

# **ANALYSIS OF GASKETED FLANGES WITH ORDINARY ELEMENTS USING APDL CONTROL**

**Yasumasa Shoji,**

**Satoshi Nagata,**

**Toyo Engineering Corporation, Narashino-City, Chiba, Japan**

## **ABSTRACT**

The analysis of gasketed flanges is now becoming an essential technique to design pressure vessels or piping, because the design code is being modified. On the other hand, the gasket has an extreme nonlinearity which causes the analysis to be very difficult, especially using linear material properties. A finite element analysis program other than ANSYS has the element library for gasket element, but ANSYS does not have it yet.

The authors developed a method to model this nonlinear gasket by using the elements available in ANSYS Rel. 5.5 for 2-dimensional axisymmetric analysis and those in Rel. 5.6 for 3-dimensional analysis. In this paper, the concept and the procedure of the modeling is introduced. The data input is possible using a template, without using any other ANSYS commands. The whole analysis execution is controlled by APDL.

The used elements are 8-noded axisymmetric quad elements, contact ( gap ) elements and a bar element for 2-D. In the same manner, 8-noded brick elements, contact ( gap ) elements and a new pre-tension elements are used for 3-D.

## **INTRODUCTION**

In these days, a new design code about bolted joints is investigated. Research on strength of joint, gasket behavior, stress distribution are conducted for better code utilization. However, an established and authorized analysis method is not yet available, because the gasket has highly nonlinear property.

On the other hand, thanks to the emerging computer technology and software programming, nonlinear analyses are becoming routine problems. For gasket modeling, ABAQUS supports gasket elements [ HKS,1999 ] in its element library, which are still special analyses.

In this paper, authors propose a new procedure to solve the bolted joint problems using ANSYS with normal elements. The procedure is controlled by APDL [ ANSYS,1999a, 1999b ], an ANSYS feature. The procedure is applied to 2-dimensional axisymmetric analysis and 3-dimensional analysis. In the axisymmetric analysis, a bar element is used to represent bolts, while pre-tension elements are used in 3-dimensional analysis. The pre-tension elements can be replaced by several bar elements.

## **BOLTED JOINTS**

Bolted joints are widely used in chemical plants and power plants to connect equipment to pipes or pipes together. This type of joints are used frequently and the design methods are specified in various codes [ ASME, 1999 ].

The structure of bolted joints is that a pair of stiff plates called "flanges" are attached to the pipe or equipment

nozzles, sandwich a gasket for fluid seal and clamped by a number of bolts and nuts. Sealing performance depends on how the gasket is compressed [Bickford, 1998 ].

Gasket stresses depend on bolt tension and whether the tension is transferred to the gasket or not is dependent on the stiffness of the flanges. Accordingly, the design codes classify flanges by service fluids and internal pressure. The classification is called "rating" and the flange design is carried out according to the rating. Normally the codes provide standard flanges which cover individual ratings. Flange design is now the selection of flanges from the standard.

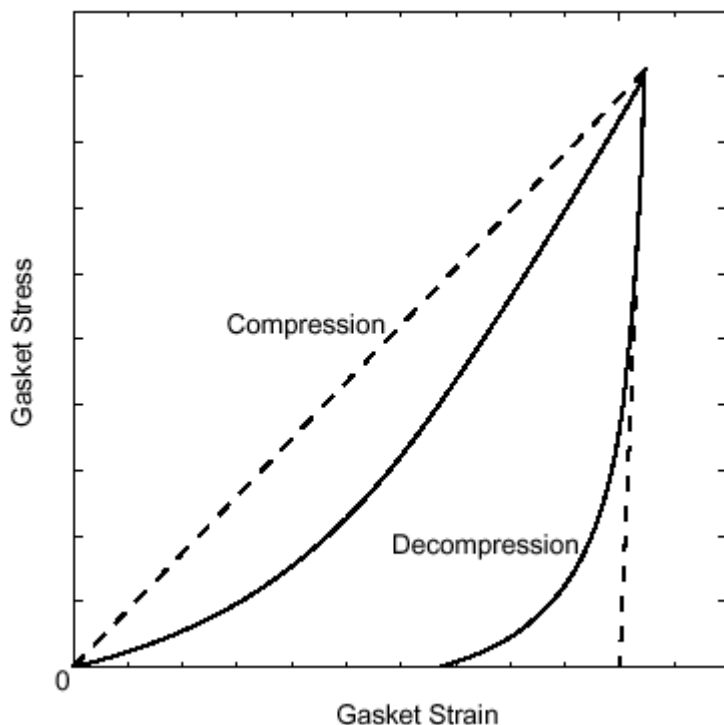
The problem of this method is that the codes are outdated and they do not predict the real behavior of the gaskets because they were established mainly based on the experiences. Many research have been conducted to overcome the problem. Especially the highly nonlinear properties of gaskets causes difficulty in understanding gasket behavior.

Thus in the field of research of bolted joints, many papers deal with gasket behavior. Many research are conducted to know how much load are to be applied to bolts, how scattered the bolt loads are and how the scatter are to be prevented [ Sawa, 1991, Bouzid, 1999, Shoji,1999 ].

In the following, some issues on gasket nonlinearity is addressed [Fukuoka, 1999 ].

Figure 1 shows a typical stress-strain relation of a gasket. The modulus of elasticity for bolting-up, or compression, is quite different from that for internal pressurization of the vessel, or decompression of the gasket. The ratio of elasticity in decompression to that in compression may exceed 10. When the gasket is decompressed, it shows strong hysteresis which is remarkably non-linear and causes permanent deformation. This deformation naturally results that the gasket does not recover the original thickness. In analyzing gasketed flanges, we must take special care about the status of the gasket, either in compression or in decompression, to choose gasket modulus of elasticity and the stress-free thickness.

**Figure 1. TYPICAL STRESS-STRAIN RELATION OF GASKET**



## ANALYSIS METHOD

In this section, method to analyze the bolted joints with ANSYS is described. Unfortunately, ANSYS does not have gasket elements, but has APDL to control analysis procedure or analysis flow. This APDL enables the gasket analysis procedures without special gasket elements.

Analyses are performed for both 2-dimensional axisymmetric models and 3-dimensional models. Axisymmetric analysis was carried out using ANSYS release 5.5 and 3-dimensional analysis by release 5.6.

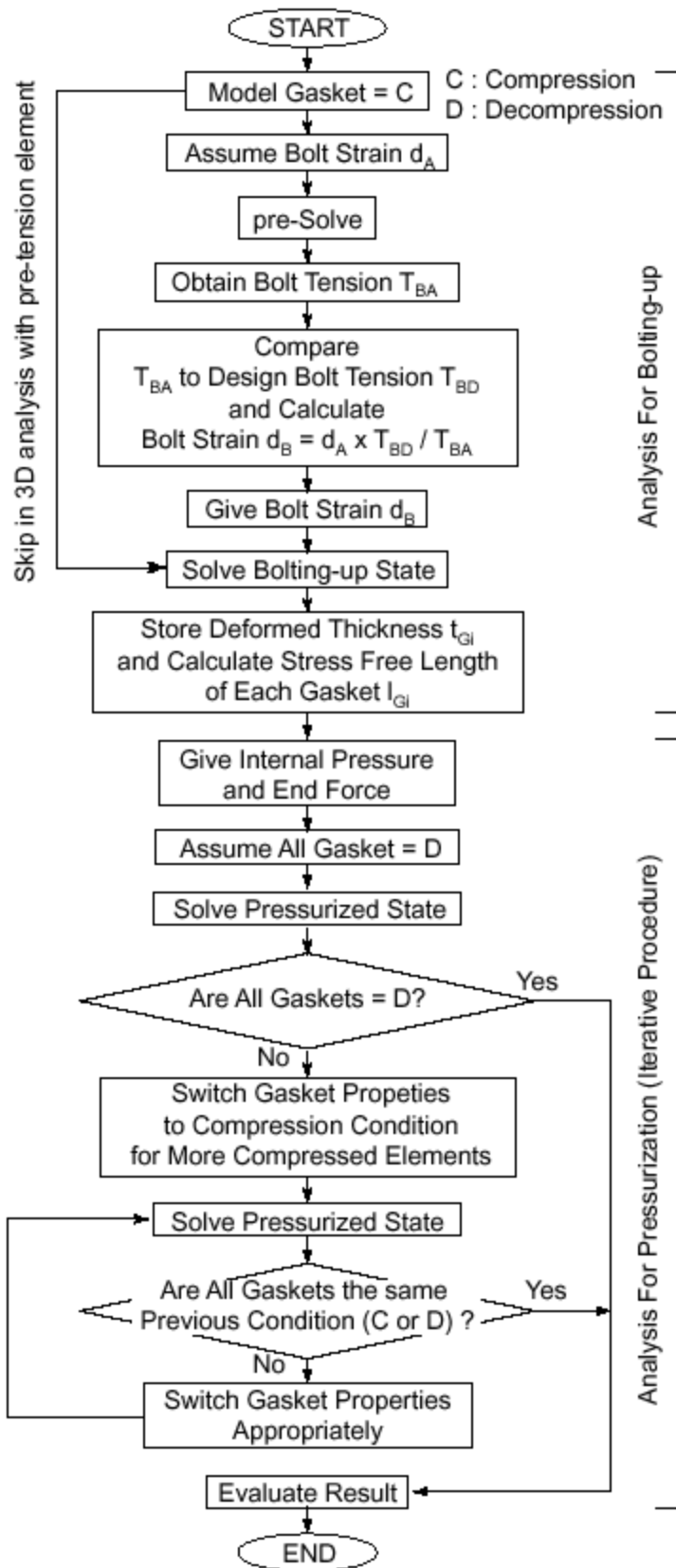
Here, although 5.6 provides special pre-tension elements which are used in 3-dimensional analysis, normal bar elements may be used for modeling of bolts. In axisymmetric analysis this bar element modeling is employed. In this bar element modeling, the bolt approximation depends on how high accuracy is needed, whether one bar may represent one bolt or several bars describe one bolt to accommodate stress distribution in the bolt.

The method for the analysis consists of basically two phases, which are bolting-up analysis and pressurization analysis. First, the gasketed flanges under the bolting-up condition is analyzed to obtain the initial condition of the flanges and gaskets. Next, the gasketed flanges under pressurized condition is analyzed considering pressure and hydrostatic end force acting to the shell. In this analysis, flange rotation may occur, if the rigidity of the flanges are not sufficiently large, then non-uniform stress appears on the gasket. If the rotation is rather large, the gasket detaches from the flange surface at the inner side, or may be more compressed at the outer side. The method simulates these behaviors of the flanges and gasket.

Figure 2 shows the overall flow diagram of the analytical procedure. Examination of the method is described in the following sections.

In the 3-dimensional model, only half of a bolt pitch is modeled considering symmetry, and radial and circumferential stress distribution on gasket as well as other parts is calculated. Using these models, analyses in 2-dimension and 3-dimension were carried out and compared. The results are described in later sections. For the 3-dimensional analysis, the effect of bolt holes on the FE model was investigated.

## **Figure 2. ANALYSIS FLOW**



Analysis under initial Bolting-up condition

Bolting-up load is normally given by designer, and currently this is calculated in accordance with a design code, mainly with ASME B&PV Code Section VIII Division 1 Appendix 2. Initial strains which are equivalent to the bolt load for the bar element representing bolts and initial compression of the gasket are calculated in this analysis. The analytical models of both 2-dimensional axisymmetric and 3-dimensional solid models are shown in Fig. 3 and Fig. 4. These models consist of solid elements ( PLANE82 for axisymmetric and SOLID185 for 3D ), bar elements ( or initial strain elements, LINK1 ) and gap elements ( or contact elements, CONTAC12 for axisymmetric and CONTAC52 for 3D ). As shown in the figures, axisymmetric solid elements are used to model the basic geometry of flanges and adjacent vessel shell or piping. As mentioned before, bolts are modeled using a bar element with initial strain. The gasket, which is the target of this analysis, is modeled using gap elements which accommodate initial gap and specified stiffness. These gap elements are connected to the corresponding nodes on flange surfaces. Gasket thickness is given by the length of the elements and it is initially the thickness of the virgin gasket. Stiffness of the gap elements is determined from the stress-strain curve of the gasket. Averaged stiffness is determined by the compression portion from zero deformation to the target deformation which is the designed thickness after bolting-up. The gasket stiffness, or modulus of elasticity, is represented by the slope of a stress-strain curve. The compression gasket stiffness and the decompression one are respectively defined to be constant which is drawn by dashed-lines in Fig. 1 in this analysis method. Here, the slope should be carefully defined considering the actual thickness change of the gasket because the slope affect considerably the result of the gasket stress and the distribution.

The gasket stiffness is converted to the spring constants of gap elements using a simple physics. To obtain the spring constant, the modulus of elasticity is multiplied by the area represented by a single gap element and divided by the element length (  $k = EA/l$ , where  $k$  is spring constant,  $E$  is Young's modulus,  $A$  is area and  $l$  is length. ).

The bolt is modeled using a bar element for the axisymmetric models and pre-tension elements ( PRETS179 ) for solid models.

For axisymmetric modeling, as the properties of the bar element is based on 360 degrees in ANSYS, modulus of elasticity and the whole bolt area are input in the model. When this analysis is carried out under the initial strain which is generated by the bar element of the bolts, the condition which is in balance among bolt tension, gasket compression and flange stress must be checked. Then we calculated the appropriate initial strain due to the bolts from this results. In order to obtain the appropriate initial strain, the model subjected to initial strain of unity is analyzed and the bolt load corresponding to the unity strain is calculated. Initial strain which generates the design clamping force given by a designer is derived from this analysis multiplying the ratio of design by the calculated forces.

In the 3-dimensional analysis, it is not necessary to calculate initial strain from the preliminary analysis, because the pre-tension elements deal with the pre-load and corresponding initial strain. After the pre-load analysis, the calculated initial strain can be used for the analysis under pressurized condition. The initial strain calculation was skipped in the flow chart.

Nuts as well as the bolt are modeled in 3-dimensional analysis and the contact between the nut face and the flange is considered using a contact pair of TARGE170 and CONTA173 with fully bonded option.

### Analysis under the pressurized condition

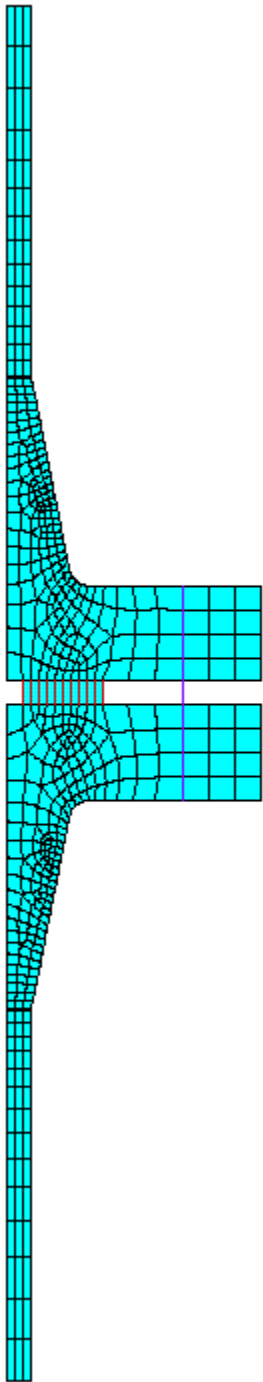
Analysis is carried out in the next step for gasketed flanges subject to internal pressure with the initial strain of the bolt which was derived from the previous analysis. In this analysis, depending on the stiffness of the flanges which results in flange rotation, all gasket area may be decompressed or some portion of the gasket may be compressed more while other portion is decompressed. In either cases, the modulus of elasticity for the decompression part in the stress-strain curve is applied for decompressed gasket portion, and the modulus of the compressed part of the same curve is used for more-compressed portion of the gasket. As it is very difficult to estimate whether the specific portion of the gasket is compressed or decompressed in a deterministic way, the following iterative approach is applied:

1. Calculate the stress-free length and deformed length of each decompressed gasket element from Fig. 1 based on the previous analysis.

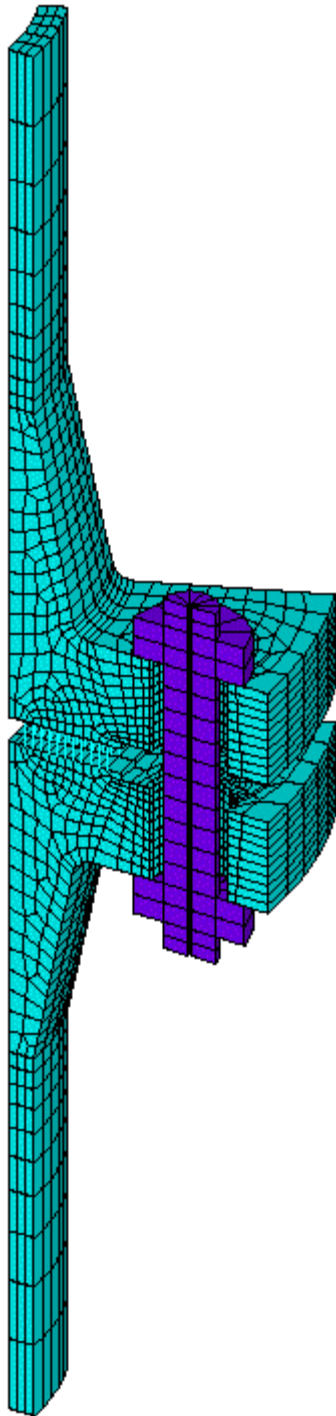
2. Assume all the gasket elements are decompressed at first.
3. Load internal pressure to the model and analyze it.
4. Check the result. If all the gasket elements are decompressed, this iterative method is finished as the gasket conditions is the same as assumed. If some elements are more compressed, change the modulus of elasticity of the decompressed by the compressed one, and change the length of the gasket by the initial length ( gasket thickness ). Proceed to the procedure (5) and go on to the next iteration.
5. Load internal pressure to the model and analyze it.
6. Check the result. If the compression/decompression states are the same as the states assumed before analysis, this iterative method is completed. If not, proceed to procedure (7) For example, if the outer two elements are "more-compressed" in the previous analysis and so assumed in this analysis and only the two outer elements compressed after this analysis iteration, the states are considered to be the same and reasonable. If it is true, the iteration is terminated.
7. Switch the initial condition and material property of the elements whose compression/decompression condition is different from their previous iteration, and return to procedure (5). In this procedure, element lengths calculated in procedure (1) and decompressing modulus of elasticity are input for decompressed elements, and the gasket thickness, which is initial length, and compressing modulus of elasticity are input for compressed elements.
8. Finally evaluate the results such as flange stress and gasket stress.

When all calculations are completed, reaction force for every gasket element is obtained. These reaction forces are converted to seat pressure of the gasket by dividing the corresponding gasket portion area. If the load is only internal pressure, all gasket elements are decompressed in most cases. This means that computational cost is much lower than the analysis with high nonlinear material elements such as gasket elements.

### **Figure 3. AXISYMMETRICAL ANALYSIS MODEL**



**Figure 4. 3-DIMENSIONAL ANALYSIS MODEL**



### ANALYTICAL OBJECT

The analysis was carried out for the standard flange of ASME. It is ASME 150 psi rated and 4 inch of nominal outer diameter with weld neck ( WN ) and raised face ( RF ). The specification of the flange is listed in Table 1. The material properties used in the analysis is on Table 2.

**Table 1. SPECIFICATION OF FLANGE**

--



ANSI 150 lb4B / 4B-SCH40	
FLANGE OD	229.0 mm
BORE	102.3 mm
LARGER HUB DIA.	135.0 mm
SMALLER HUB DIA.	114.3 mm
RAISED FACE DIA.	157.0 mm
FLANGE THICKNESS	23.9 mm
RAISED FACE	1.6 mm
FLANGE LENGTH	76.2 mm
BOLT CIRCLE DIA.	190.5 mm
BOLT ROOT DIA.	13.0 mm
NUT DIA.	27.0 mm
NUT THICKNESS	9.5 mm
NO. OF BOLTS	8 nos.
PIPE OD	114.3 mm
PIPE THICKNESS	6.0 mm
BOLT LOAD PER BOLT	13 kN
INTERNAL PRESSURE	1.0 MPa

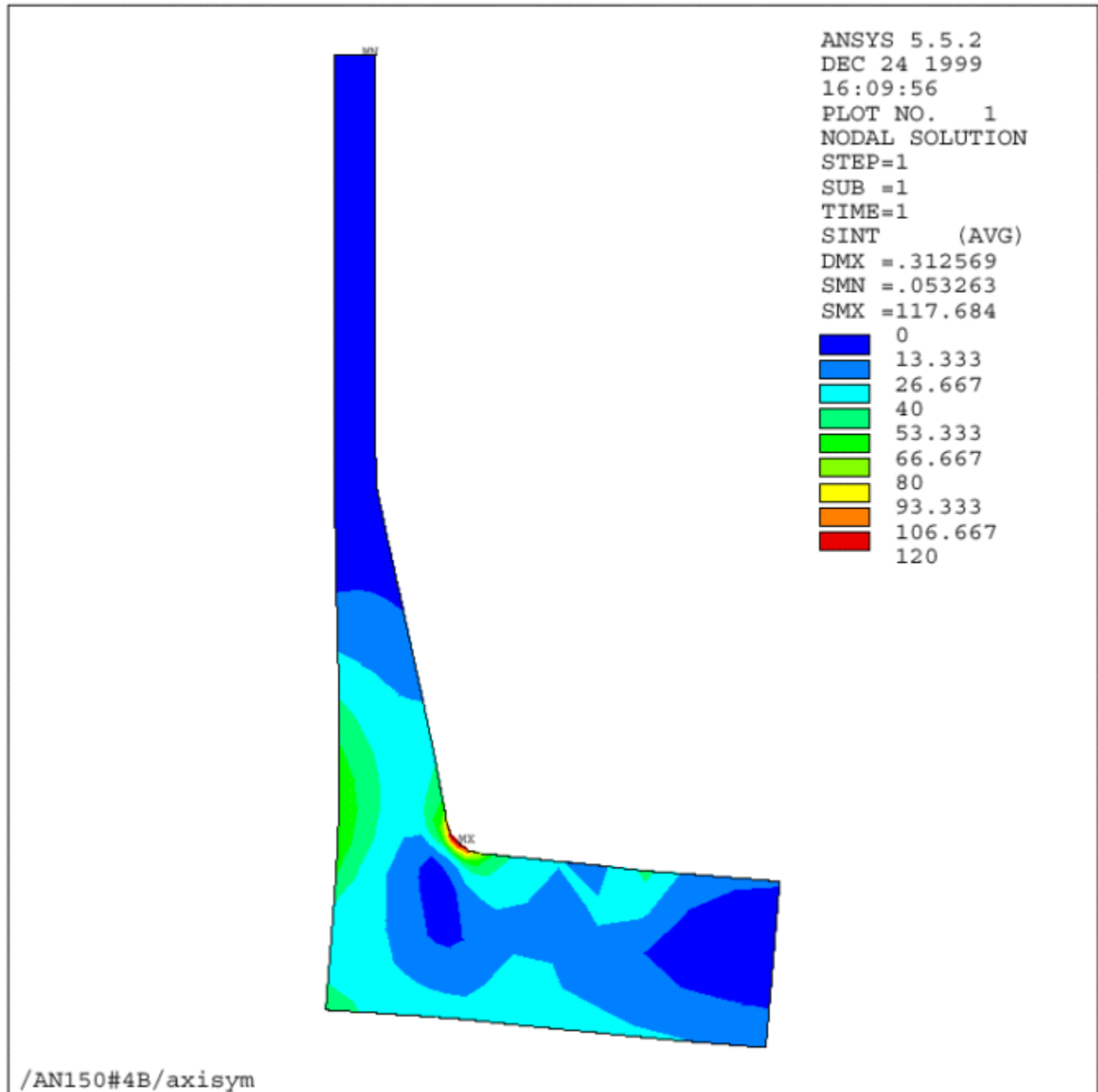
**Table 2. MATERIAL PROPERTIES**

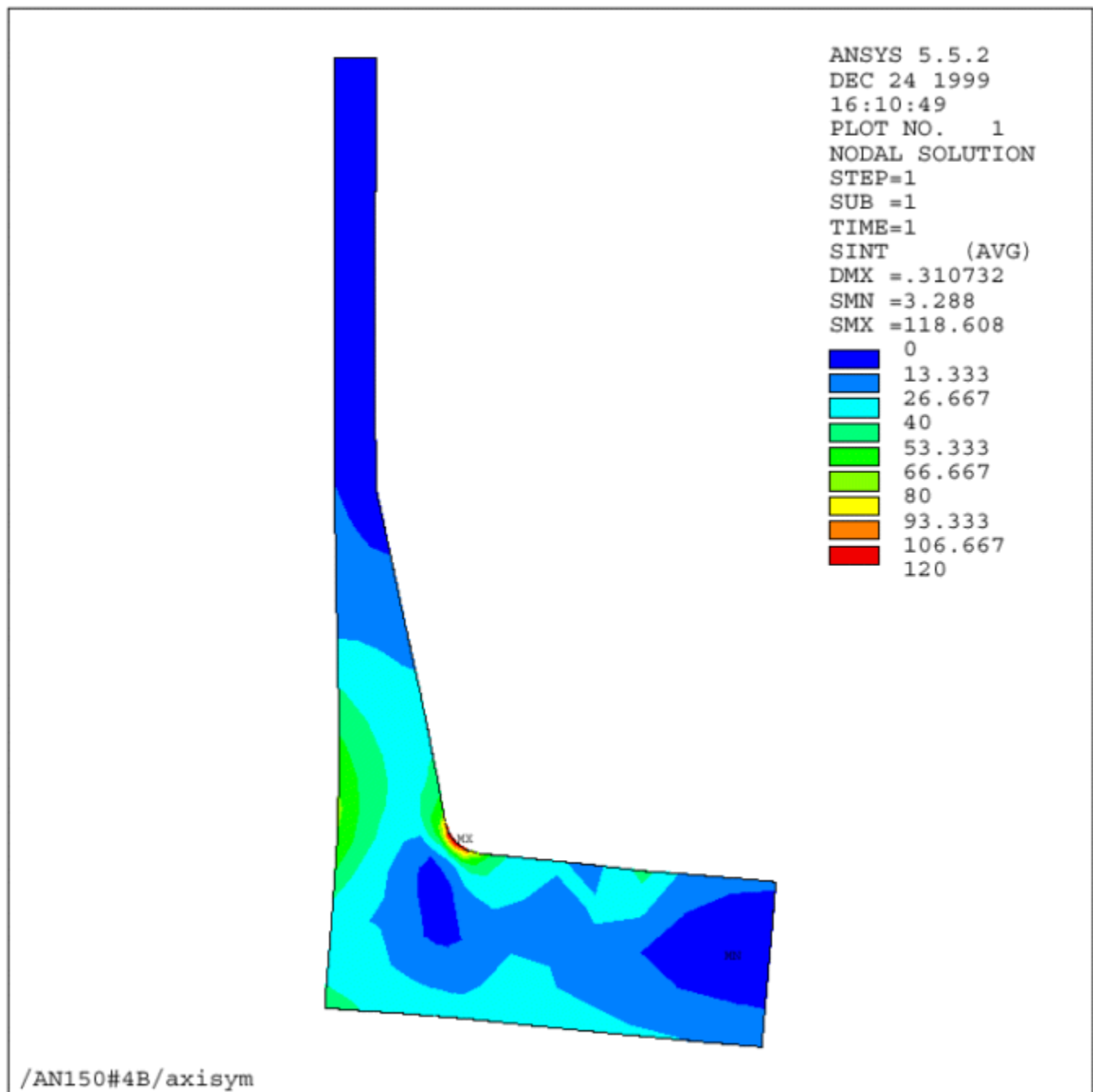
FLANGE ( SA105 )	
MODULUS OF ELASTICITY	200000 MPa
POISSON'S RATIO	0.3
BOLT ( SA193 B7)	
MODULUS OF ELASTICITY	200000 MPa
POISSON'S RATIO	0.3
GASKET	
COMPRESSION $E_C$	300 MPa
DECOMPRESSION $E_D$	6000 MPa

## ANALYSIS RESULTS

### 2-dimensional Axisymmetric Analysis

In this section a result of a 2-dimensional axisymmetric analysis is presented. The model is shown above ( Fig. 3 ). Stress intensity distribution in pre-loaded condition is shown in Fig. 5, and that in pressurized condition is shown in Fig. 6. These two figures have the same deformation scale ( /DSCA.,100 ), and flange rotation is observed larger in pressurized condition. In order to examine the stress distribution of the gasket, the gasket stress results are plotted in Fig. 7. They seem reasonable comparing with experiences.

**Figure 5. STRESS INTENSITY IN AXISYMMETRIC ANALYSIS UNDER BOLTING-UP CONDITION****Figure 6. STRESS INTENSITY IN AXISYMMETRIC ANALYSIS UNDER PRESSURIZED CONDITION**



### 3-dimensional Analysis

In the same manner, a result of a 3-dimensional analysis is described in this section. The model is shown above ( Fig. 4 ). Stress intensity distribution in pre-loaded condition and in pressurized condition are shown in Fig. 8 and 9, respectively. These two figures have the same deformation scale ( /DSCA,,100 ), and flange rotation is observed larger in pressurized condition. The gasket stress results are plotted in Fig. 10. They seem reasonable.

**Figure 7. GASKET STRESS DISTRIBUTION OBTAINED BY AXISYMMETRIC ANALYSIS**

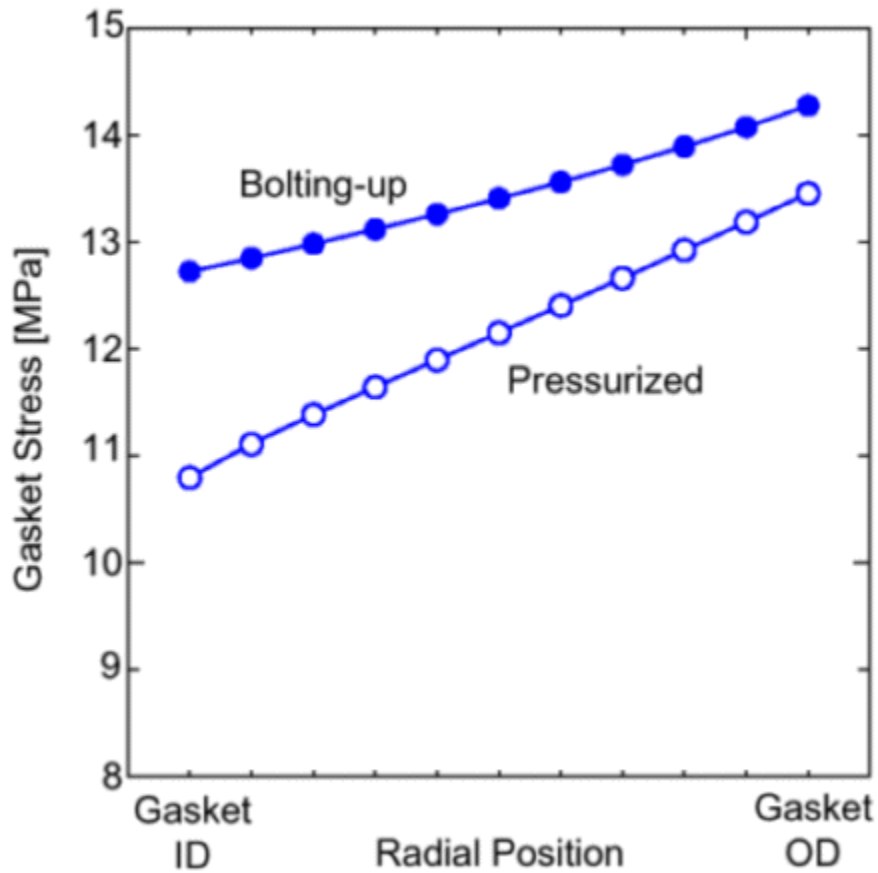


Figure 8. STRESS INTENSITY IN 3-DIMENSIONAL ANALYSIS UNDER BOLTING-UP CONDITION

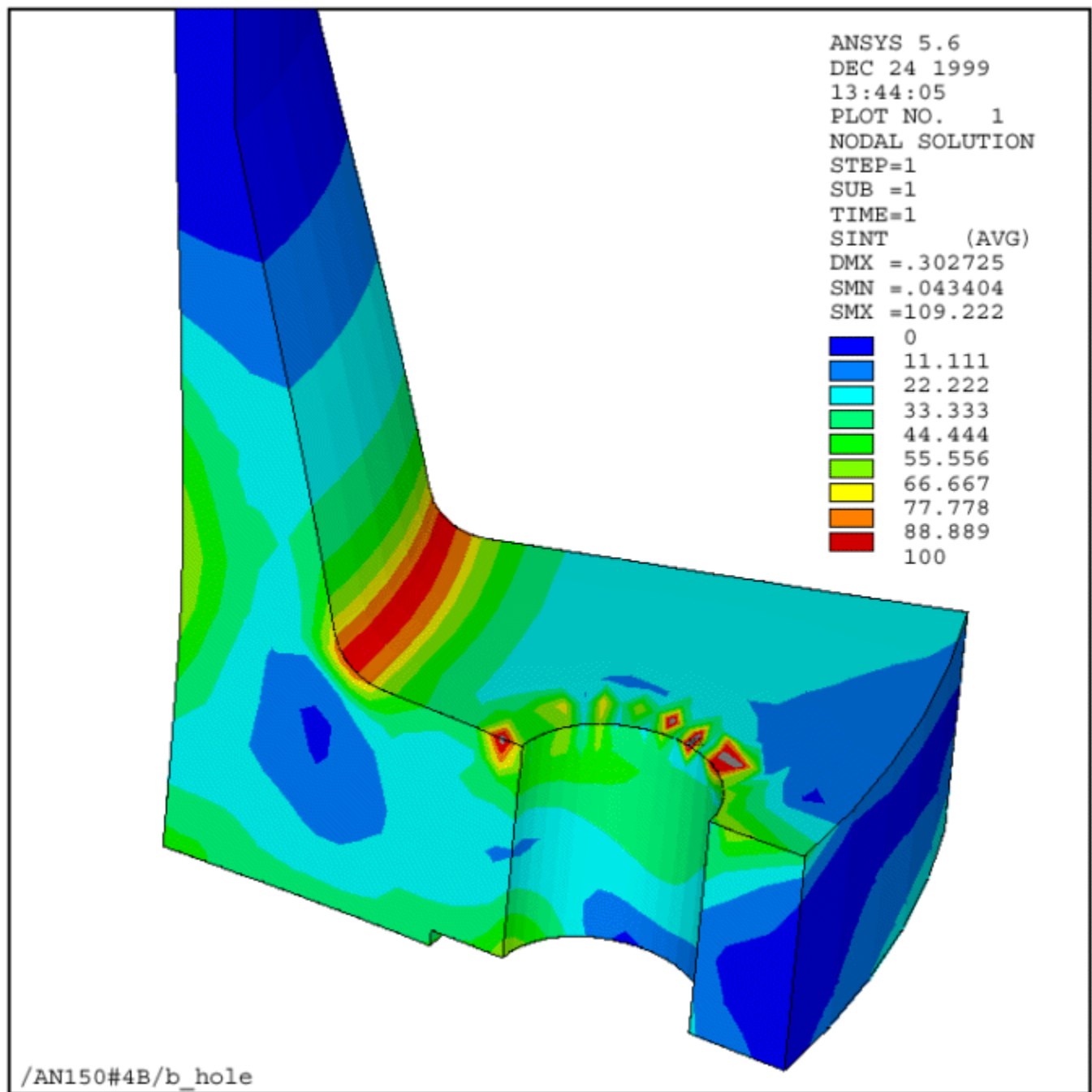


Figure 9. STRESS INTENSITY IN 3-DIMENSIONAL ANALYSIS UNDER PRESSURIZED CONDITION

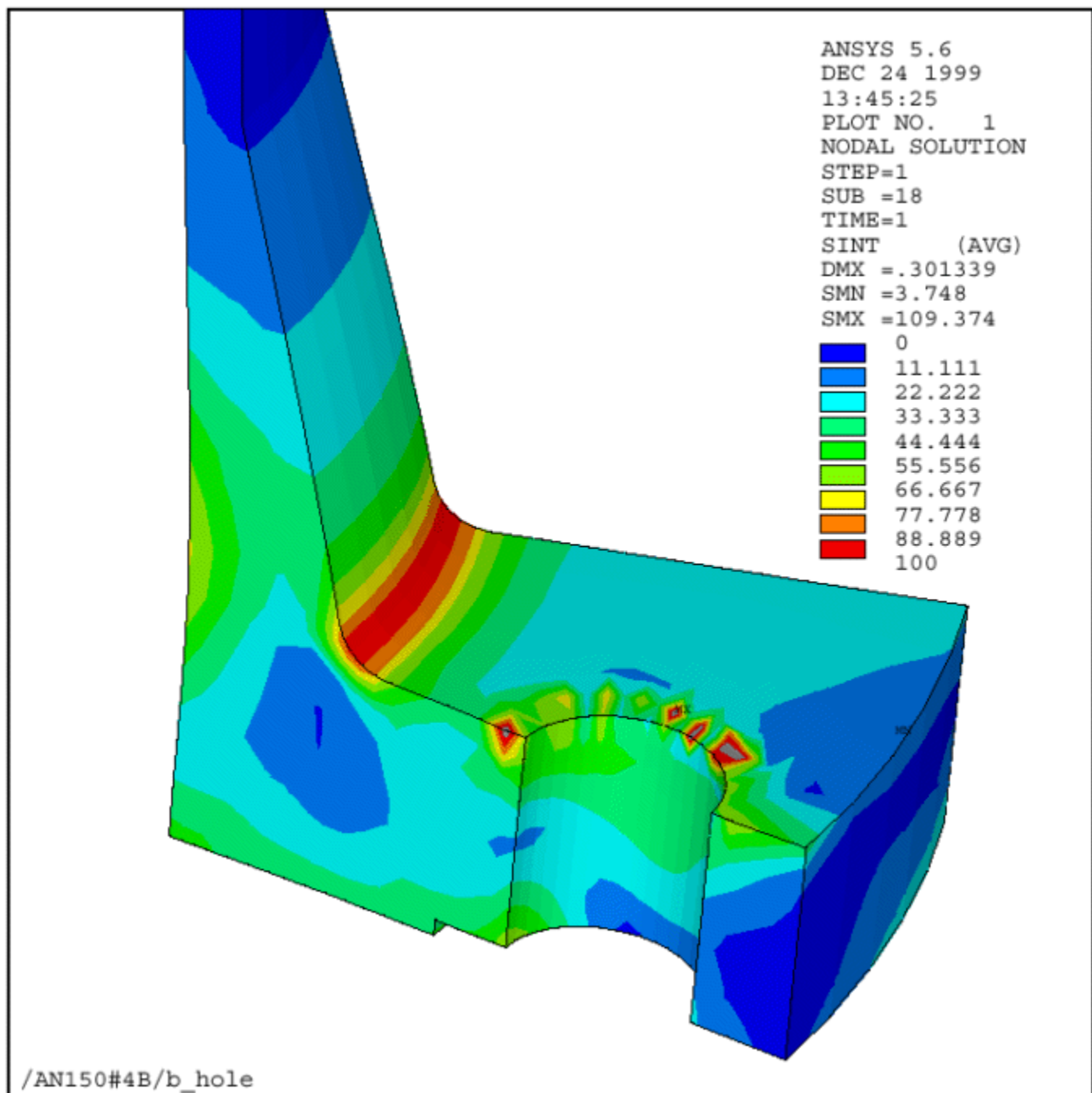
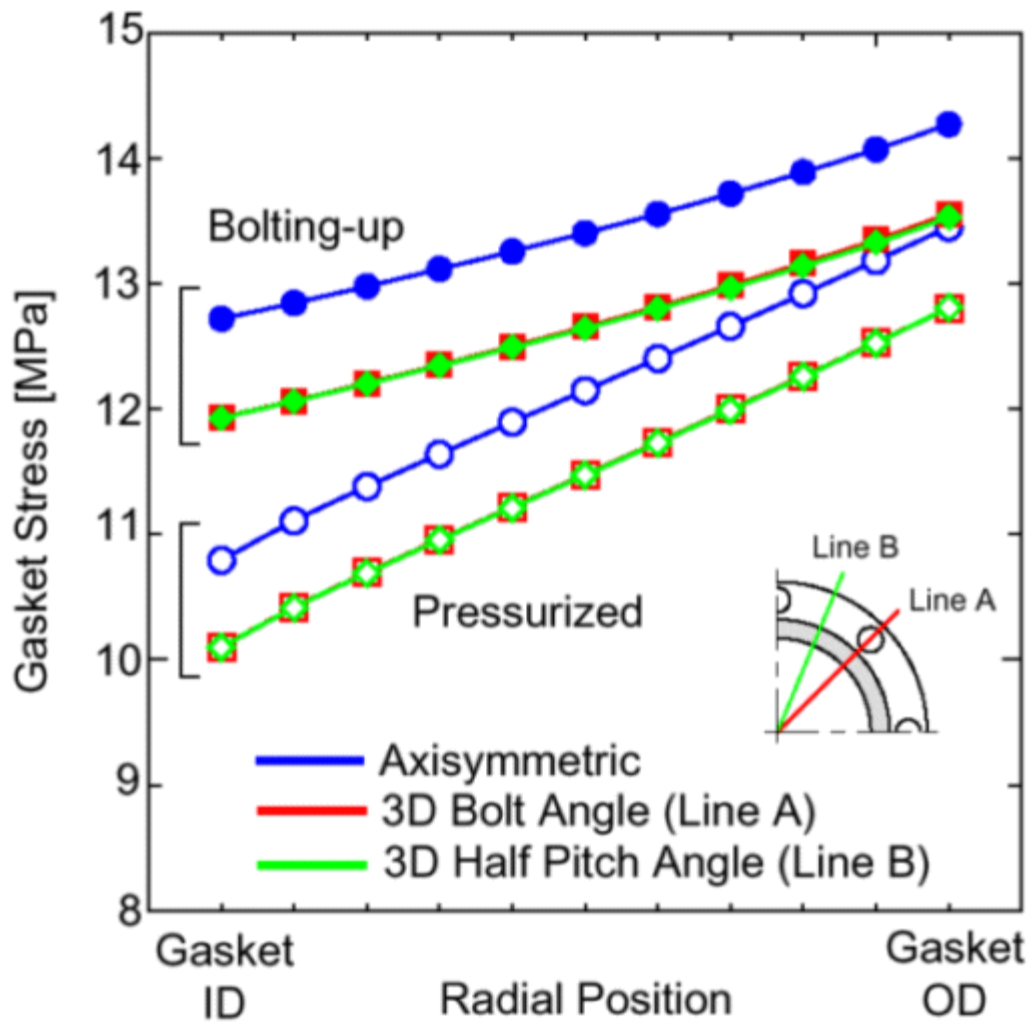


Figure 10. GASKET STRESS DISTRIBUTION OBTAINED BY 3-DIMENSIONAL ANALYSIS

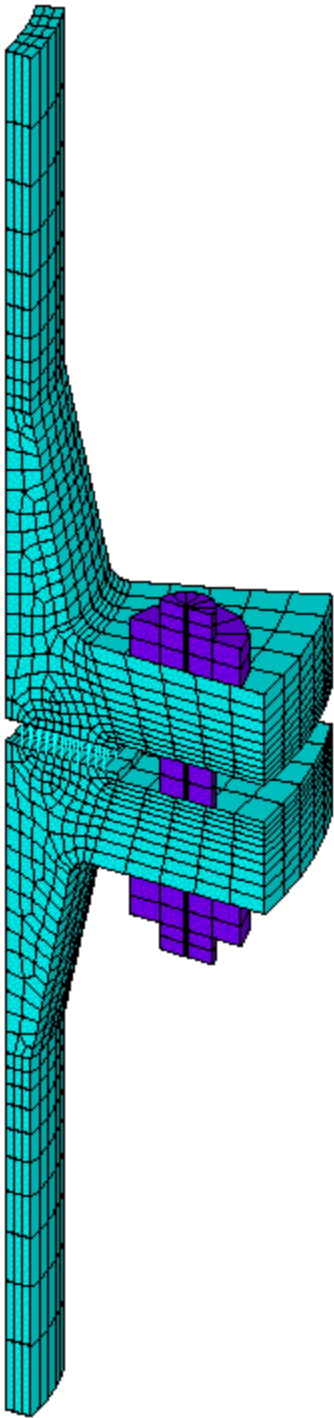


## DISCUSSION

### Effect of Bolt Holes

In this paper, bolt holes were modeled as they are. However, the modeling of holes takes a lot of work and it is questionable whether it is necessary or not in FEA. In FEA, it is not a problem even if bolts are installed through flanges without going through bolt holes. If the stress distribution is the similar in both cases with and without bolt holes, there is no need to model bolt holes. In this section, the effect of bolt holes are discussed.

Figure 11. 3-DIMENSIONAL ANALYSIS MODEL WITHOUT BOLT HOLE



The result of the stress intensity of the flanges without bolt holes are shown in Fig. 12 under pressurized condition. Figure 13 shows the comparison of the gasket pressure with and without bolt holes. It is observed that both results are nearly the same. It means that bolt holes are not necessary to be modeled in the analysis.

**Figure 12. STRESS INTENSITY IN 3-DIMENSIONAL ANALYSIS ( WITHOUT BOLT HOLE ) UNDER PRESSURIZED CONDITION**



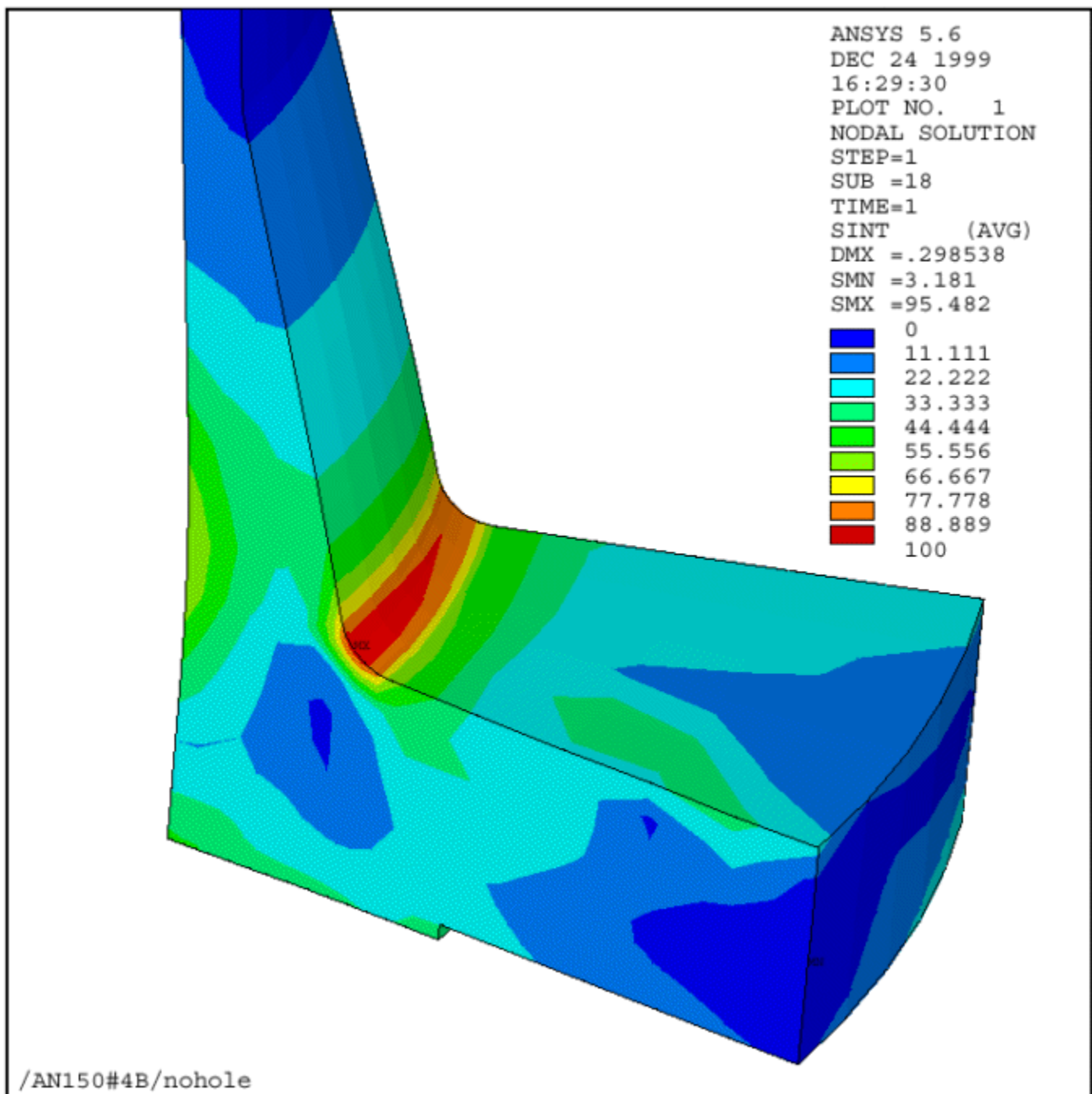
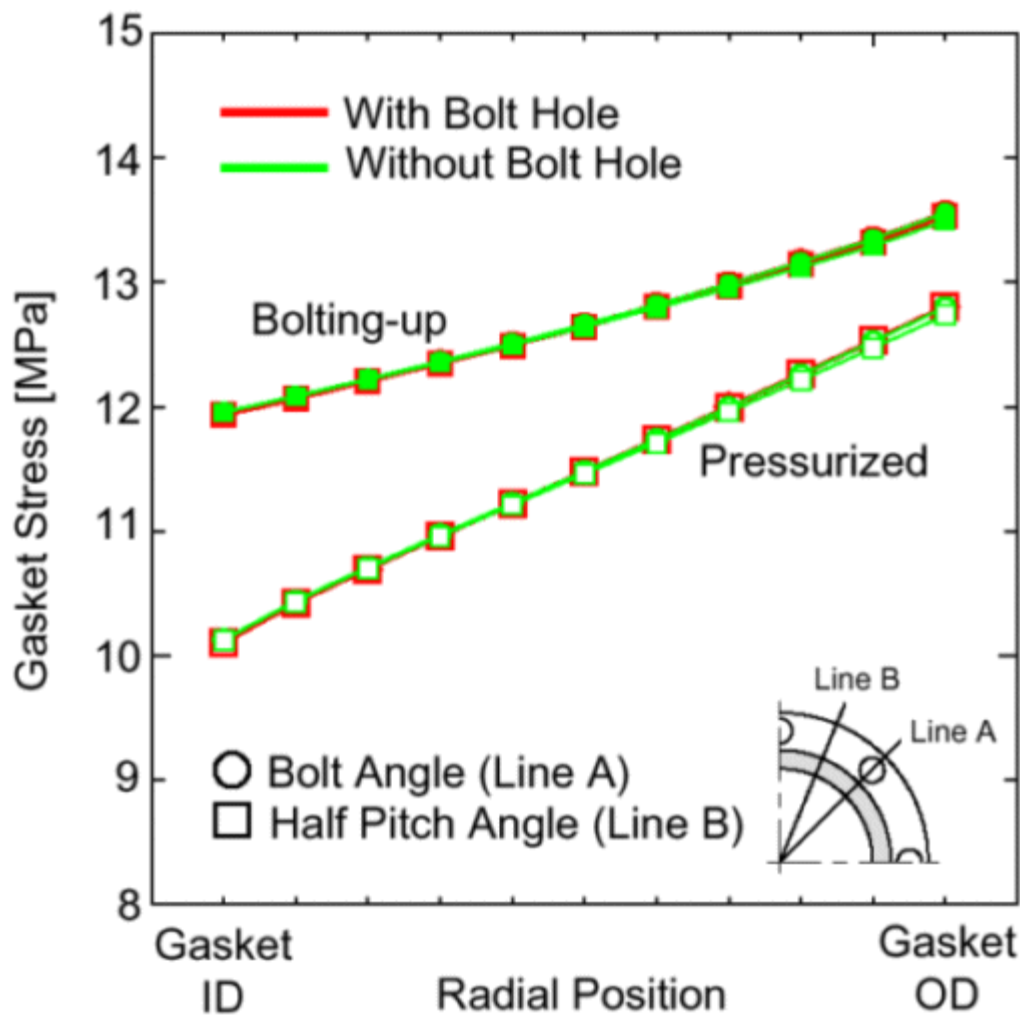


Figure 13. EFFECT OF BOLT HOLE TO GASKET STRESS DISTRIBUTION



## REFERENCES

1. HKS (Hibbit, Karlsson & Sorensen, Inc.), 1999, ABAQUS/CAE User's Manual
2. ANSYS Inc., 1999a, Analysis Guide ( On Line )
3. ANSYS Inc., 1999b, Command Manual ( On Line )
4. ASME (American Society of Mechanical Engineers), 1999, Boiler and Pressure Vessel Code Section VIII Division 1
5. Bickford, 1998, "Gasket and Gasketed Joints", Marcel Dekker Inc.
6. Sawa, T., Higurashi, N. and Akagawa, H., 1991, "A stress Analysis of Pipe Flange Connections", Journal of Pressure Vessel Technology, ASME, Vol. 113, pp 497-503
7. Bouzid, A. and Derenne M., 1999, "A Simple Method for Analyzing the Contact Stress in Bolted Flange Joints with Non-linear Gaskets", Proceedings of ASME PVP99, Vol. 382, pp.103-111
8. Shoji, Y. and Nagata S., 1999, "A New Analysis Method for Flange-Gasket Systems" , Proceedings of

ASME PVP99, Vol. 382, pp.113-120

9. Fukuoka, T. and Takaki, T., 1999, "Evaluations of Bolt-up Sequence of Pipe Flange Using Three-Dimensional Finite Element Analysis", Proceedings of ASME PVP99, Vol. 382, pp.87-93