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DESIGN OF A LARGE RECTANGULAR FLANGE

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ABSTRACT

A large rectangular flange (5' wide x 12.5' Long) was designed using finite element analysis for a horizontal mixer vessel. The mixer vessel contained a large horizontal agitator with the shaft protruding through the two flat ends of the vessel. The horizontal vessel was split in the middle horizontal plane creating a large rectangular opening to be sealed by the two large rectangular flanges.

The size of the flange, the type of gasket, the bolt preload required to obtain a reasonable seal made it a design challenge to design this bolted flange assembly. To start with, an estimate was made based on the calculation of the thickness of the flange using an equivalent circular flange.

The finite element analysis of the whole assembly was preformed using the FEA software ANSYS. After several iterations, an acceptable solution was found with acceptable flange and bolt stresses. The seating stress in the gasket was also above the recommended gasket seating stress. Thus, the flanged joint was designed to be in compliance with ASME B&PV Code, Section VIII, Div-1.

The vessel and the bolted flange assembly was successfully fabricated and hydrotested based on this design and it is successfully operating in the field.

INTRODUCTION

A mixer vessel (Vessel Shell: SA-240 Grade 316L) was to be designed for a large horizontal agitator with the shaft protruding through the two flat ends of the vessel. The horizontal vessel was split in the middle horizontal plane creating a large rectangular opening.

This rectangular opening was to be sealed using the two large rectangular flanges (5' wide x 12.5' Long flange, Material: SA-516 Grade 70). The large rectangular flanges were designed using finite element analysis. The software used for FEA was ANSYS. The dimensions for the finite element model are shown in Fig. 1.

The design conditions for the vessel were 50 psig @ 300 °F. The vessel had flat heads without stiffeners. The preliminary engineering design was performed using the pressure vessel design software COMPRESS.

The finite element analysis results were validated using the criteria given in ASME Boiler & Pressure Vessel Code, Sec. VIII, Div. - 2, Appendix - 4. The FEA plots for the model and results are attached here. The vessel digital photographs taken in fabrication shop are attached at the end of the paper.

Apart from finite element analysis for large rectangular flange as presented here, separate FEA's were performed for rectangular access opening reinforcement and bolted cover plates for access door openings. These FEA's were performed to complete the vessel design.

GASKET SELECTION

The gasket selection for this large flange was a challenging task as it required a balance between the complete gasket seating and avoiding the leakage from the gasket during operation of the vessel. After talking to gasket manufacturers, it was decided to use PTFE roll gasket (GARLOCK Style 3535, 0.75" wide x 0.300" thick) requiring seating stress in the range of 3000 - 4000 psi. This roll of gasket was installed along the entire flange length inside the bolting pattern.

The gasket curves (Compression vs. Load curve, Gasket thickness vs. Clamping pressure) were used to achieve proper sealing of the gasket during design and testing stages. The flange and gasket dimensions are shown in Fig. 2.

PRELIMINARY ENGINEERING DESIGN

The pressure vessel design software COMPRESS was used to design the vessel shell, jacket, nozzles, solid unstiffened flat welded covers and saddle supports. The COMPRESS software was also used to design circumferential integral flanges with the same design conditions and gasket materials etc. The flange dimensions obtained by this method were used as guidelines to obtain preliminary sizes for the large rectangular flanges and gasket seating arrangements.

FINITE ELEMENT MODEL

The coordinate directions for Finite Elements analysis are as follows:

X: Along the vessel axis
Y: Perpendicular to the vessel axis
Z: DOWN (Vertical)

The one half (1/2 model) of the vessel with rectangular flange was meshed to generate finite element model for the analysis. Two different cases (gasket seating and operating case) of this model were run to perform the analysis. The FEA cases are covered in this report and are listed as follows:

The modeling was done using SOLID45 elements which are 8-noded brick elements with 3 DOF at each node. The pretension elements PRETS179 were used to apply bolt preload of 25000 psi. The mesh plots for the finite element model are shown in Fig. 3 and 4.

BOUNDARY CONDITIONS

The symmetry boundary conditions were applied to all the nodes on the cut surface of the one half (1/2) model. Two additional nodes were constrained in lateral direction for stability of the model.

The 50 psig design pressure was applied to all the internal surfaces of the vessel.

The boundary condition plots are shown in Fig. 5.

DISCUSSION

The circular cylindrical block at the center of the flat welded head is 4.5" wide based on the 4" flange width and 0.5" thickness of flat head. The modeled block was 5.75" wide with 4" flange width and 1.75" flat head plate.

STRESS ANALYSIS RESULTS

Please refer to the attached FEA analysis and FEA plots. The following discussion covers the explanation of couple of high stress spots in the FEA model:

- The peak stress intensity of 105,637 psi at the juncture of thick head (FEA model only) and shell is due to stress concentration factor at the sharp discontinuity present there. This peak stress will not be present for the head that will be used. The actual head being used for this application has a radius at this juncture and it is in compliance with the ASME code.
- The peak stress of 93,918 psi (present in a very tiny zone) in the flange is near the 1.75" thick flat head. This is secondary stress caused by constraints of thick material and it is self relieving and therefore, it is not a concern.

Apart from these two zones discussed above, the majority of the FEA model has following listed values of stress intensity:

- Maximum Stress Intensity in the longitudinal flange:
  Sint = 70,479 psi < Sallow (= 71,600 psi) **OK**
- Maximum Stress Intensity in the shell:
  Sint = 47,040 psi < Sallow (= 50,100 psi) **OK**
- Average Gasket Seating Stress:
  GKSX = 4156 psi > GKSXreqd (= 3,000 psi) **OK**
- Maximum Gasket Compression:
  GKD = 0.127" < GKT (= 0.300") **OK**

The displacement plot for vessel and flange model is shown in Fig. 6. The reaction loads for bolts and gaskets and gasket closure plot for PRELOAD and PRELOAD + PRESSURE cases are shown in Fig. 7 and 8. The displacement contours for the model are shown in Fig. 9.

The Von Mises stress plots for PRELOAD and PRELOAD + PRESSURE cases are shown in Fig. 10 & 11. The Stress Intensity plots for PRELOAD and PRELOAD + PRESSURE cases are shown in Fig. 12 & 13.

The gasket stress plots for PRELOAD case are shown in Fig. 14 and 15. The gasket stress plots for PRELOAD + PRESSURE case are shown in Fig. 16 and 17.

DESIGN SUMMARY

Based on the engineering calculations and finite element analysis, following are the specific designs based on the specified design parameters:

Vessel Shell & Jacket:

- Main Cylinder (Mat. SA-240 Grade 316L): 60" ID X 150" Length x 0.50" thk.
- Jacket Cylinder (Mat. SA-240 Grade 316L): 0.25" thk.
- Left and Right flat welded heads (Mat. SA-240 Grade 316L): 1.75" thk. (Without stiffeners)

Longitudinal Flange (Welded Integral Flange):

- Flange Material: SA-516 Grade 70
- Flange Dimensions: 4.0" Wide x 3.0" thk.
- Approx. Bolt Spacing: 3.0"
- Bolt Dia.: 1.00"
- Gasket Material: GARLOCK Style 3535
- Gasket Dimensions: 0.75" wide x 0.300" thk.

Split Flange Block & Cover Flange:

- Material: SA-182 Grade F316L
- Size & Rating: 14” Class 150 Studding Outlet - Flat faced With Thickness of 6” (Please note that this is different form the standard 2” thickness)
- Outside Diameter: 21”
- Inside Diameter: Approximately 15” (can be customized as per requirement)
- Number of Stud holes: 12
- Stud Circle Diameter: 18.75”
- Tap Size: 1” (8 threads per inch)
- Tap Depth: 1.12”
- Cover Flange: 14” Class 150 Blind Flange (Material: SA-182 Grade F316L)
- Special Request: SPLIT the studding outlet in the middle to have two semi-circular blocks

Gasket:

- Gasket Material: GARLOCK Style 3535
- Gasket Dimensions: 0.75” wide x 0.300” thk.
- Recommended gasket seating pressure: 3,000 - 4,000 psi

Bolts:

- Bolt Material: SA-193 Grade B7
- Nut Material: SA -194 Grade 2H
- Minimum Recommended Bolt Preload: 25,000 psi
- Maximum Recommended Bolt Preload: 60,000 psi

The vessel digital photographs taken in fabrication shop are shown in Fig. 18, 19 and 20.

CONCLUSION

From the detailed plots for gasket stresses, it can be seen that the gasket stresses are above 3000 psi at some point across the gasket width and that is sufficient to seal the gasket for both the Gasket Seating and Operating conditions.

Based on the results of the finite element analysis, it can be concluded that the flanges and the associated gasketed joint are in compliance with ASME Boiler & Pressure Vessel Code, Section VIII, Div. - 1.

REFERENCES

- ASME Boiler & Pressure Vessel Code, Sec, VIII, Div. - 1, 2001 Edition with 2003 Addenda.
- ASME Boiler & Pressure Vessel Code, Sec, VIII, Div. - 2, 2001 Edition with 2003 Addenda.
- ASME Boiler & Pressure Vessel Code, Sec, II, Part D, 2001 Edition with 2003 Addenda.

ALLOWABLE STRESS CALCULATIONS:

Material: SA-240 Grade 316L

Allowable Stress in tension at 70 °F per ASME Code, Section VIII, Div.-1: Sma= 16,700 psi
Allowable Stress in tension at 300 °F per ASME Code, Section VIII, Div.-1: Smo= 16,700 psi
Per ASME Section VIII, Div.-2, Appendix - 4:

General Primary Membrane Stress Limit =Pm = Smo = 16,700 psi

Primary Local Membrane and Bending Stress Limit = PL + Pb = 1.5*Smo = 25,050 psi

Primary Local Membrane + Bending + Secondary Stress Limit:
= PL+Pb+Q = Larger of (3*Smavg , 2*Syavg)

Average of allowable stresses: Smavg = 0.5*(16,700 + 16,700) = 16,700 psi
Sallow1 = 3*Smavg = 3*16,700 psi = 50,100 psi

Yield Strength of SA-240-316L @ 300 °F: Syo = 19,000 psi
Yield Strength of SA-240-316L @ 70 °F: Sya = 25,000 psi

Average of Yield Stresses: Syavg = 0.5*(19,000 + 25,000) = 22,000 psi
Sallow2 = 2*Syavg = 2*22,000 = 44,000 psi

Allowable stress per ASME Sec. VIII, Div. - 2, Appendix - 4 (PL + PB + Q):
Sallow = Larger of (Sallow1 , Sallow2)
Sallow = Larger of (50,100 psi , 44,000 psi)
Sallow = 50,100 psi

Material: SA-516 Grade 70

Allowable Stress in tension at 70 °F per ASME Code, Section VIII, Div.-1: Sma= 20,000 psi
Allowable Stress in tension at 300 °F per ASME Code, Section VIII, Div.-1: Smo= 20,000 psi
Per ASME Section VIII, Div.-2, Appendix - 4:

General Primary Membrane Stress Limit =Pm = Smo = 20,000 psi

Primary Local Membrane and Bending Stress Limit = PL + Pb = 1.5*Smo = 30,000 psi

Primary Local Membrane + Bending + Secondary Stress Limit:
= PL+Pb+Q = Larger of (3*Smavg , 2*Syavg)

Average of allowable stresses: Smavg = 0.5*(20,000 + 20,000) = 20,000 psi
Sallow1 = 3*Smavg = 3*20,000 psi = 60,000 psi

Yield Strength of SA-240-316L @ 300 °F: Syo = 33,600 psi
Yield Strength of SA-240-316L @ 70 °F: Sya = 38,000 psi

Average of Yield Stresses: Syavg = 0.5*(33,600 + 38,000) = 35,800 psi
Sallow2 = 2*Syavg = 2*35,800 = 71,600 psi

Allowable stress per ASME Sec. VIII, Div. - 2, Appendix - 4 (PL + PB + Q):
Sallow = Larger of (Sallow1 , Sallow2)
Sallow = Larger of (60,000 psi , 71,600 psi)
Sallow = 71,600 psi

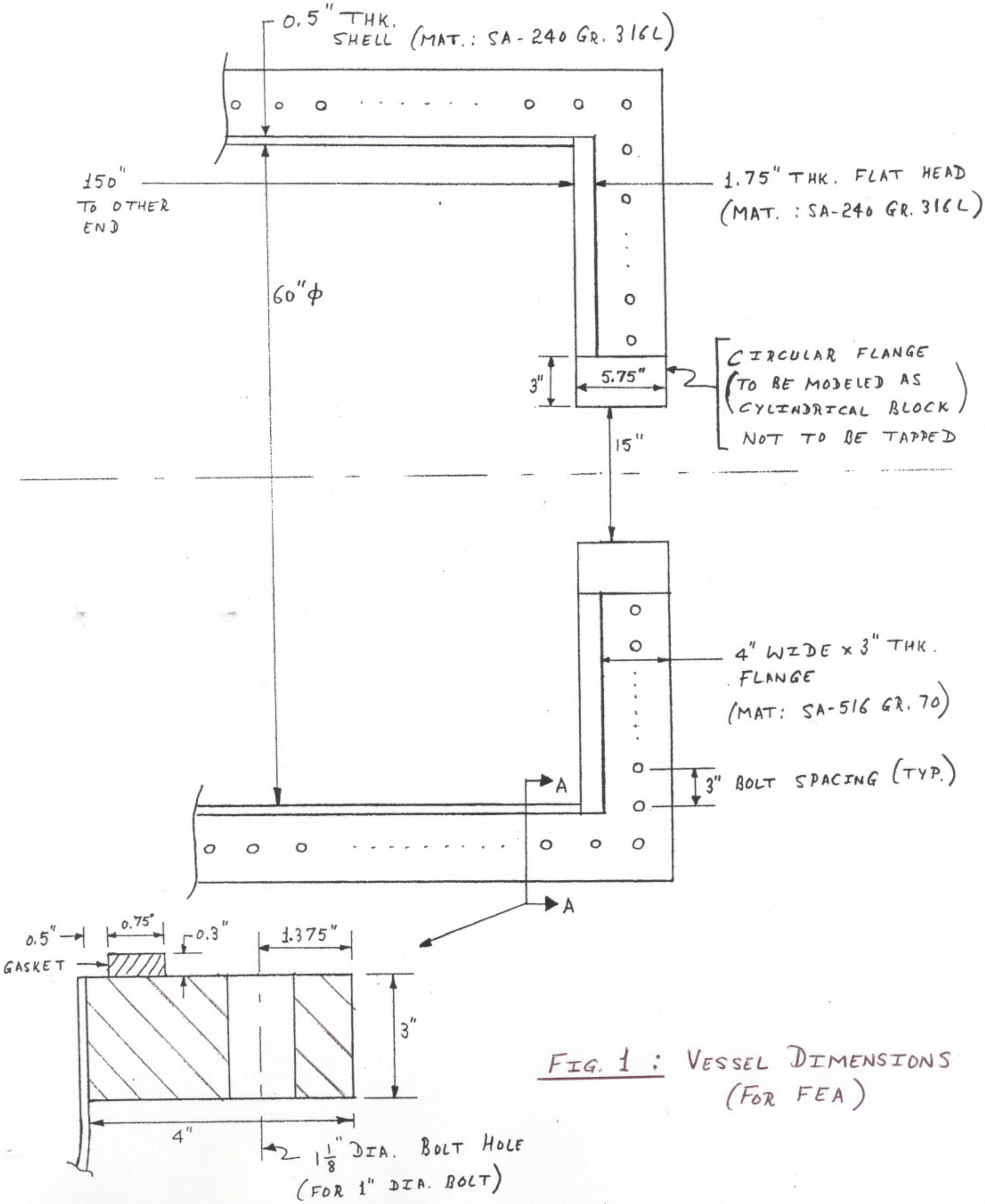

FIG. 1 : VESSEL DIMENSIONS (FOR FEA)

FEA PLOTS

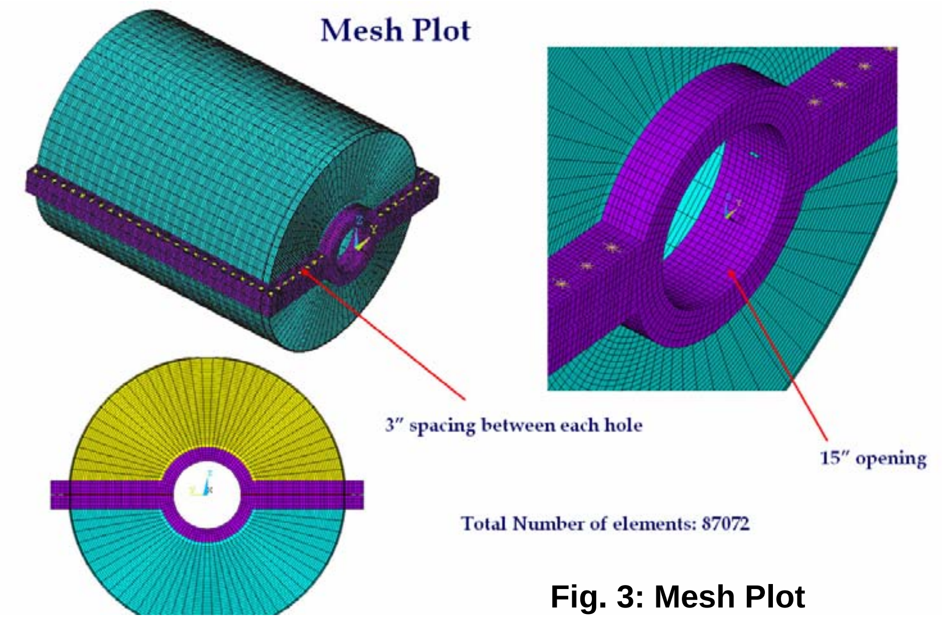

Fig. 3: Mesh Plot

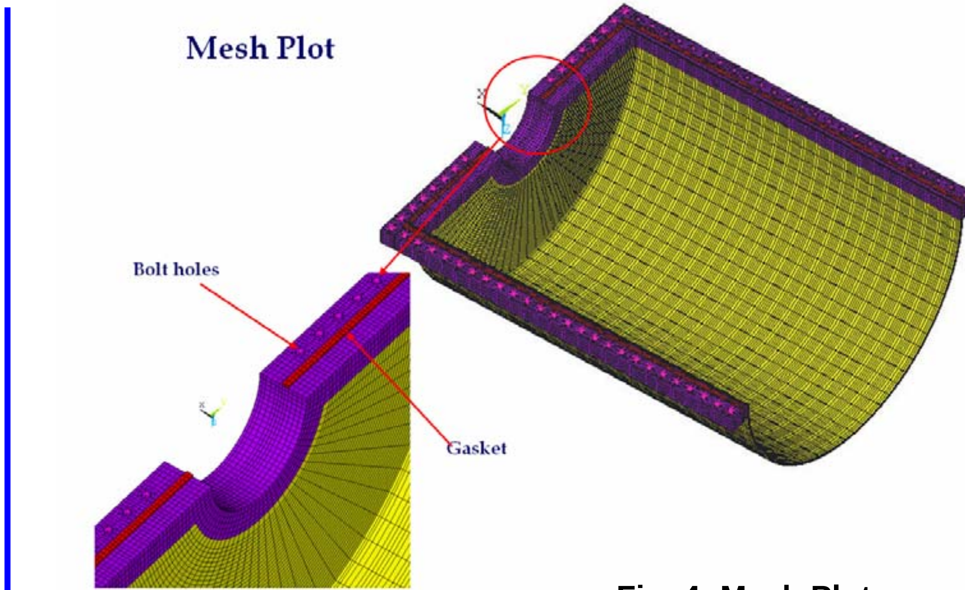

Fig. 4: Mesh Plot

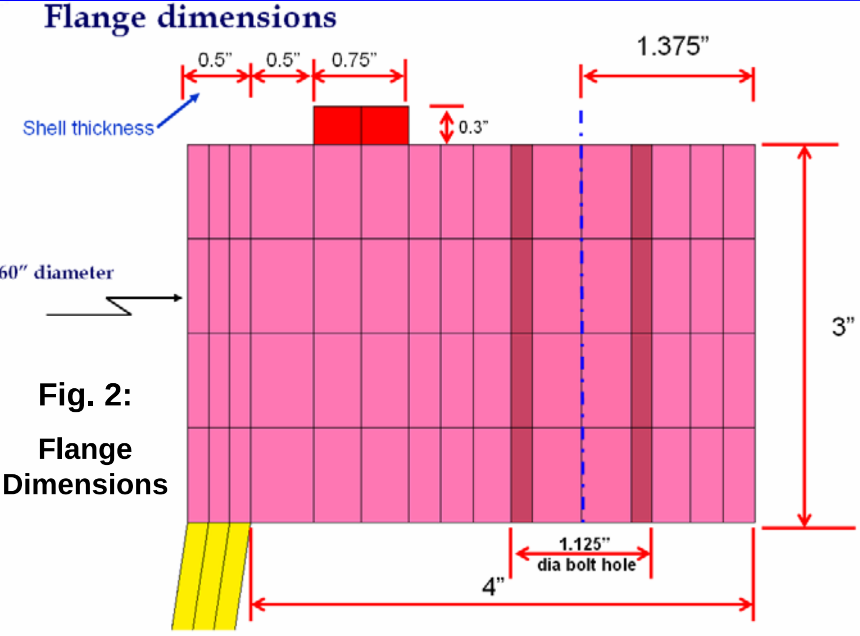

Fig. 2:

Flange Dimensions

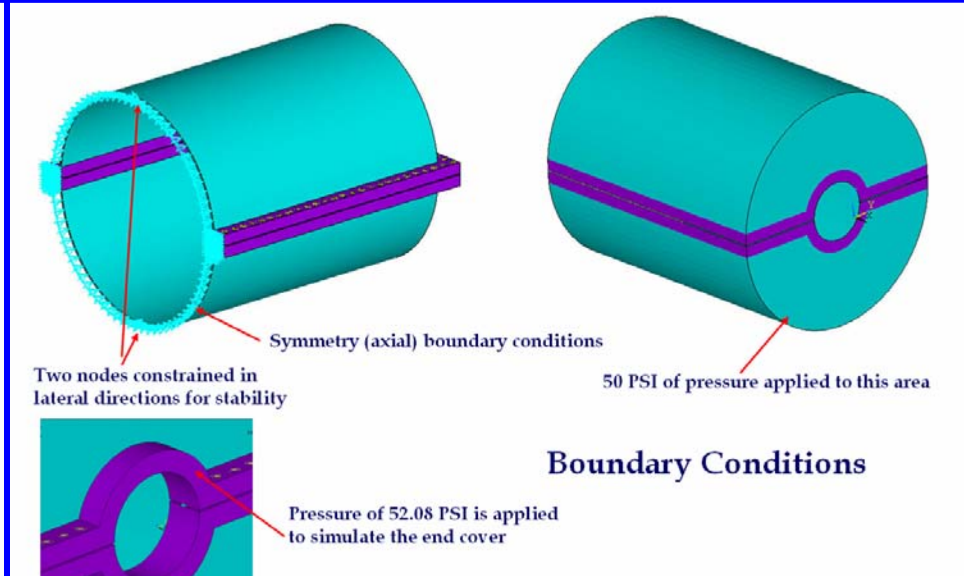

Fig. 5: Boundary Conditions

FEA PLOTS

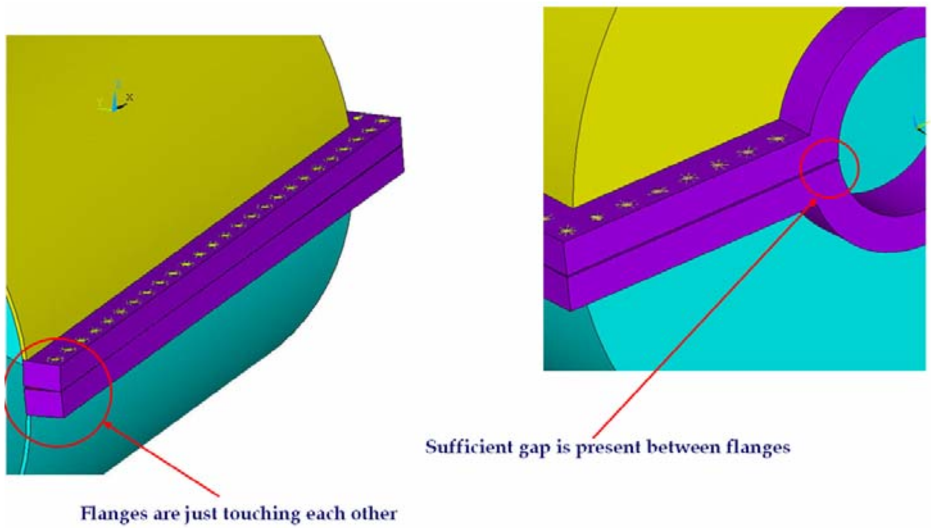

Fig. 6: Displacements

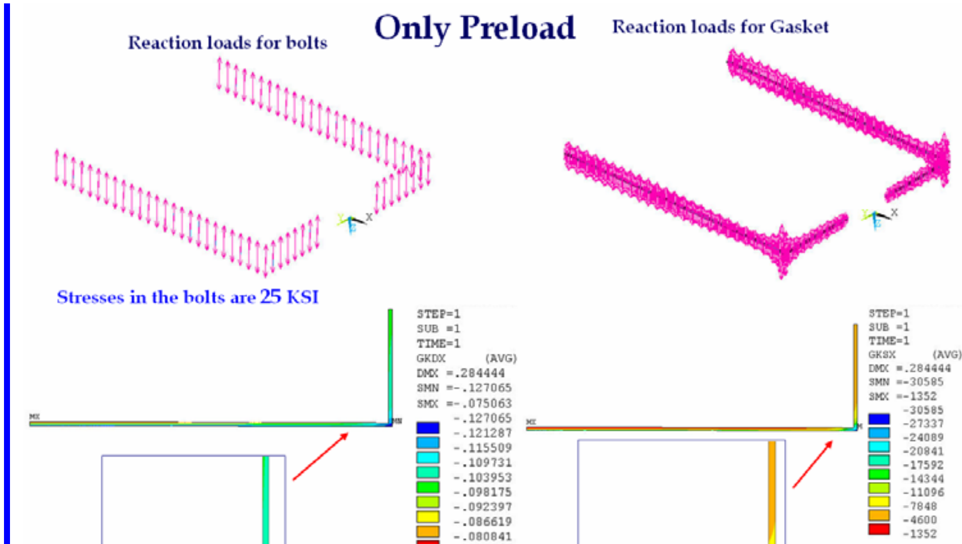

Fig. 7: Reactions for Preload

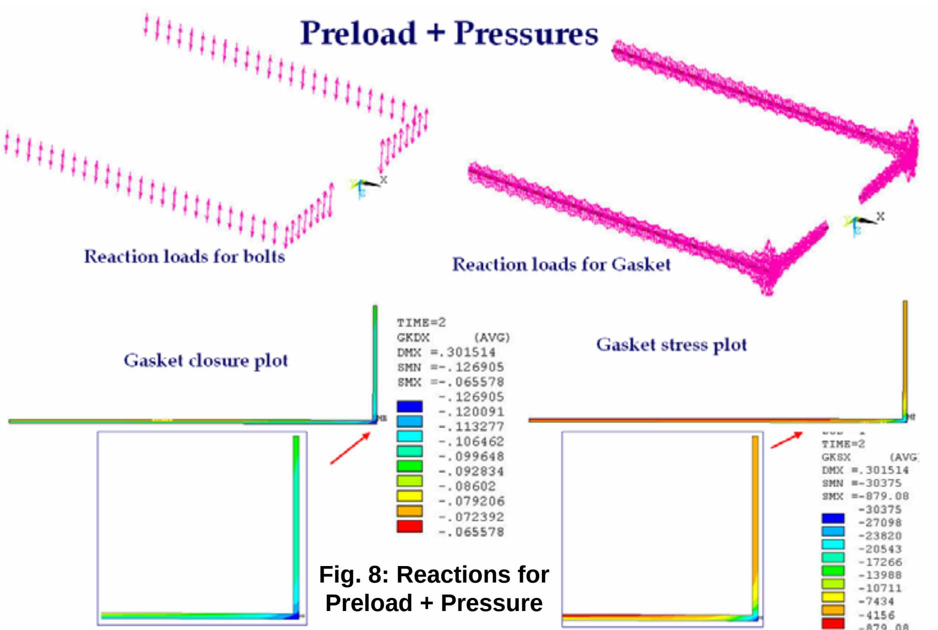

Fig. 8: Reactions for Preload + Pressure

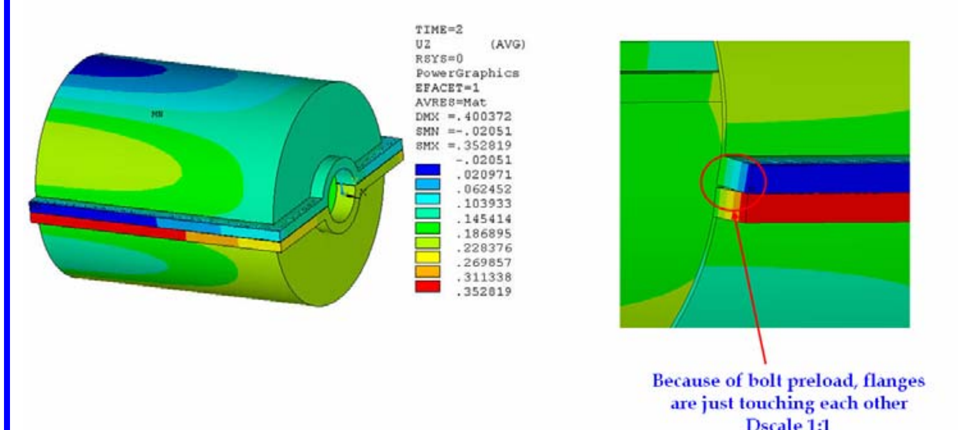

Fig. 9: Displacement Contours

FEA PLOTS

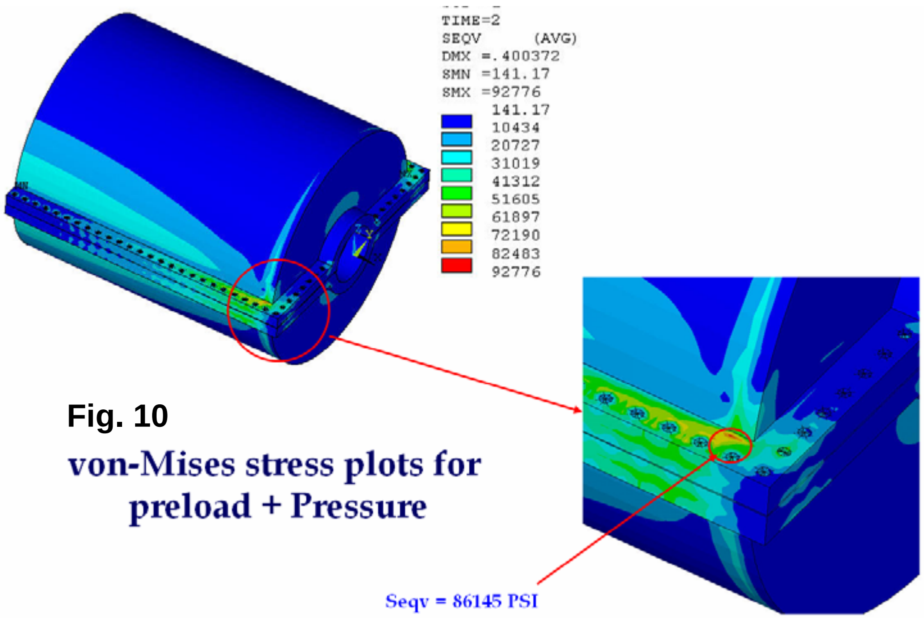

Fig. 10

von-Mises stress plots for preload + Pressure

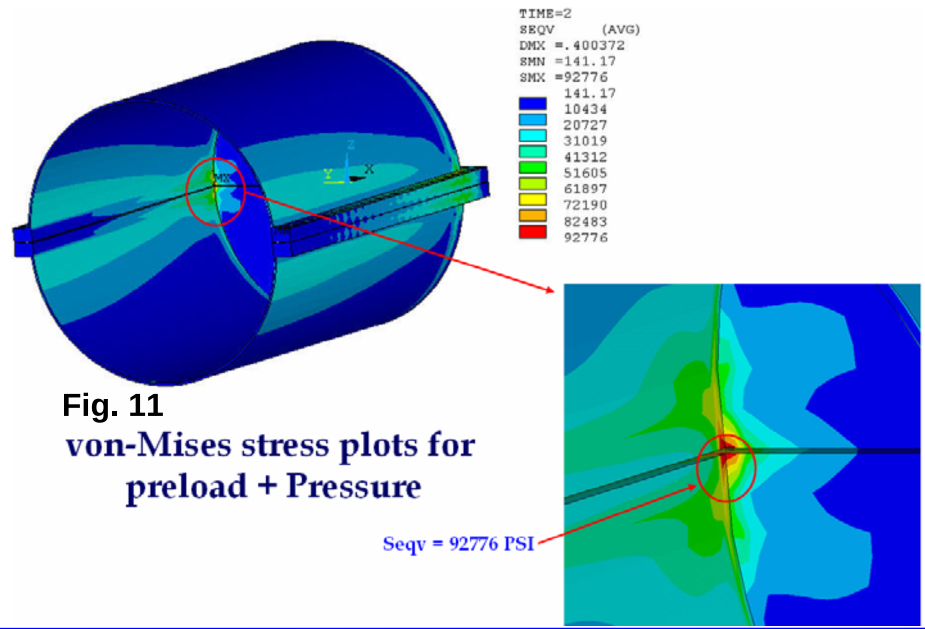

Fig. 11

von-Mises stress plots for preload + Pressure

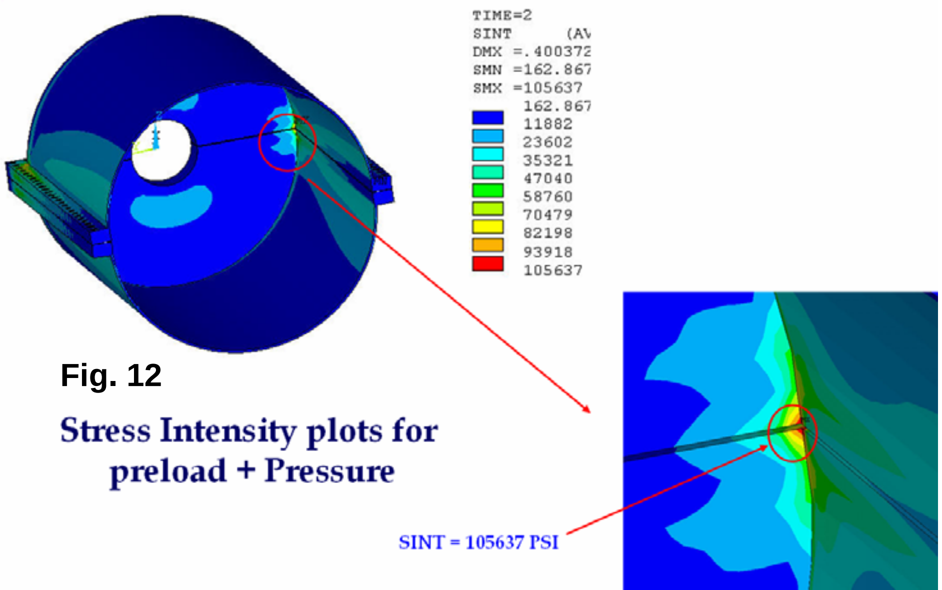

Fig. 12

Stress Intensity plots for preload + Pressure

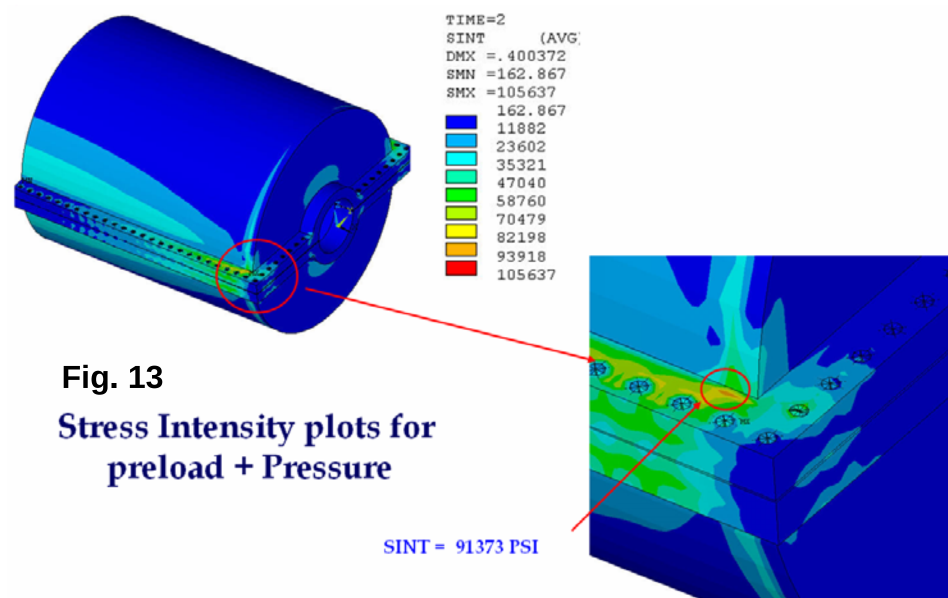

Fig. 13

Stress Intensity plots for preload + Pressure

FEA PLOTS

Gasket Stresses for Preload

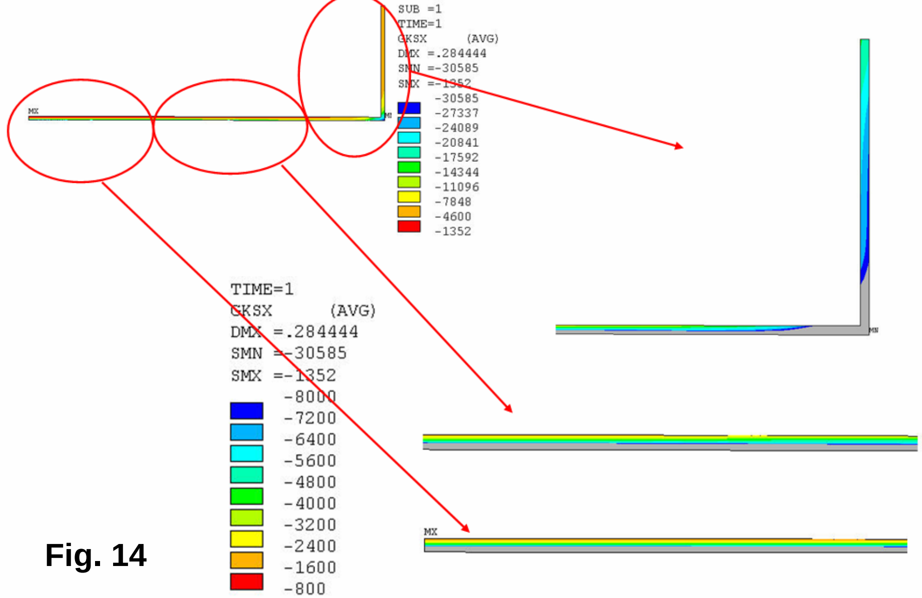

Fig. 14

Gasket Stresses for Preload

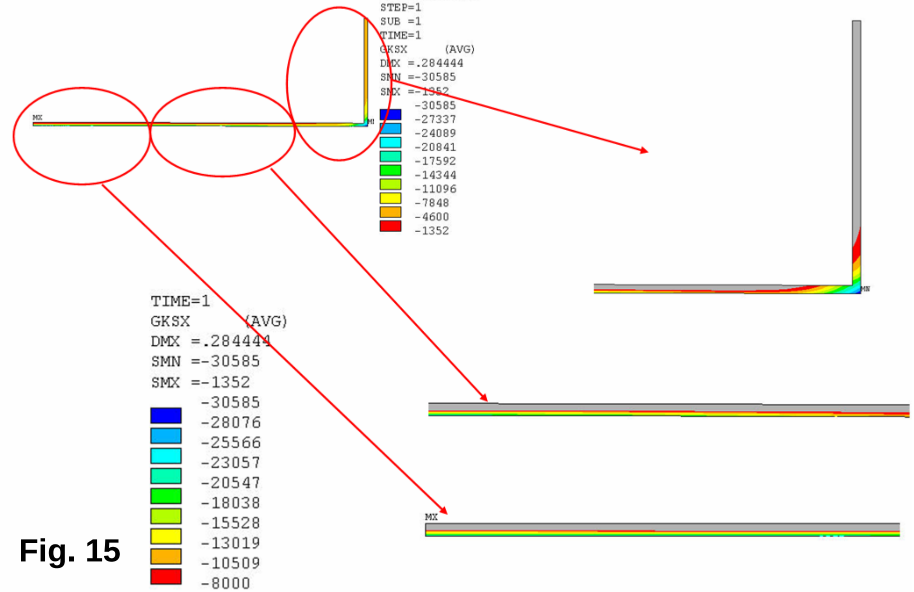

Fig. 15

Gasket Stresses for Preload + Pressures

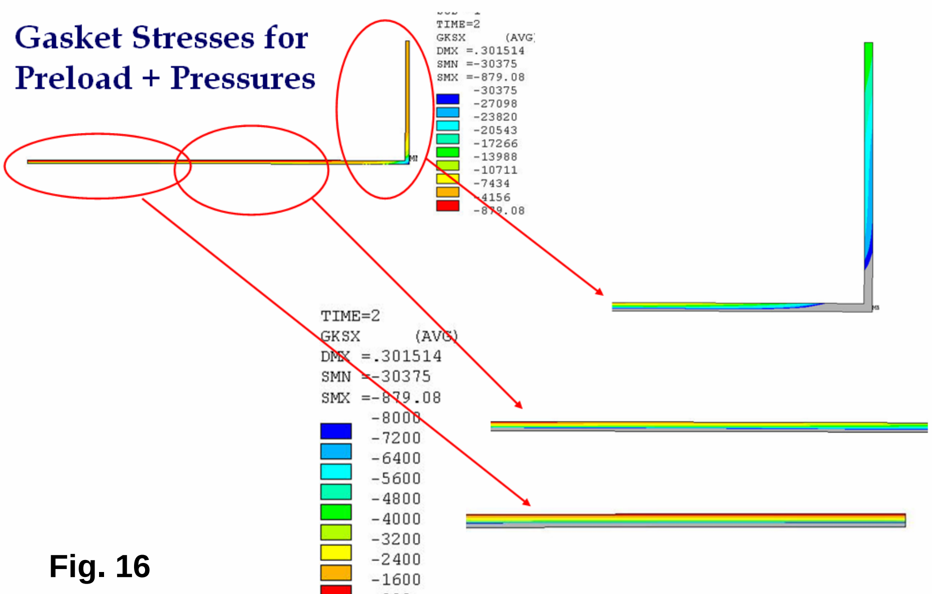

Fig. 16

Gasket Stresses for Preload + Pressures

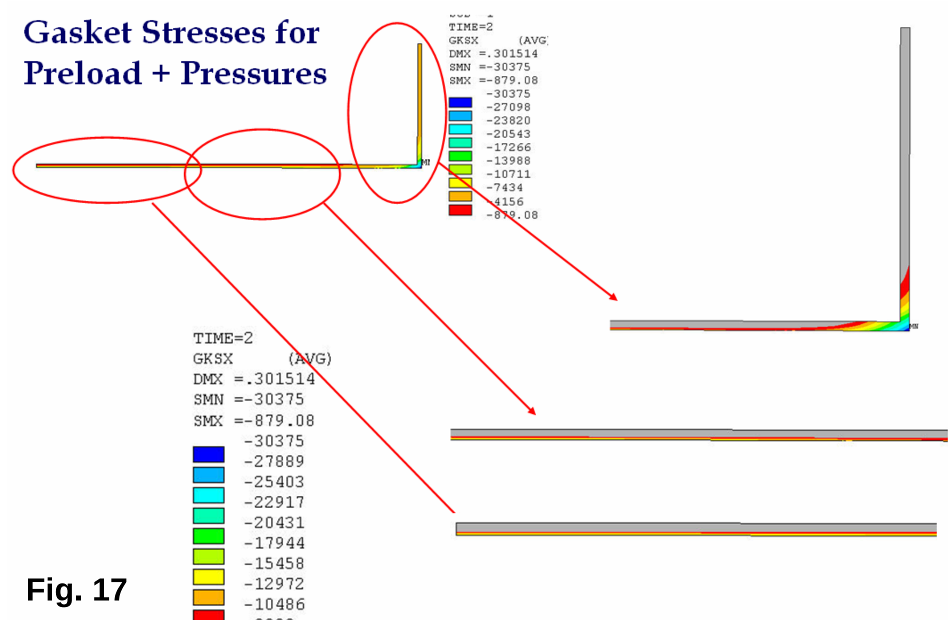

Fig. 17

Fig. 18

Fig. 19

Fig. 20