Proceedings of GT2005
ASME Turbo Expo 2005: Power for Land, Sea and Air
June 6-9, 2005, Reno-Tahoe, Nevada, USA

GT2005-68434

INVESTIGATION OF DESIGN PARAMETERS AND FAILURE CRITERIA OF O-RING SEAL STRUCTURE

Dianyin Hu
School of Jet Propulsion, Beijing University of Aeronautics & Astronautics, Beijing, 100083, China

Rongqiao Wang
School of Jet Propulsion, Beijing University of Aeronautics & Astronautics, Beijing, 100083, China

Quanbin Ren
Shanxi Power Machinery Institute, Xi'an, 710000, China

Jie Hong
School of Jet Propulsion, Beijing University of Aeronautics & Astronautics, Beijing, 100083, China

ABSTRACT

First, this paper established the seal structural 2D axisymmetric model of a certain Solid Rocket Booster (SRB) and calculated the deformation and stresses at ignition through a large displacement, incompressible, contact finite element analysis. The results show that the maximum contact stress appears at the contact area and the maximum shear stress at groove notch. Then, some typical parameters of the seal structure which might have the impact on the sealing performance, such as the gap breadth, initial compressibility, fillets of the groove notch and bottom, groove width, were analyzed. We can find that the gap breadth and initial compressibility do great contributions to the maximum contact normal stress, and the groove notch and bottom fillets act upon the maximum shear stress obviously. In order to verify the validity of the 2D axisymmetric model, 3D structural finite element analysis of the structure was conducted, and the results indicate that in service, the upper flange is inclined relative to the nether flange, which seems to mean that the gap breadth can not be considered as a constant during the 2D axisymmetric analysis. However further calculations say that if using the minimum gap breadth gotten in 3D analysis as its constant gap value, the above 2D axisymmetric model can rationally take the place of 3D model to analyze the sealing performance. Finally, the failure modes & criteria of the O-ring seals based on the maximum contact normal stress and shear stress were determined to ensure the reliability of this structure.

Keywords: O-ring seal; finite element analysis; stresses; large deformation; failure criteria; contact

INTRODUCTION

There is a high requirement for sealing characteristics and reliabilities of O-ring seal structure in solid rocket engine. The O-rings' design greatly influences the whole engine's operation. Unreasonable design can cause seal failure, then the gas in combustion will leak out. Under such conditions, the general result is that the working pressure curves turn worse and the engine loses control and even explodes. Therefore it is essential to deeply investigate the O-ring seal structure of solid rocket booster (SRB). In this paper, the finite element (FE) model was established to calculate the stresses during the service process. Then the parameters influencing sealing performance were studied to optimize O-ring seal structure. The 2D axisymmetric and 3D structural analyses were conducted respectively and the results of 2D structure were compared with those of 3D. Meanwhile, to ensure structural reliability, sealing failure modes & criteria were obtained based on the maximum contact normal stress and shear stress

2D AXISYMMETRIC CALCULATION MODEL

Figure 1 shows the cross-section sketch of the O-ring seal structure, in which the parameters' meanings are: r means groove notch fillet radius, R groove bottom chamfer radius, δ represents the breadth of the gap between upper and nether flanges, b contact width, $b1$ the cross-section width after the compression, B is the groove width, and H denotes the groove depth.

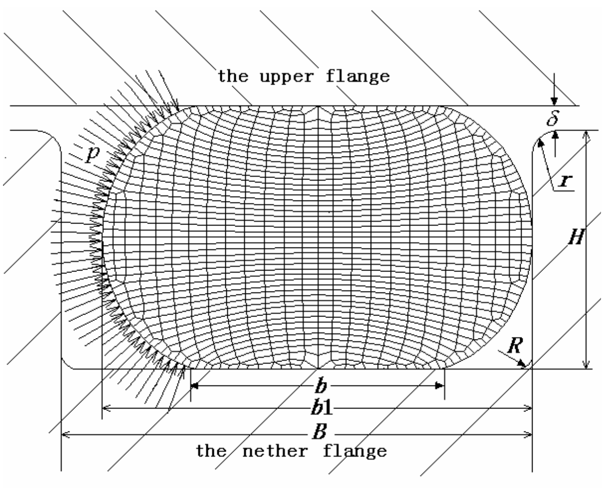

Fig.1 Sketch of O-ring seal structure

Basic hypotheses

To simplify the analysis, it is rational that all the 2D calculations are conducted based on the following three basic hypotheses:

(1) The upper and nether flanges are considered as rigid boundaries since their Young's modulus is several ten thousand times larger than that of the seal material, rubber. (2) The O-ring material is incompressible. (3) O-ring and its contact boundaries are approximately considered as axisymmetric.

Finite element model

The O-ring material is silicon rubber with Young's modulus, E=10Mpa. The maximum service pressure, p, is 10Mpa. The initial structural parameters of the seal structure are chosen as: O-ring diameter D=7mm, groove depth H=4.7mm, groove width B=9.1mm, notch fillet radius r=0.5mm, and groove wall friction coefficient f=0.1. The Neo-Hookean's constitution equation[1] is applied in which the function of strain energy density is stated as $w=C_1(I_1-3)$, and the stress as $\sigma = 2C_1(\lambda - \lambda^{-2})$, where, C_1 is Neo-Hookean constant, I_1 is the first strain invariant and the principal extension ratio λ equals $1+\varepsilon$ (ε represents the engineering strain). Then the true strain can be expressed as $e = \ln(1+\varepsilon)$ and the true stress $s = \sigma(1+e)$.

Then a two-dimensional axisymmetric, large displacement, incompressible, contact finite element model (2D model) of the above structure was established with FE software MSC.Marc, as shown in Fig.1.

The tentative calculations showed that using common meshing technique, the deformation of O-ring in service was so large that some elements became badly distorted and the analysis would abnormally halt. To solve this problem, global remeshing technique was introduced on the up-half model (Fig.2). Limited by the capability of Marc's remeshing technique (8-noded quadric element is not convergent), 4-noded axisymmetric quadrilateral elements were employed to model the 2D seal structure, and then the calculation fidelity can be held by another approach, increasing the number of elements.

ANALYZING SEAL STRUCTURAL PARAMETERS

Figure 2 demonstrates the FEA results of deformation and Von Mises stress of O-ring structure in the ignition process. One of the interesting phenomena in Fig.2 is that part of O-ring was crushed into the gap between the upper and nether flange. As known, the necessary condition for sealing is that the maximum contact normal stress at the area where O-ring contacts with upper and nether flange is no less than maximum service pressure p . Meanwhile, experimental investigation shows that the O-ring fails to keep seal due to the large shear stress at the groove notch. Hence, the parameters influencing the maximum contact compressive stress $(\sigma_x)_{\max}$ (absolute value) and shear stress σ_{xy} near the groove notch fillet were mainly discussed hereafter.

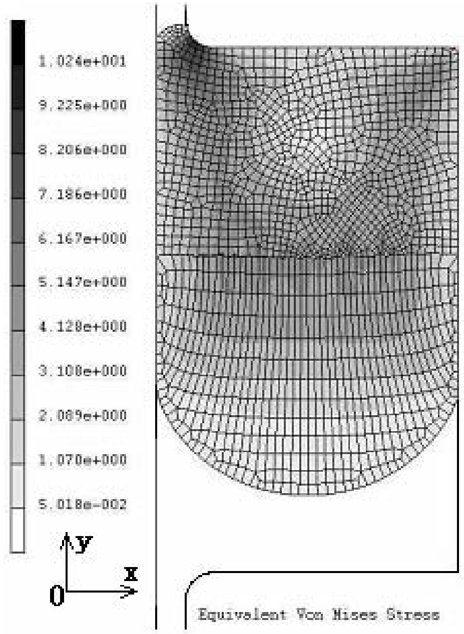

Fig.2 The deformation and Von Mises stress of O-ring in service (Mpa)

To choose an O-ring seal structure, one of the main factors considered is the appropriate compression influenced by the initial compressibility, the gap breadth between upper and nether flange and the O-ring seal's cross-section diameter etc. Meanwhile, the groove notch fillet, and the groove width also have great effects on the sealing performances.

The gap between upper and nether flange

It is observed from FEA calculations and experiment that increasing the breadth of gap between upper and nether flange, δ ,due to service pressure, causes an increase in the quantity of rubber crushed into the gap, influenced by O-ring's hardness (Material's hardness is in logarithmic correlation with Young's modulus E) as well. Hence, critical crushed curves (Here, the crushed standard is considered that the position of crushed rubber is on the groove notch fillet center) for the benchmark model with two materials (E=10Mpa and E=6.03Mpa) are plotted in Figure 3. The area below each curve represents no extrusion and the above with some extrusion. Two noticeable conclusions are drawn from Figure 3: To avoid the extrusion, for the same material, the breadth of δ should be smaller when pressure is bigger; for the same p, the breadth of δ should be smaller when Young's modulus E is less. Thus, the maximum δ without extrusion can be gotten and the material of large hardness is recommended to avoid seal failure.

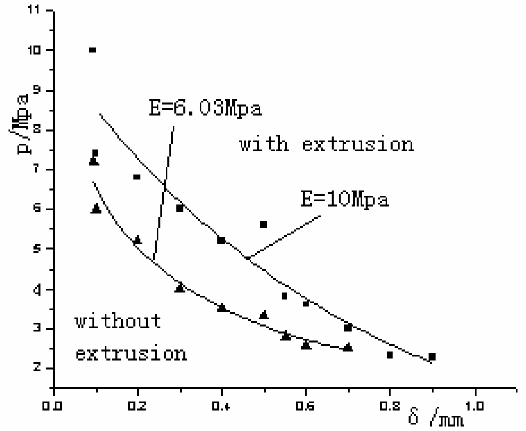

Fig.3 Critical crushed curves of two materials

The relation of δ with shear stress at groove notch and maximum contact stress derived from FEA (as shown in Fig.4), indicates the bigger δ is, the larger shear stress and the smaller maximum contact normal stress are. We can imagine when the breadth of gap is beyond a certain value, too much extrusion will probably cause the O-ring to destroy. Therefore appropriate δ must be considered during the design process.

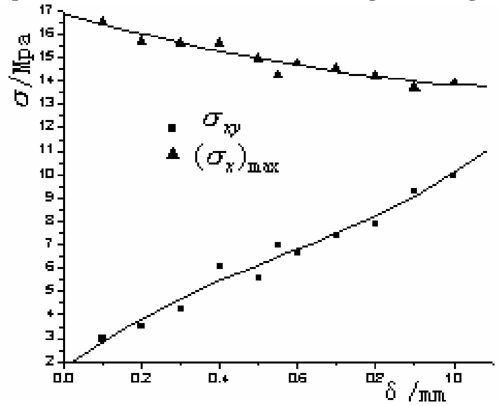

Fig.4 Analysis of stresses in terms of gap breadth

Initial compressibility

The initial compressibility, ε_0 is defined that the change of O-ring diameter relative to groove depth. It can be expressed as $\varepsilon_0 = (D - H)/D \times 100\%$.

The FEA results (see Fig. 5) show that with the increase of the initial compressibility the maximum contact normal stress enhances a great deal, while the shear stress mildly augments, which demonstrates that increasing the initial compressibility does a benefit to seal. Considering too large initial compression causes residual deformation[2], we should select the initial compressibility as small as possible under the condition of keeping seal.

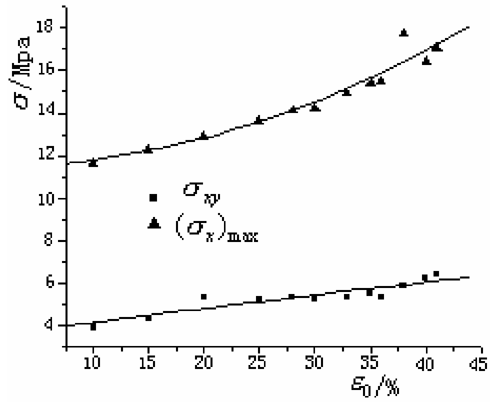

Fig.5 Curve of stress vs. initial compression

Groove notch fillet

As indicated in Figure 6 (point of which is gotten from FEA calculations), the groove notch fillet, r has no effect on the contact normal stress, while it influences the shear stress mainly.

If the fillets are unreasonably designed, seal failure will probably happen because of shear destruction in service. From Figure 6, we can see that if the groove notch fillet is less than a certain constant, the shear stress changes a lot. When the groove notch fillet radius gets to above *0.5mm* for this type engine, it becomes stable.

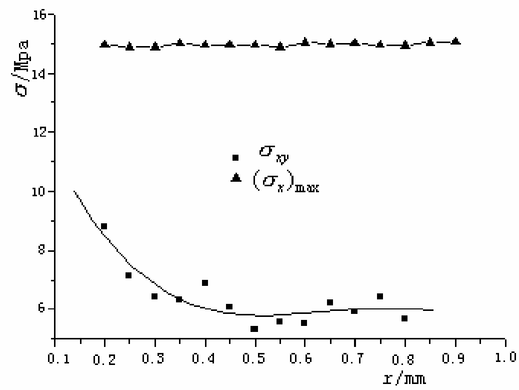

Fig.6 Influence of the groove notch on stresses

Groove width

The groove width, *B* influences the modes of pressure-load superimposed on the O-ring structure, then affects the sealing performances (The possible pressure-modes are indicated in Fig.7). The results of the maximum contact stress at the key position are shown in Table 1. (Where in case 1a and case 1b, the groove width *B*=9.1mm; in case 2a and case 2b, *B*=7.9mm.)

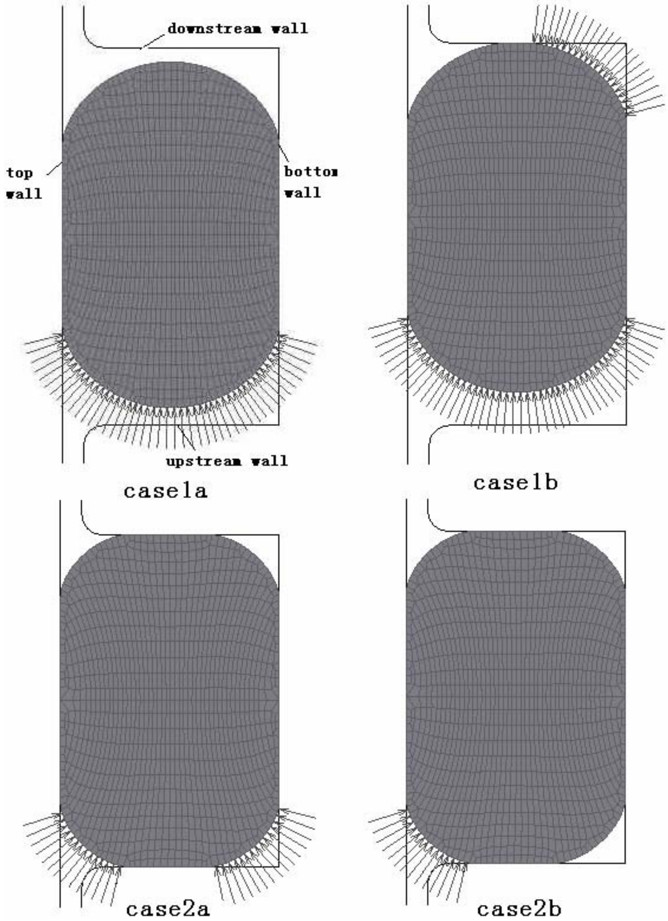

Fig. 7 Sketches of various types of pressure applied in various groove width

Table1a Results of O-ring contact stress with unconstrained O-ring wide groove

p/Mpa	Maximum contact stress against walls/Mpa					
	Case1a			Case1b		
	T	B	DS	T	B	DS
0	4.81	4.80	0	4.81	4.80	0
5	10.21	9.87	6.29	9.85	9.59	9.04

Table1b Results of O-ring contact stress with constrained O-ring narrow groove

p/Mpa	Maximum contact stress against walls/Mpa						
	Case2a			Case2b			
	T	B	DS	T	B	DS	US
0	5.72	5.68	3.84	5.72	5.68	3.84	4.67
5	9.11	9.14	6.28	7.48	8.03	5.66	6.16

In Table 1 T, B, DS and US represent the top wall, bottom wall, downstream wall and upstream wall respectively. The case 2b structure (narrow groove and pressurization with single side) weakens the maximum contact normal stresses against the walls. Thus O-ring will probably deviate from the contact surface resulting in seal failure when the work pressure reaches a certain value. For case 2a (narrow groove with double sides pressed), although the maximum contact normal stress has a little increase, there is no sufficient room for constrained O-ring seal to freely roll, which causes large friction force and serious wear, even crack. For case 1b (wide groove under pressure from double sides) the quantity of crushed rubber is more than that of case 1a (structure with single side pressurization) in service, which is unfavorable.

Through comparing the stresses in Table 1, it shows that the best structure is case1a and the groove width *B* should be no less than O-ring cross-section width *b*1, which is the reason that wide groove is recommended. However, too wide groove causing O-ring to roll in large scale and destroy due to too much room in the groove, is not suitable. By the way, the most dangerous seal structure case2b must be considered.

Besides the typical parameters above, material properties, O-ring cross-section diameter[3], friction coefficient *f* between O-ring and walls, groove depth *H*, contact width *b* and cross-section width *b*1[4] influence sealing characteristics as well. The FEA calculations show that: It is important to use large hardness material under the high-pressure condition, otherwise there are small contact normal stress and large extrusion, which do harm to the seal; The O-ring with big cross-section is advantageous because it can offer large contact width for the same compression when it satisfies structural arts and crafts; If groove wall friction *f* is less than 0.2, the O-ring will rotate. Meanwhile, *f* =0.2 is the transition point at which the maximum stresses occur; Since groove depth relates to the initial compressibility, we do not discuss it; The relationship between contact width and initial compressibility obtained from FE results is shown in Figure 8, where other empirical formulations are referred to [5] and [6]. The curve of this paper had a good agreement with the other formulations. According to these analyses, a better seal structure can be designed.

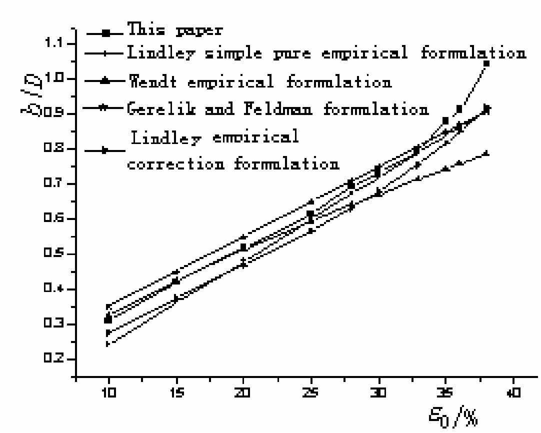

Fig. 8 Comparison of *b*/*D* value

THREE DIMENSIONAL EFFECT ON 2D MODEL

The two-dimensional axisymmetric model was calculated without considering the nonuniformitiy in circumferential direction due to tightening the bolts, and neglecting the uneven gap between upper and nether flange because of axial-displacement of flanges, so it is necessary to conduct real structural analysis to comprehend three dimensional effect through comparing the results with those of 2D model. Then three-dimensional FEA of a certain SRB (FEA model was shown in Fig.9, where the upper flange was linked with the nether by 40 equidistant bolts in the circumferential direction, Z denotes the axial direction) was carried out in order to verify and correct the results obtained from two-dimensional axisymmetric structure.

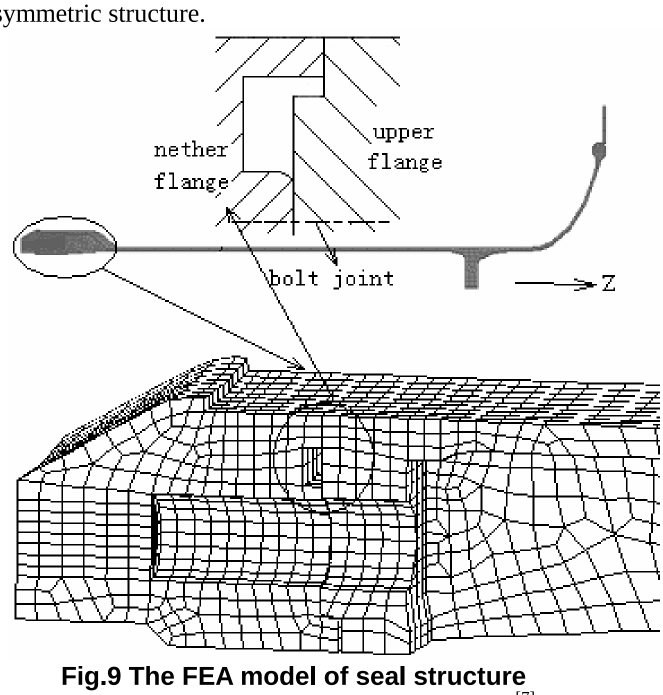

Fig.9 The FEA model of seal structure

Referring to the handbook of mechanics design[7], we can get that the bolt's deformation Δl_1, caused by the pre-tightening force is 0.02106mm, and the one by work pressure (the maximum pressure *p*=10Mpa) is $\Delta l_2 = 0.0402mm$. The expression $\Delta l_1 < \Delta l_2$ means that a gap between upper and nether flange will arise because of the service pressure.

Three-dimensional flanges model established assumed that the bolt was gluing with the nether flange and contacted with the upper one. The corresponding flange deformation was regarded as the seal structural boundary conditions. One-fortieth model was built and treated as cyclic symmetry to obtain the flanges' axial-displacement when the maximum pressure *p* reached 10Mpa. (As shown in Fig.10, 11, and 12)

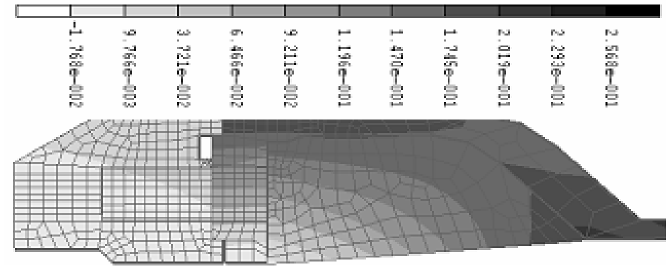

Fig.10 Axial-displacement of the upper and nether flanges at *p*=10Mpa

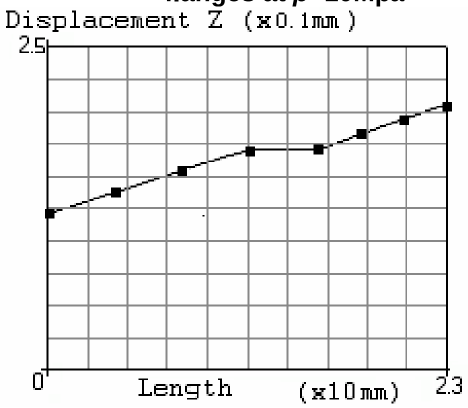

Fig.11a Axial-displacement of upper flange

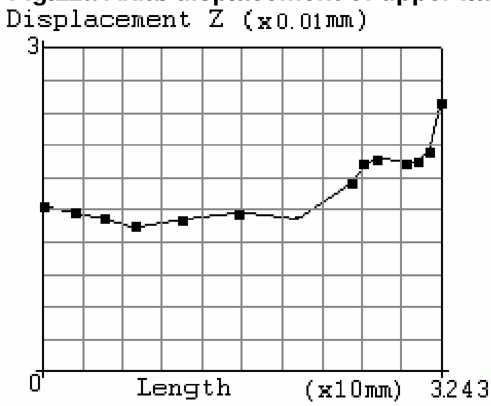

Fig.11b Axial-displacement of nether flange

In Figure 11 the abscissa means the relative distance of the nodes away from the bolt orifice center. Figure11a and Figure 11b show both axial-displacements of the upper and nether flange at the groove cross-section are small. We can see from Figure 11a that the axial-displacements of nodes on upper flange along the groove cross-section are unequal: the displacements become a bit larger when the nodes are away from the bolt, so a slope is generated (that is to say the gap δ is not a constant), which we called uneven gap.

In Figure 12 the abscissa means the relative distance of the nodes between two bolts center bordering upon each other in the circumferential direction. The maximum deformation in the circumferential direction, 0.0068mm is very small compared with the maximum upper flange's axial-displacement, 0.2568mm. Furthermore, for the rubber-like materials, this deformation has little effect on the whole structure. Therefore three-dimensional influence upon sealing performance can be regarded as that the effect manifests as the emergence of the uneven gap neglecting circumferential deformation.

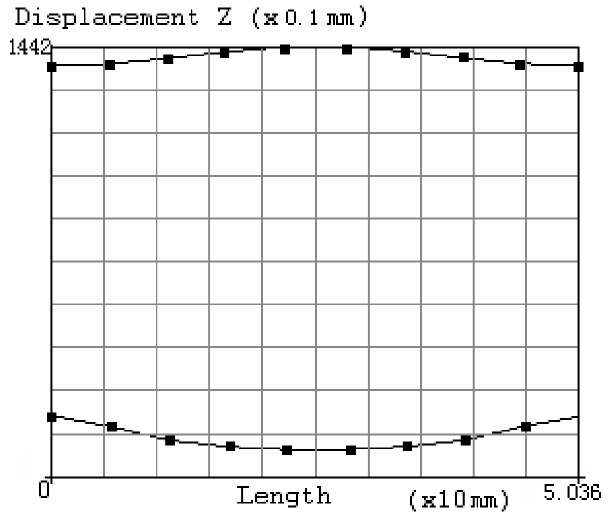

Fig. 12 Circumferential changes of axial-displacement of upper and nether flange

However, the uneven gap may be considered in 2D model to reduce the workload greatly. Pseudo-three dimensional (pseudo-3D) model is presented to calculate the actual seal structural deformation and stresses, since it is described by two-dimensional axisymmetric structure with uneven gap between upper and nether flange considering the three-dimensional effect. (See Fig.13 where δ_{min} means the axial-displacement in the bolt's center circle)

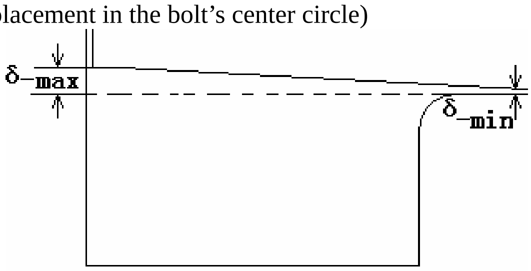

Fig.13 Sketch of the uneven gap between upper and nether flange

With the conclusion stated above, pseudo-3D model is established using the corresponding flanges axial-displacement as the uneven gap.

The FEA results (Fig.14, Fig.15) show after three-dimensional correction, the maximum shear stress also occurs at the groove notch fillet and the maximum contact normal stress happens at the contact area between upper and nether flange. Meanwhile, it is noted that the results of pseudo-3D and 2D axisymmetric models for the parameters influencing sealing characteristics are quite similar. Table 2 compares the stress calculations of the even gap with those of the uneven gap.

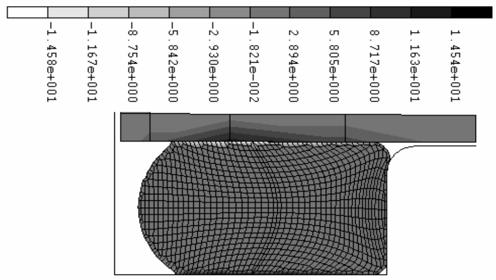

Fig.14 Contact normal stress at *p*=10Mpa (Mpa)

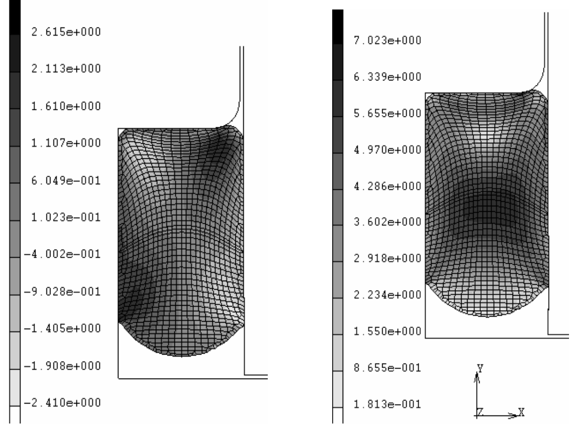

Fig.16 shear stress (left), Von Mises stress (right) at *p*=10Mpa (Mpa)

Table 2 Comparison of stresses for different models with even and uneven gap

Stress/ Mpa	Uneven gap	Even gap (δ_{max})		Even gap (δ_{min})	
			Ratio		Ratio
Maximum contact stress	14.58	14.30	-1.92%	14.83	1.71%
Shear stress at groove notch	2.615	2.317	-11.4%	2.528	-3.33%
Von Mises stress	7.023	5.746	-18.18%	6.865	-2.25%

In Table 2, δ_{max} and δ_{min} structure respectively mean two-dimensional axisymmetric model with the maximum and the minimum gap between upper and nether flange. It is shown from Table 2 that with the minimum gap structure, the stress variation-ratios are small with the bound of allowable error. So we refer that two-dimensional axisymmetric model with the minimum gap breadth obtained from 3D analysis can be used to analyze sealing performance in place of three-dimensional model based on which the O-ring seal structure can be optimized.

FAILURE MODES & CRITERIA

To ensure the safety and reliability of the seal structure, failure modes & criteria should be investigated based on above analyses. According to calculation and experiment, we know that during the service process, the shear stress at the groove notch is maximal, and the quantity of the rubber crushed into the gap is increasing according with the increase of the breadth of gap between the upper and nether flange, and the O-ring might be destroyed when the corresponding maximum circumferential tensile stress, is larger than allowable tensile strength of the material, about 5-7Mpa with reference to the material handbook. Meanwhile, seal failure caused by the maximum circumferential tensile stress, about 1Mpa for the O-ring in this paper, will not arise for the normally designed seal structure. Therefore, failure modes & criteria of the maximum contact normal stress and shear stress were established.

Maximum contact normal stress criteria

If the maximum contact normal stress between O-ring and the upper, nether flange is less than maximal service pressure *p* during the working process, there will be a leakage of gas. The maximum contact normal stress is the essential condition for seal failure criteria. To keep seal it requires

$$(\sigma_x)_{\max} \geq p \tag{1}$$

The maximum contact normal stress relates to O-ring materials, cross-section shape, compression and pressure, and it can be expressed as

$$(\sigma_x)_{\max} = \sigma_0 + kp \tag{2}$$

Where k is a linear proportional coefficient of p to $(\sigma_x)_{\max}$, and it is affected by Poisson's ratio γ, $0 < k \leq 1$; σ_0 is pre-contact stress, and it is a function[2] of initial compression ε_0 and friction coefficient f between O-ring and flanges. It can be written as

$$\sigma_0 = g(f, \varepsilon_0) E \varepsilon_0 \tag{3}$$

Combined equation (1), (2) and (3), we can get

$$g(f, \varepsilon_0)\varepsilon_0 \geq p/E\,(1-k) \tag{4}$$

If k is known by experiment or calculation, the minimum initial compressibility under the condition of keeping seal can be gotten from equation (5).

The curve of the maximum contact stress versus pressure for this type engine was calculated and shown in Figure 17. From figure 17 we knew that $\sigma_0 = 4.695 Mp_a$ and $k \approx 0.998$, and every point in the curve satisfied the equation (1).

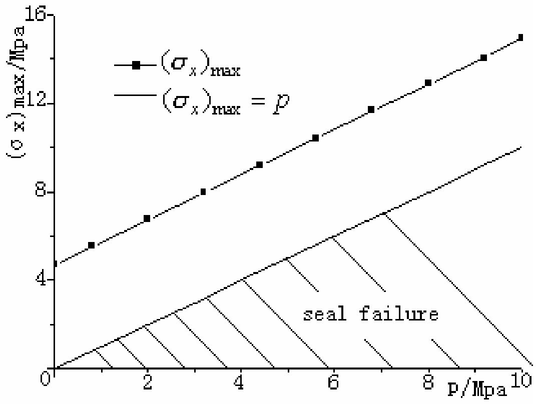

Fig.17 Curve of maximum contact stress versus pressure

Maximum shear stress criteria

The FEA results show that the maximum shear stress, about several Mpas, occurs at the groove notch where the O-ring is laniated, then another failure criteria can be stated as

$$\sigma_{xy} < [\tau_b] \tag{6}$$

Which means that the seal structure integrity can be kept if the shear stress gotten in FEA calculations, σ_{xy} does not exceed the allowable shear strength of the material, $[\tau_b]$.

CONCLUSIONS

(1) The FEA results show that the maximum contact stress appears at the contact area between upper and nether flange and the maximum shear stress at groove notch.

(2) The parameters influencing the sealing characteristics, such as the gap breadth between upper and nether flange, initial compressibility, the fillets of the groove notch and bottom, groove width were discussed. The calculation results show that the gap and initial compressibility do great contributions to the contact normal stress, and the groove notch and bottom fillets act upon the maximum shear stress greatly. The analyses are useful to optimize the seal structure.

(3) After the flanges' deformation having been calculated, O-ring stresses during the service process were gained with the pseudo-3D model. The conclusion is that two-dimensional axisymmetric model with the minimum gap breadth gotten in 3D calculations can be used to analyze sealing performance of SRB in place of the three-dimensional model.

(4) The failure modes & criteria of the O-ring seals based on the maximum contact normal stress and shear stress are presented to assure the sealing integrity and reliability of the O-ring structure.

(5) Relaxation, creep and bounce-back of O-ring seal structure due to the effects of temperature and time are to be considered in future study.

REFERENCES

[1] R.Metcalfe, S.B.Baset, R.Lesco and W.N.Selander. Modeling of Space Shuttle Solid Rocket O-rings, 12th International Conference on Fluid Sealing, 1989, pp.3-25.

[2]Chen Ruxun. Sealing for Solid Rocket Motor, Structure & Environment Engineering, 1995, pp.1-5. (in Chinese)

[3] Zuo Zhengxing and Liao Ridong. Finite Element Analysisi of Rubber Sealing Ring for Model 12150 Diesel Engine. Neiranji Gongcheng ,1996(2), pp.46-49.(in Chinese)

[4] Ren Quanbin. Deformation and Stress Analysis of Rubber O-ring. Journal of Aerospace Power, 1995(3), pp.241-244. (in Chinese)

[5] A F George, A Strozzi and JI Rich. Stress Fields in a Compressed Unstrained Elastomeric 'O' Ring Seal and a Compression of Computer Prediction with Experimental Results, 11th International Conference on Fluid Sealing, 1987, pp.117-137.

[6] Itzhak Green and Capel English. Stresses and Deformation of Compressed Elastomeric O-ring Sealing, 14th International Conference on Fluid Sealing ,1994, pp.83-95.

[7] Qiu Xuanhuai. Mechanics Design. Fourth Version. Beijing: Higher Education Book Concern,1997, pp.100-109.(in Chinese)