HEFAT2007
5th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
Sun City, South Africa
Paper number: PX1

APPLICATION AND DESIGN OF A NEW QUICK-OPENING SEAL DEVICE CONNECTED BY D-SHAPE SHEARING BOLTS IN HYPERSONIC WIND TUNNEL

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ABSTRACT:

Application and design of a new quick-opening seal device connected by D-shape shearing bolts in hypersonic wind tunnel is introduced in this paper. This device is compact in structure, reliable in sealing, easy in assembly and disassembly, appropriate for end closure of pressure vessels or joints of pipes. Mechanical models are established for all major components, and strength calculation formulas are obtained which can be used for the design of the structure.

INTRODUCTION

For improvement the wind tunnel flow field quality, two stabilization sections have been manufactured in Φ0.5 Meter Hypersonic Wind Tunnel at China Aerodynamics Research and Development Center（CARDC）. The one is the high mach number(M8、9、10)and the other is the low mach number （M5、6、7）. And its operation pressure is 12MPa, Operation temperature is 800℃, diameter is 500mm. The two stabilization sections will be exchanged continually during wind tests, and it is necessary that design a quick-opening seal connected device between the stabilization section and the transition section for improvement the wind testing and decreasing the working intensity. The device position is shown in Fig.1.

At present, traditional bolted flange connections are mostly used both in pressure vessels and piping systems in industries and new design procedures and criteria have been developed[9]~[12], but it is difficult for them in assembly or disassembly and their sizes and weights are large especially with increasing in pressure and size required for pressure vessels or piping systems as well as their costs are expensive in production and maintaining etc. The end closure of pressure vessels, which (including top flange, head etc.) weights 10~30 percent of the total vessel and their costs can be as high as 15~40 percent [3][4] , or seal joint of pipes must have sufficient strength and sealing reliability. In addition, they should be simple in structure, easy in production, assembly and disassembly as well as convenient in maintaining. Two structural factors affect the sealing behavior: the type of flange connection and the form of sealing elements for end closure of pressure vessels or joints of pipes.

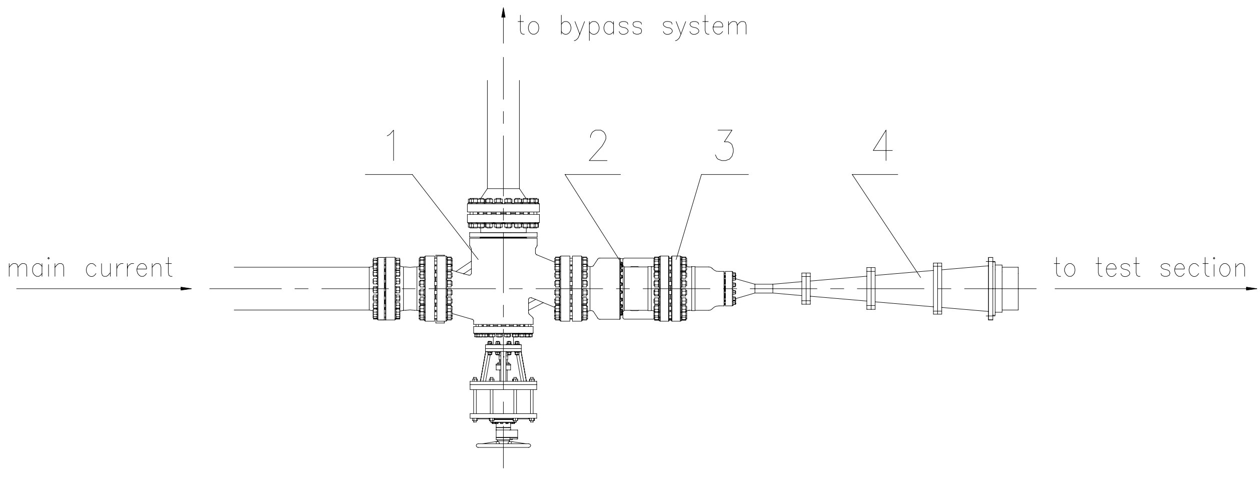

Figure.1 A quick -opening sealing device in the wind tunnel
1. High temperature valve 2. Quick -opening sealing device 3.settling chamber 4. Nozzle

So some easy or quick opening flange connections have been developed such as end closure with retaining ring joint, Casale's closure and self-seal Uhde's closure etc. for pressure vessels[1], pipe clamp connector (PCC) for pipe joints[2][13], but their other shortcomings like traditional bolted flange connections are still existed. In general, the sealing structure connected by shearing bolts with the head placed into the top flange of the cylinder for pressure vessels has the most compact structure and lightest weight comparing with above other joint structures, but general shearing bolted flange connection in the construction are often deadly seized and, as a result, it is very difficulty to dismount the head[4]. To solve this problem, some specially designed shearing bolts called flat shearing bolts[3] and D-shape shearing bolts[5] have been proposed and the primary researches present that their connection devices have the features of compact design, reasonable load-carrying and convenient manufacture etc..

For the sealing forms, O-rings, C-rings, double-cone etc. are widely used owing to their satisfactory sealing performances[7][8].

It is often increasingly needed for the flange connections to have a quick assembly and disassembly in many applications, for example, the process of supercritical CO_2 extraction, which requires batch production, convenient operation and reduced off-time to increase economic returns.

We selected a new quick-opening seal device connected by D-shape shearing bolts that it is invented by Dr. Ping Chen who is working in Beijing University of Chemical Technology by abroad investigation。In this paper, the structure design and the stress analysis for each component is performed in an actual application.

QUICK-OPENING STRUCTURE CONNECTED BY D-SHAPE SHEARING BOLTS

The quick-opening flanges connected by D-shape shearing bolts is a new sealing structure[5]. D-shape shearing bolt is shown in Fig.2. A 1/4 arc length of the circular screw thread along the longitudinal direction is cut off into a flat side from the traditional bolt, leaving a D-shape cross-section. Rotating all the D-shape shearing bolts in a definite angle, the two parts such as the head and the top flange for pressure vessels end closure are connected together which can transfer loading forces (i.g. the operating pressure for pressure vessels) by such shearing bolts. Per contra, rotating all the D-shape shearing bolts in the same angle in a reverse direction, the two components is disconnected without taking such bolts out.

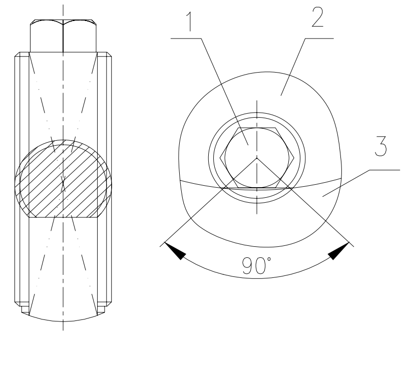

Figure. 2 D-shape shearing bolt and its joint
1. D-shape shearing bolts 2. joint component A
3. joint component B

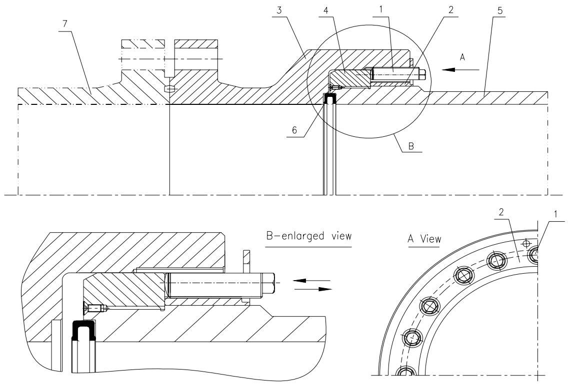

Figure.3 A quick-opening sealing device for a pressure piping system connected by D-shape shearing bolts
1. D-shape shearing bolts 2.bolt sleeve 3. transition section 4. screwed flange 5. stabilization section 6. C-ring gasket 7. pipe flange

STRUCTURE DESIGN

The quick-opening seal device is consist of transition section, sell stabilization section, D-shape shearing bolts, screwed flange and bolt sleeve , D-shape orientation pin. The self-energized C-ring is employed. It is shown in Fig.3.

The C-ring is mounted on the stabilization section by several seat screws. The screwed flange and the stabilization section are connected by thread. The seal specific pressure required for the C-ring is provided by rotating the D-shape shearing bolts, the end of which can push on the screwed flange and the stabilization section. If disassembled, all D-shape shearing bolts are rotated only about less than 180^o in reverse direction, the transition section and the stabilization section can be unlinked and the latter with its subassembly including such C-ring, all D-shape bolts, bolt sleeve etc. can be put out all together, so does it if assembled. D-shape bolt thread holes on the transition section and the bolt sleeve should only be required to be processed accurately together after the processing and heat-treatment of the stabilization section.

STRESS ANALYSIS OF THE STRUCTURE

In order to design the above quick-opening structure shown in Fig.3, stress analysis of the major components will be performed in this section. Analysis conditions are set as follows:

Design pressure：15MPa；

Design temperature：300□；
Inner wall diameter of system：Ø500mm；
Material of C-ring gasket: 0Cr18Ni9；
Material of D-shape shearing bolts: 35CrMoA；
Material of other components: 20MnMoNb.

(1) mechanical model and stress analysis of D-shape bolts

Forces acting on the D-shape bolt are shown in Fig.4. Among them, Q_1 and q_1 are respectively the shearing force and the normal force exerted by the top flange of the transition section, N_b and F_b are respectively the normal force and the friction force exerted by the screwed flange, q_2 is the normal force exerted by the bolt sleeve. The distributions of q_1 and q_2 are such that they should balance the forces acted on the bolt.

Each force can be calculated as follows:

The axial force Q_0 resulting from the medium internal pressure is

$$Q_0 = \frac{\pi}{4} D_G^2 p;$$

The pre-tightening unit sealing reaction F_a from the C-ring gasket is

$$F_a = \pi D_G q_0 \tag{1}$$

$$Q = Q_0 + F_a = \frac{\pi}{4} D_G^2 p + \pi D_G q_0 \tag{2}$$

The shearing force Q_1 exerted on a D-shape bolt by the transition section is $Q_1 = \frac{Q}{n} = \frac{Q_0 + F_a}{n}$, where, n is the number of the bolts.

The load equilibrium equations of the D-shape bolt can be given as follows:

$$Q_1 = N_b$$

$$\frac{1}{2} q_1 l = \frac{1}{2} q_2 l + F_b \tag{3}$$

In equation (3), $F_b = N_b f$.

The distributed shearing force Q_1 along the 1/4 thread arc length of each D-shape bolt (the angle of thread arc is about $\pi/2$) is simplified as a concentrated force which applies through the mass center of this arc length. Its normal distance from the axis of the D-shape bolt is x_1 (shown in Fig.4).

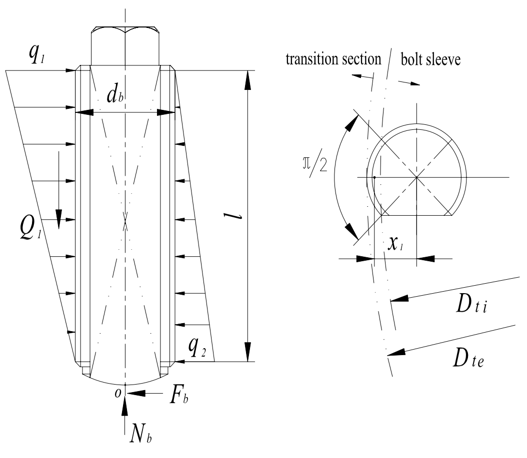

Figure.4 Mechanical model of D-shape shearing bolt

$$x_1 = \frac{\sin \frac{\pi/2}{2}}{\pi/2} d_b = \frac{\sqrt{2}}{\pi} d_b$$

The moment about the bottom center point of the D-shape bolt is:

$$\sum M_o = 0$$

Or $$Q_1 x_1 + \frac{1}{2} q_2 l \cdot \frac{1}{3} l - \frac{1}{2} q_1 l \cdot \frac{2}{3} l = 0 \tag{4}$$

Solving equations (3) and (4) yields q_1 and q_2 as follows:

$$q_1 = \frac{6(Q_0 + F_a)}{nl^2} \left(\frac{\sqrt{2}}{\pi} d_b - \frac{1}{3} lf \right)$$

$$q_2 = \frac{6(Q_0 + F_a)}{nl^2} \left(\frac{\sqrt{2}}{\pi} d_b - \frac{2}{3} lf \right)$$

Thus, shearing stress at the thread on the transition section side of the D-shape bolt can be given as follows:

$$\tau = \frac{Q_1}{d_b \sin \frac{\pi}{4} \cdot l\varepsilon} = \frac{\sqrt{2}\left(\frac{\pi}{4} D_G^2 \cdot p + \pi D_G \cdot q_0 \right)}{n d_b l \varepsilon}$$

The normal stress of the D-shape bolt resulting from the normal force N_b is

$$\sigma_m = \frac{N_b}{A_b} = \frac{\frac{\pi}{4} D_G^2 \cdot p + \pi D_G \cdot q_0}{n \cdot A_b}$$

Stresses obtained in above sections should be limited as follows.

$$\tau \le [\tau] = 0.577 \min\left\{ [\sigma]_b^t, [\sigma]_t^t \right\}, \quad \sigma_m \le [\sigma]_b^t$$

Where $[\sigma]_t^t$ is the allowance stress of transition section, $[\sigma]_b^t$ is allowance stress of D-shape bolts.

(2) mechanical model and stress analysis of transition

section

The mechanical model of the transition section is shown in Fig.5. The forces acting on the transition section include: internal pressure p ; the shearing force Q and the normal distributed forces $q(x)$ on the inner wall thread section surface transferred from D-shape bolts and the both are assumed to be uniformly applied along its circumference and the force Q is also uniformly distributed but the latter $q(x)$ is supposed to be linear along its longitudinal direction; the pre-tightening unit sealing reaction F_a is the force at the sealing diameter from the C-ring gasket, and the force F_1 is from kingbolts and the sealing reaction, F_{1a} from octagonal gasket.

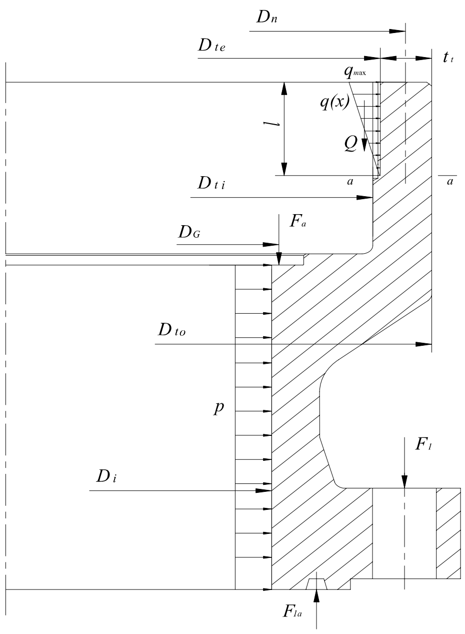

Figure.5 Mechanical model of transition section

The lower flange part can be designed as an integral steel pipe flange, of which the strength calculation can be performed based on the ASME, BPV Code, Section □ and/or GB150-1998 Steel Pressure Vessel in China. Here we concentrate on the design of top flange of the transition section connected by D-shape shearing bolts.

The maximum pressure value $q_{\max}$ of normal distributed pressure $q(x)$ can be calculated as follows[8].

$$q_{\max} = \frac{q_1 \cdot n}{\pi D_{ti}} = \frac{6(Q_0 + F_a)}{\pi D_{ti} l^2} \left(\frac{\sqrt{2}}{\pi} d_b - \frac{1}{3} lf\right)$$

$$= \frac{6\left(\frac{\pi}{4} D_G^2 p + \pi D_G q_0\right)}{\pi D_{ti} l^2} \left(\frac{\sqrt{2}}{\pi} d_b - \frac{1}{3} lf\right)$$

The strength calculation of the top flange of transition section connected by D-shape bolts resulting from the distributed force Q and $q(x)$ is based on the assumption of half infinite beam supported by an elastic basis, as a result, the maximum axial stress is found to be located at the inner surface on section a-a as shown in Fig.5. The axial stress resulting from Q as shown in Fig.5 can be calculated similarly with other top flange of end closure such as Kazali seal as follows[1]:

$$\sigma_{b\max}^{Q} = \frac{6M_{\max}}{t_t^{\,2}}$$

where,

$$M_{\max} = \frac{t_t}{4\pi\beta \cdot D_n \cdot l} \times$$
$$\left[1 + 4e^{-\beta l} \sin \beta l - e^{-2\beta l} \left(\cos 2\beta l + \sin 2\beta l\right)\right]\left(Q_0 + F_a\right)$$

with, D_n being diameter of centroid surface of top flange,

$$D_n = D_{te} + 2t_t$$

$$\text{If} \frac{D_{te}}{D_{t0}} \le 1.45, \quad t_t = \frac{D_{t0} - D_{te}}{4}$$

$$\text{If} \frac{D_{te}}{D_{t0}} \ge 1.45, \quad t_t = \frac{D_{t0} - D_{te}}{6} \cdot \frac{2D_{t0} + D_{te}}{D_{t0} + D_{te}}$$

$$\sigma_a^Q = \frac{4Q}{\pi\left(D_{t0}^2 - D_{te}^2\right)}$$

The maximum bending stress on a-a section as shown in Fig.4 resulting from linear distributed normal pressure $q(x)$ can be deduced based on the assumption of half infinite beam supported by an elastic basis as follows[6][14]:

$$\sigma_{b\max}^{q(x)} = \frac{6q_{\max}}{t_t^{\,2}} \cdot \frac{\phi(\beta l)}{\beta^3 l}$$

Where, $\beta = \sqrt[4]{\dfrac{12\left(1-\mu^2\right)}{D_n^2 t_t^{\,2}}}$ $\qquad D_n = D_{te} + t_t$

$$\phi(\beta l) = \frac{1}{8}\left\{1 - e^{-2\beta l} \cdot \left[1 - 2\beta l \sin^2(\beta l) - e^{(-2\beta l)}(1 + \beta l)\sin(2\beta l)\right]\right\}$$

The hoop stress due to $q(x)$ can be calculated conservatively by $q_{\max}$ uniformly exerted on the inner wall surface of the top flange as follows.

$$\sigma_{\theta\max}^{q(x)} = \frac{q_{\max}\left(D_{t0} + D_{te}^2\right)}{D_{t0} - D_{te}}$$

Stresses obtained in above sections should be limited as follows:

$$\sigma_{b\max}^{Q} + \sigma_{a}^{Q} + \sigma_{b\max}^{q(x)} \le 0.9[\sigma]_t^t$$

$$\sigma_{\theta\max}^{q(x)} \le [\sigma]_t^t$$

(3) Mechanical model and stress analysis of stabilization section

The loads applied on the stabilization section include: a shearing force Q on the outer thread from the screwed flange and the pre-tightening unit sealing reaction F_a at the sealing diameter from the C-ring gasket. The mechanical model of the stabilization section is shown in Fig.6.

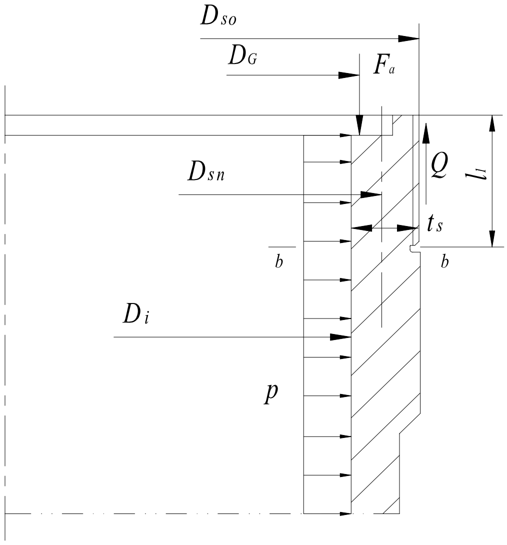

Figure.6 Mechanical model of stabilization section

The analysis of the stabilization section is similar to the transition section. It is found that the maximum axial stress locates at the outer on section b-b. Similarly, the axial membrane stress and the maximum axial bending stress on the section b-b can be expressed as follows.

$$\sigma_{\max} = \frac{6M_{\max}}{t_s^{\ 2}}$$

$$M_{\max} = \frac{t_s}{4\pi\beta \cdot D_{sn} \cdot l_1}[1 + 4e^{-\beta l_1}\sin\beta l_1 - e^{-2\beta l_1} \cdot (\cos 2\beta l_1 + \sin 2\beta l_1)](Q_0 + F_a)$$

Where, D_{sn} is diameter of centroid surface of end flange,

$$D_{sn} = D_i + 2t_s$$

If $\frac{D_i}{D_{s0}} \le 1.45$, $t_s = \frac{D_{s0} - D_i}{4}$

If $\frac{D_i}{D_{s0}} \ge 1.45$, $t_s = \frac{D_{s0} - D_i}{6} \cdot \frac{2D_{s0} + D_i}{D_{s0} + D_i}$

The membrane axial stress is σ_a, $\sigma_a = \frac{4(Q_0 + F_a)}{\pi(D_{so}^2 - D_i^2)}$

Stresses obtained in above sections should be limited as follows:

The equivalent hoop stress is

$$\sigma_{ma} + \sigma_a \le 0.9[\sigma]_s^t$$

(4) mechanical model and stress analysis of screwed flange

The loads applied on the screwed flange include: a pressing force N_B at the bolt circle and the friction force F_B from D-shape bolts and the shearing force Q on the inner wall thread from the stabilization section. The mechanical model of the screwed flange is shown in Fig.7.

Each force can be calculated as follows:

$$N_B = nN_b = Q_0 + F_a$$

$$F_B = nF_b = f(Q_0 + F_a)$$

The screwed flange is similar to a loose flange, under the action of a torque on the rectangular section ring, the maximum bending stress of this screwed flange, which locates at the inner wall of the screwed flange, can be obtained as follows.

$$\sigma_b = 0.75\frac{Q \cdot H_f}{t_f^2 D_{f0} \log\frac{D_{fi}}{D_{f0}}}$$

Stresses obtained in above sections should be limited as follows:

$$\sigma_b \le [\sigma]_f^t$$

(5) Mechanical model and stress analysis of c-ring gasket

The C-ring gasket seal is a linear contact seal. The axisymmetric finite element model can be established because of its axisymmetric size, loads and boundaries. The initial sealing force F_a is a total compressive force which applies at the contact circular line of the C-ring gasket, the direction of which is perpendicular to the plan which is formed by such the contact circular line.

The deformation of the C-ring gasket, which can be obtained from FEA, can be used to determine the axial moving distance between the ends of transition and the stabilization section, which is obtained by rotating the D-shape bolts during the pretightening process. The axial deformation of the C-ring gasket can be expressed as

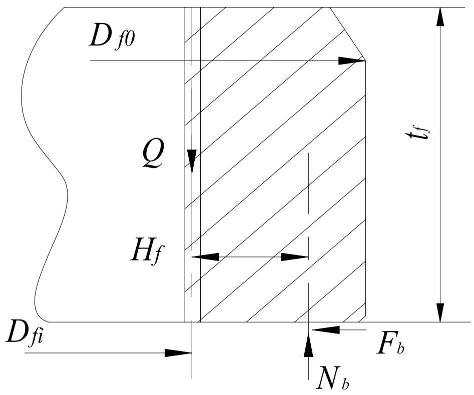

Figure.7 Mechanical model of screwed flange

follows.

$$u_z = F_a / K = \pi D_G q_0 / K$$

The value of K is determined through FEA and listed in table 1.

Table 1. Stiffness coefficient of the C-ring gasket

D_i (mm)	400	500	600
K(N/mm)	1.42×10^6	1.74×10^6	1.65×10^6

(6) Design result

Thus, the proposed quick-opening sealing structure can be designed based on above strength calculation formulas. As a result, under the above designing condition and materials chosen for each components, the size d_b and l of the D-shape shearing bolt designed is M42×3 and 105mm, the number of it is taken 20; the D_{ti}, D_{to} and l of the top flange of transition section is Ø690mm,Ø800mm and Ø105mm; the size of end screw of stabilization section is M598×5; the D_{fo} and t_f of screwed flange is Ø690mm and Ø102mm.

CONCLUSION

A new quick-opening sealing structure connected by D-shape shearing bolts have advantages with compact in structure, reliable in sealing, easy in assembly or disassembly and appropriate for end closure of pressure vessels or pipe seal joint.

Hydraulic test of this equipment has been carried out successfully, and the equipment is now used in practice with good seal ability and high safety. The highest operation pressure can reach 8MPa, and the highest temperature 800℃. Moreover, it is so convenient in its assembly/ disassembly and weak in its labor intensity that the work efficiency can be improved about 3~4 times comparing with the traditional bolt connecting method in stabilization section.

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