DEFORMATION BEHAVIOUR OF LARGE, HIGH-PRESSURE VESSEL FLANGES

H.A.C.M. SPAAS, D.G.H. LATZKO

Laboratory for Thermal Power and Nuclear Engineering, Delft University of Technology, Delft, The Netherlands

SUMMARY

1. *Introduction*

The analysis of the deformation behaviour of large high-pressure vessel flanges poses a much more difficult problem than for low-pressure flanges due to their particular geometry. The narrow flange faces cause the gasket reaction moments to be relatively large as compared to the bolting moment, thereby requiring an exact knowledge of the radii of application of these reactions. The general problem of the elastic analysis of such flanges can be solved by a schematization of the flanges by perfectly rigid rings joined to cylindrical and spherical shells. Recently this approach was further refined with respect to the influences of local gasket geometry, bolt behaviour and gasket closing and opening mechanism.

However, none of these approaches has taken into consideration the detailed effects of plastic deformation of the gasket faces. Furthermore, experimental information obtained on full-scale reactor vessels indicates that the approximation of the flange rings may lead to erroneous results when coupled to the assumption of purely elastic behaviour of the gasket faces.

The aim of the present paper is to supply information on these latter subjects.

2. *Elastic analysis*

For a particularly narrow flange geometry (typical of PWR flanges) a finite-element analysis (MARC-IBM-program, eight-node, isoparametric ring elements) was used to predict the behaviour of the flange rings. The nonlinear elastic problem resulting from the local closing and/or opening of the partial gap between the gasket faces was solved by an incremental technique using gap elements. The resulting deformation behaviour of the flange system has been compared to that obtained from an analysis using the refined rigid ring concept for both bolt-tightening and hydro-testing conditions. From the results the desirability of an elasto-plastic analysis became evident.

3. *Elasto-plastic analysis*

This part of the problem was solved by the same finite element program system as mentioned above. The incremental steps describing the nonlinear material behaviour are allowed to be larger than those for the gap-closure mechanism. Besides a comparison with the former elastic analyses an interpretation will be given of the local plasticity effects, which result in a shift in location of the gasket reaction.

4. *Experimental evidence*

Experimental data on local gasket face deformation was obtained by a specially developed laser beam apparatus, with the leak detection channel of the flange serving as a beam hole. Additionally strain gauges were used on flanges and bolts, in combination with special sensing pins for the determination of relative flange rotations.

5. *Conclusions*

Results obtained so far indicate that for high-pressure flanges of the narrow design investigated here the deformation behaviour is best described by an elasto-plastic finite element analysis. Where the rigid ring concept is retained for practical purposes a correction factor for local plastic behaviour of the gasket faces is suggested.

1. INTRODUCTION

Satisfying the stringent leak-tightness requirements for light water reactor flange joints (typically: zero leakage past the outer O-ring for both normal and upset conditions) requires a detailed knowledge of the deformation of these bolted joints under all relevant types of loading. The general approach to the deformation analysis of such flanges, viz. division of each member into its three constituent elements: ring, hub and shell, has found fairly universal acceptance as has the assumption of an undeformable radial cross section for the former element. Its implementation, however, poses the following problems:

- elastic behaviour of the constituent elements, notably the tapered hub.
- precise location of the point of application of the gasket reaction (required because of the relative importance of the gasket reaction moment, due to the small difference between the bolt pitch and flange ring centroïd radii).
- time lag between the flange and bolt temperatures during thermal transients - notably reactor startup and shutdown - due to the layer of stagnant air in the bolt holes of the cover flange ring.

Previous publications from the author's laboratory ([1], [2]) have dealt mainly with the first problem and presented some experimental information pertinent to the second one.
The aims of the present paper are twofold:

- to sound a note of caution concerning the rigid ring approach for those ring geometries now generally used for PWR vessel flanges.
- to present information on actual gasket face behaviour, including plastic deformation.

Both subjects will be dealt with on the basis of experimental and analytical work.

2. DEFORMATION OF FLANGE RING CROSS SECTION

2.1. Scope

The discussion refers to the ring elements of the flange assembly - shown as shaded areas in the cross-sectional figure 1 - and more specifically to rings characterized by the following shape factors:

$$1{,}5 \leq \frac{t}{b} \leq 1{,}65 \qquad\qquad 0{,}15 \leq \frac{b}{r_{cf}} \leq 0{,}22$$

viz. by a considerably greater $\frac{\text{height}}{\text{width}}$ ratio than the rings studied in references [1] and [2] (t = height, b = width, r_{cf} = radius of centroïd of flange ring).

2.2. Experimental evidence

2.2.1. Test objects and conditions

The measurements described below were carried out on the flange connections of four full-size PWR vessels, the shape and main dimensions of which are indicated and tabulated in figure 1, during a full hydrotest load cycle including bolt tightening, pressurization up to the full hydrotest pressure, subsequent depressurization and bolt loosening.

2.2.2. Measuring principle, equipment and procedure

Tangential strains ϵ_t were measured at both the inner and outer surfaces of the vessel and cover flanges on at least four levels per flange ring. From the resulting radial displacements w ($w = \epsilon_t . R$) the rotation ϕ of the flange between any two levels A and B follows directly from

$$\phi = \frac{w_A - w_B}{x} \qquad (1)$$

where x is the axial distance between A and B. Strains were measured in at least two meridional sections per flange. By taking one section through a bolt centerline and the other midway between two adjacent bolt holes a possible weakening effect of the bolt holes could be accounted for. Actually no significant effects of this nature were observed. Hottinger strain gauges type LA 11 with 6 mm active length were used on both inner and outer surfaces. In the former case a protective layer of Hottinger type AK 22 adhesive was applied after prior verification of its halogen-free composition and of its effectiveness in high-pressure water. For further details concerning terminals, wiring, vessel penetration etc. the reader is referred to earlier publications e.g. [3].
Gauge outputs were converted to punched tape data by Peekel/A.I. 50-point data logging units with automatic scanning at a speed of 4 points/sec, amply sufficient for these static measurements. (De)pressurization generally took place in steps of 2MN/m^2 (20 bar), a complete set of measurements being taken after each step. The punched tape data were converted off-line to tables and plots by the University computing center's IBM 360/65 and associated Calcomp plotter.

2.2.3. Results

For the sake of clarity only the radial displacements after bolt tightening and at maximum pressure (about 21 MN/m^2) are presented, as dotted and dash-dotted lines respectively, in figures 2 and 3 of this paper. The former figure refers to flange shape A, the latter to flange shape B of figure 1.

2.3. Analysis

2.3.1. Methods

Two computer programs based on different approaches were used to predict the flange deformations.
In the first the joint is divided into a number of constituent elements schematically indicated in figure 4, the deformation of each element being expressed in terms of the redundant shear forces and moments at their junctions. These are thus determined from equilibrium and compatibility conditions. This well-known approach was somewhat refined compared to earlier publications ([1], [2]) by increasing the number of elements and applying numerical integration to the differential equations describing the behaviour of the "short" cylindrical elements between the flange rings and the cover and vessel shells. The focal elements in the context of this paper, viz. the flange rings proper (shaded areas in figure 1), are treated as rings with perfectly rigid meridional cross sections in this program. Their moments of inertia are adjusted for bolt hole weakening by reduction of the outer ring radii.

The second program is based on the finite element method and divides the entire structure

into isoparametric ring elements with eight nodal points. The mesh size is varied to provide for a fine subdivision in the gasket area, as indicated in figure 4. The MARC program system [4] - implemented on the University's new IBM 370/158 computer - was used since it is particularly suited for the analysis of non-linear materials behaviour. As the inclusion of plasticity effects should hardly affect the deformations predicted by the program, the latter was run for both linear elastic and elastic-plastic materials behaviour to provide a check on its results. In the plasticity model isotropic hardening was assumed.
Another interesting capability of the program is the use of so-called "gap"elements, whose elastic moduli may be changed in response to the magnitude of strain. This option was used for modeling the non-linear contact problem posed by the local closing and/or opening of the partial gap between the gasket faces in the following way. Opposite points on the upper and lower gasket surfaces were connected by "gap" type beam elements with the following prescribed elastic modulus changes:

- E approaching infinite for negative strains exceeding in absolute value - 0.001%
- E approaching zero for positive strains.

2.3.2. Results

The radial displacements predicted by the rigid ring approach for maximum pressure loading are drawn as full lines in figures 2 and 3. They can thus be directly compared to the experimental results presented in the same figures for the two types of flange under consideration.
In figure 3 the radial displacements predicted by the finite element approach with and without plasticity effects are shown together with those predicted by the rigid ring approach, all referring to the B type flange. As in the preceding figures the results are given for both bolt tightening and maximum pressure conditions.
In view of the excellent agreement between the two finite element options indicated by figure 3 only the elasto-plastic version - being the more realistic one with concern to computed stresses - will be considered in the remainder of this paper.
Figures 2 and 3 compare the radial displacements for bolt tightening and maximum pressure conditions with the corresponding experimental results, for type A and type B flanges respectively.

3. GASKET FACE BEHAVIOUR

3.1. Scope

The discussion refers to the rotation of the lower gasket face relative to the vessel flange ring of which it forms a part. The reason for this limitation to the lower part of the flange joint is given below in the section on experimental techniques. The aim of this part of the study was to verify the analysis predicting the elasto-plastic gasket face behaviour and thereby to increase the level of confidence in the localization by analysis of the gasket reaction.

3.2. Experimental evidence

3.2.1. Test object and conditions

The measurements described below were carried out on vessel number 3, equipped with the A type flange joint (cf. figure 1), during a full hydrotest load cycle.

3.2.2. Measuring principle

The principle is indicated in figure 5. A laser beam generated by a He-Ne emitter is projected through the leak detection channel and reflected by a small mirror fastened in the channel immediately below the gasket face. The reflection shows as a dot on a scale mounted on the mounting pad. The displacement of this dot indicates the gasket face rotation relative to the laser emitter. In order to obtain the rotation of the gasket face with reference to the vessel flange possible effects of relative motion between the emitter and the flange must be accounted for. This is done by means of a second or reference laser beam emitted from the lower end of the instrument and reflected by a mirror fastened to the instrument's mounting pad onto a scale on the vessel.

For the sake of completeness it might be mentioned that the measuring principle could have been extended to the upper gasket face by attaching a mirror to it too and dividing the surface of the mirror attached to the lower gasket face in two parts, one for direct reflection as described above and the other for deflecting part of the incident beam onto the upper gasket face mirror. However, the risk of this latter mirror coming loose during the test and damaging the gasket surface was considered unacceptable for a production vessel.

3.2.2. Results

The rotation of the lower gasket face with reference to the vessel flange outer surface - i.e. after correcting for the mounting pad flexibility in the manner described above - is shown in figure 6 from bolt tightening through the maximum pressure of 21,4 MN/m^2. Two explanatory remarks seem in order. The first refers to the change in sign of the rotation observed at about 18 MN/m^2 internal pressure. This is due to the well-known fact that the stiffness of the cover flange exceeds that of the vessel flange. The resulting tendency of the latter towards a relative radial outward movement is prevented by the gasket face friction. The decrease of the latter with increasing pressure causes a sudden outward movement of the vessel flange relative to the cover flange to occur at the above mentioned pressure, which explains the phenomenon shown in figure 6. The second remark concerns the absence of data for the depressurization phase. This is due to the fact that the light dot disappeared after the first depressurization step. Subsequent inspection after return to atmospheric conditions and removal of the cover revealed that the vapour-deposited reflecting layer of the internal mirror had disappeared for unknown reasons. Time did not permit a repeat test.

3.3. Analysis

3.3.1. Method

The elasto-plastic finite element program used for this purpose has been mentioned in section 2.3.1.

3.3.2. Results

Although lower gasket face rotations relative to the vessel flange outer surface have been computed for bolt tightening and step-wise pressurization up to 18 MN/m^2 and found to show the same trend as the experimental results shown in figure 8, direct comparison to the latter does not appear to be relevant, because of the relatively large disturbance created by the leak detection channel (diameter 33.7 mm) containing the mirror in the deformation pattern of the gasket face (radial width 112 mm) and the relatively large distance between mirror and yield zone.
The incremental spread of plastic deformation with gasket face pressure, computed by the program for the A type vessel flange, is shown in figure 7.

4. COMPARISON OF EXPERIMENTS AND ANALYSIS

4.1. Flange ring behaviour

Although the relative rotations of the flange joint are fairly well predicted by the rigid ring approach for both bolt tightening and pressurization conditions, comparison with the experimental results shows that this approach seems less suitable for describing the deformation behaviour, characterized by the radial displacements of flange configurations typical for present PWR vessels. It will thus also result in an underestimation of the stress distribution.
The flange deformation behaviour will be adequately predicted by an elastic finite element analysis, provided that a suitable option is available to simulate the gap-closure mechanism. Deviations in the computed displacement fields presented - particularly in the near-gasket face region - are mainly due to the fact that in the finite element analysis the phenomenon described in section 3.2.2 was not taken into account, in other words that the gasket friction factor was assumed to be infinite throughout the entire analysis. From figs. 2 and 3 it can be concluded that the results will improve qualitatively through the use of a more realistic friction factor.

4.2. Gasket face deformation

The overall deformation behaviour of a flange joint is hardly affected by local yielding of the gasket face as may be concluded from a comparison of the elastic and elastic-plastic calculations. Although a direct quantitative comparison of computational results and experimental data did not appear useful for reasons stated above, it is important to note that they indicate similar tendencies. Figure 7 shows how the program predicts yielding to occur over a fairly large region in the gasket-face layer. For the B-type flange yielding over an even larger region was calculated.

5. CONCLUSIONS

- The rigid ring approach underestimates the deformation of a PWR flange joint, in particular as regards the vessel flange.
- An analysis based on the use of elastic finite elements adequately predicts the

deformation of these flanges, notwithstanding extensive yielding in the immediate vicinity of the gasket face.

- Since the non-linear contact problem posed by the closure mechanism of the gap between head and vessel flanges requires an incremental solution, an elastic-plastic analysis may be made without notable increase in computational costs, but with an evident improvement in the accuracy of the calculated stress distribution.

ACKNOWLEDGEMENT

Permission of the Rotterdam Dockyard Company and Rotterdam Nuclear to publish the experimental results, obtained during hydrotests at their works, is gratefully acknowledged.

REFERENCES

[1] VAN CAMPEN, D.H., DEEN, P.J., LATZKO, D.G.H.: "Deformation of large-diameter, high-pressure vessel flanges", Proc. First Int. Conf. on Pressure Vessel Technology, Delft, 1969, Part I, paper I-43, pp. 529-549.

[2] VAN CAMPEN, D.H.: "Computational methods for predicting tapered hub behaviour in large-diameter, high-pressure vessel flanges", Proc. First Int. Conf. on Structural Mechanics in Reactor Technology, Berlin 1971, paper G3/4.

[3] VAN CAMPEN, D.H., SPAAS, H.A.C.M.: "On the stress distribution in nozzle-to-cylinder connections" Nucl. Engineering & Design, vol. 21, 1972, pp. 368-395.

[4] MARC-CDC, User Information Manual, Revision G, November 1974.

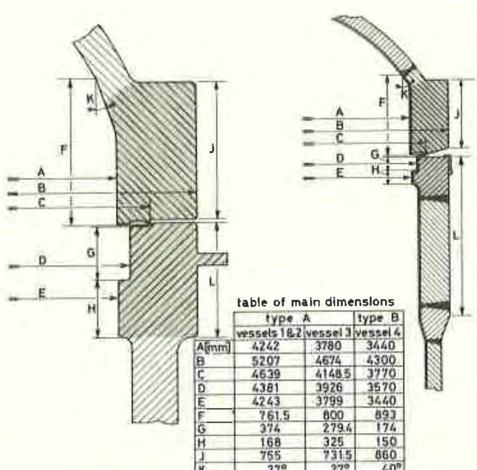

table of main dimensions

	type A		type B
	vessels 1&2	vessel 3	vessel 4
A[mm]	4242	3780	3440
B	5207	4674	4300
C	4639	4148.5	3770
D	4381	3926	3570
E	4243	3799	3440
F	761.5	800	893
G	374	279.4	174
H	168	325	150
J	765	731.5	860
K	27°	27°	40°
L	535	610	2100

Fig. 1. Main dimensions of PWR vessel flanges.

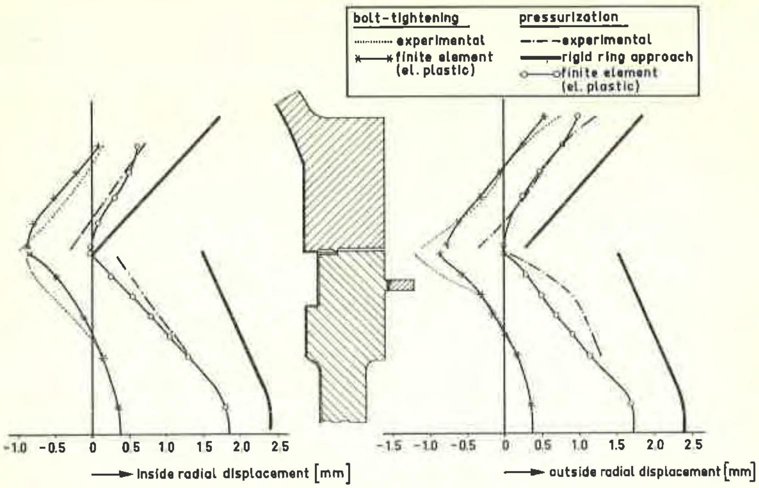

Fig. 2. Experimental and theoretical inside and outside surface displacements for A type flanges.

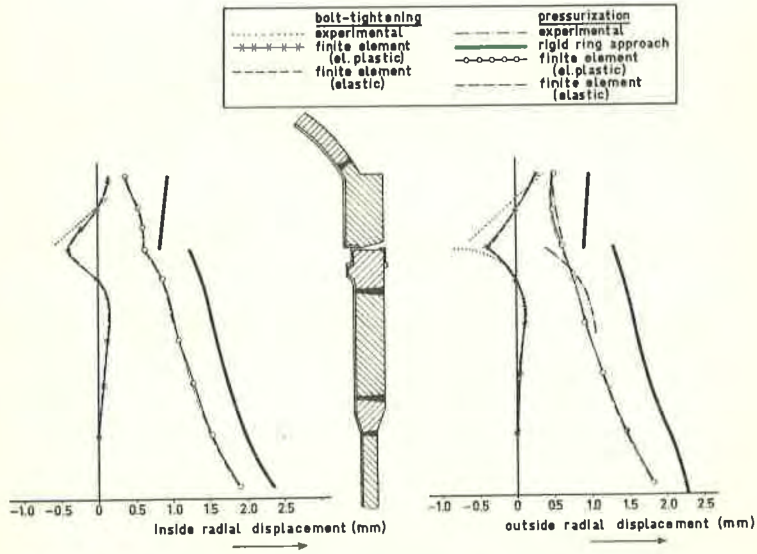

Fig. 3. Experimental and theoretical inside and outside surface displacements for B type flanges.

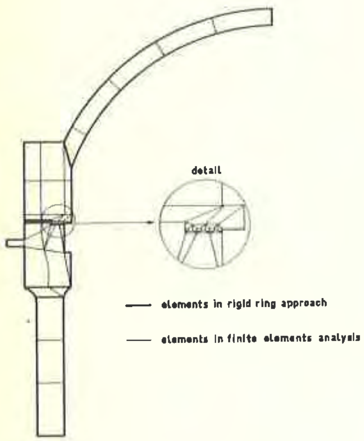

Fig. 4. Element divisions.

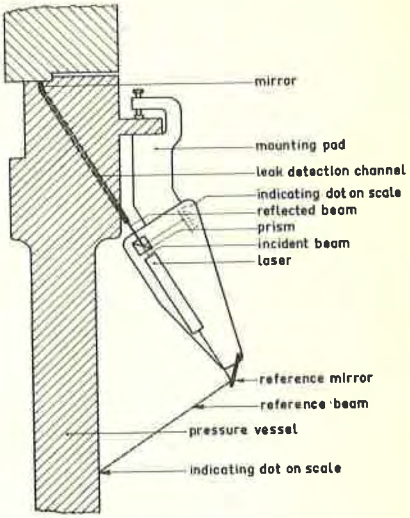

Fig. 5. Mounting of laser and use of reference beam.

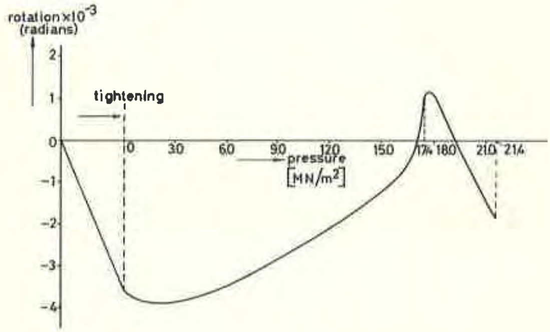

Fig. 6. Measured gasket face rotation during bolt-tightening and pressurization.

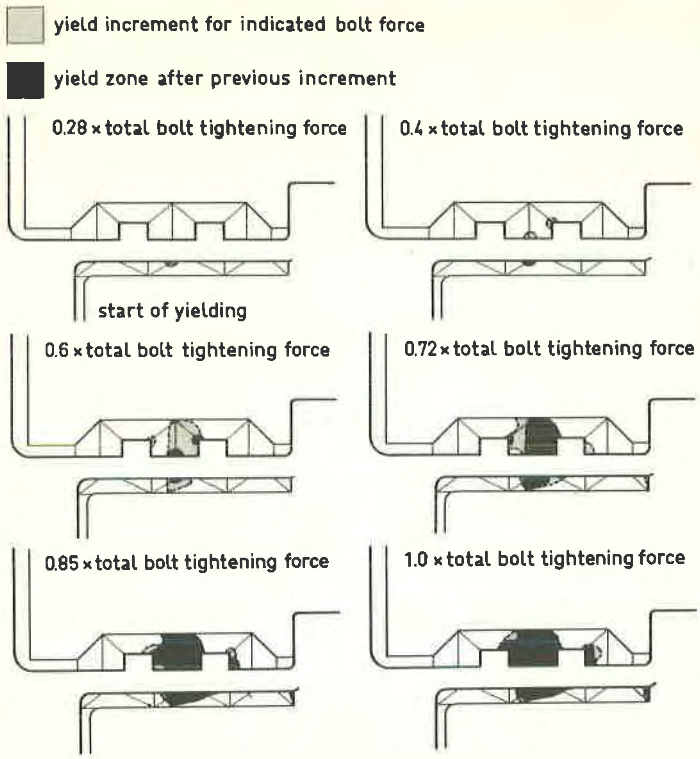

Fig. 7. Incremental plasticity zones during bolt-tightening.