaviour, where this is important.
- (v) Benchmark standard results can be used to help verify the output, e.g. membrane or bending stress well away from structural discontinuities.
- (vi) A mesh/grid whose element spacing varies smoothly throughout the structure is selected
- (vii) Boundary conditions (e.g. planes of symmetry and imposed loads) can be readily verified.

**B3 EVALUATION OF RESULTS** In order to evaluate the stresses calculated in Paragraph B2 for non buckling structures, the stresses shall be classified according to their-

- (a) distribution through a thickness; and
- (b) nature, whether self-limiting (secondary) or non-self-limiting (primary).

When stresses have been suitably classified as above, they can then be compared to the appropriate limits in Appendix SH of AS 1210 Supplement 1 using the appropriate basic design stress,  $f$ .

Extreme caution and considerable experience is required to evaluate FEA buckling results due to the highly variable sensitivity of structures to initial imperfections. Such sensitivities will greatly influence the choice of safety factors which can vary from 3 for cylinders to more than 14 for spheres.

It is also useful to inspect the results to ensure consistency and credibility using the following criteria:

- (i) Output contours are free of local meshing anomalies such as scalloping.
- (ii) The deflection of the structure appears reasonable in shape and magnitude.
- (iii) The maximum variation in stress across any element as proportion of the total variation in Tresca stress does not exceed the following:

Element order	Maximum stress variation
0	10%
1	20%
2	30%
>2	40%

**B4 DISTRIBUTION OF STRESS** The distribution between membrane stress  $\sigma_m$  (constant across thickness) and bending stress  $\sigma_b$  (proportional to distance from mid-plane) is found by the following equations:

$$\sigma_m = \frac{1}{t} \int \sigma dx \quad \dots \text{B4(1)}$$

$$\sigma_b = \frac{6}{t^3} \int \sigma x dx \quad \dots \text{B4(2)}$$

where

$x$  = distance from mid-plane of thickness.

For plate elements whose formulation assumes linear distribution through thickness these stresses are most easily found from:

$\sigma_m$  = mid-plane stress

$\sigma_b$  = surface stress – mid-plane stress

**B5 NATURE OF STRESS** In the absence of elaborate non-linear (plastic) analysis, the nature of stresses (whether self-limiting or not) shall be inferred using linear superposition by:

- (a) Separating mechanically induced stresses (e.g. from pressure) from known secondary stresses (e.g. thermal).
- (b) Estimating the subtracting out the component of a stress in the vicinity of a structural discontinuity due to known stresses, which can be readily calculated by simple analytical techniques e.g. membrane pressures stresses and flat plate bending stresses.
- (c) Calculating the component of a stress due to mismatch, e.g. cladding, interface or other self-limiting effects.

**B6 REPORTING RESULTS** When finite element results are used to establish the integrity of critical equipment, it is important to report the results in such a way as to facilitate their verification. Such a report shall include, but not be limited to the following:

- (a) Plot(s) of the deflected shape(s) of the structure under all relevant loading conditions.
- (b) Type of mesh used.
- (c) The loads used.
- (d) The boundary conditions used.
- (e) Evidence that the solution has converged.
- (f) Sufficient data to show that away from structural discontinuities the stresses are those of simple shell or strut models.
- (g) A description of the model and the assumptions used.
- (h) Software package and version used.