INFLUENCE OF THE BOLT SYSTEM ON THE RESPONSE OF THE FACE-TO-FACE FLANGED CONNECTIONS

J.T. PINDERA*, Y. SZE

Civil Engineering Department,
University of Waterloo, Waterloo, Ontario, Canada

(at present: Laboratoire de Mécanique, Université de Poitiers, F-86022, Poitiers, France)*

SUMMARY

Influence of the bolt tensile stiffness and the bolt preloading on the response of the face-to-face flanged connections and flanges alone has been studied in full size steel and plastic models of bolted and boltless connections.

The results show that bolt tensile stiffness and the bolt assembly forces are major design parameters since the elastic response of the flanged connection is influenced significantly by both these parameters. The influence of the variation of the axial shell force or of the internal pressure in the pressure vessel on the bolt working forces and on the deformation of the flanges significantly decreases when the bolt tensile stiffness and the bolt assembly forces increase. The bolt preload produces significant change of the geometry of flanges which result in a particular assembly stress state in the adjacent parts of the shell. The width and the location of the interface contact area depends mainly on the bolt preload and the bolt tensile stiffness.

The elastic response of the bolt assembly as described by the tensile stiffness is non linear and depends strongly on the load level in some particular cases.

Results indicate that model studies of such elastic systems shall be based on rigorous conditions of similarity. Consequently, the results of typical photoelastic stress freezing techniques cannot be—in general cases—either interpolated or extrapolated linearly.

1. Introduction

One of the consequences of the rapidly growing use of pressure vessels of various types is the increasing interest in optimization of the connections, especially bolted and boltless flanged connections.

At the present knowledge, it is still not possible to develop sufficiently accurate and sufficiently convenient analytical relations which satisfactorily predict the stress state, strain state and deformation state even for the simple case of the boltless face-to-face flanged connection [1]. In actual fact, the problem is theoretically very complicated. Typical industrial flanges are already outside of the range of validity of all solutions based on the thin plate (Kirchhoff's Model) theory, for which the plate thickness should be negligible with respect to all other dimensions [2].

The behavior of industrial bolted flanged connections is more stochastic than deterministic. For economical reasons the surfaces where contact occurs between the flanges, bolt heads, washers and nuts cannot be machined according to close tolerances. The clearance between a one-inch diameter bolt and its bolt hole is around 1/8 of an inch. As a result, bolt assembly forces produce bending stresses in flanges and in the shells which depend on the degree of warping and of waviness of all contacting surfaces. For the same reasons, bolts are usually loaded by a bending moment and an axial bolt force which does not necessarily act along the axis of the bolt hole. The values of bolt tensile stiffness as calculated from the well known elementary formula may deviate strongly from the actual values. All these facts are well known, however the influence of these facts in the real response of the bolted assembly is usually not taken into account in design.

In spite of what theory is used in design, it is obvious that the response of the bolted assembly is depended on the relative displacements of each component. Therefore, the bolt tensile stiffness and the level of prestressing of the bolt system are certainly two of the major design parameters. The former parameter characterizes the load-displacement relations of the bolt system, the latter parameter characterizes the initial displacement of the bolt system and the initial displacement which imposed to the flange connections.

The term "bolt system" denotes bolts with washers and nuts.

The purpose of this paper is to present and discuss some experimental data directly related to the elastic response of the bolt systems and its influence on elastic response of flanged connections.

The following quantities have been chosen as representing the behavior of connections: the bolt working force (operation force), interface contact force, the deformation of flanges and the interface contact area.

The independent parameters are the internal pressure, or the axial shell force, and the bolt assembly force.

2. Design of Experiments

Ten-inch diameter ASA 300 psi rated flanges have been chosen as the object of this study. The geometry of the system, the materials, material constants, allowable stresses, yielding stresses, elements and the acting loads have been chosen according to the recent Pressure Vessel Code [3,4].

Three types of model have been built based on the model similarity conditions. Since the behavior of such systems as bolted flanged connections is strongly nonlinear, [1,5,9], the ratio of corresponding strains in the object and its models should be equal to one.

Consequently, full size models were built and the applied loads were proportional to the corresponding ratios of moduli of elasticity of the model and object materials:

$$\frac{P_1}{P_2} = \frac{E_1}{E_2} \qquad (1)$$

The influence of differences in the Poisson's ratio has been neglected, since in the case under investigation this influence has been estimated to be less than 5% on the basis of the thin plate theory and the short beam theory.

The following loads have been considered as independent variables:

a) either the internal pressure "p", or the axial shell force "P_s" resulting from an internal pressure, calculated for the case when no flange separation occurs at the internal diameter.

b) the bolt assembly force "P_{bA}" with a certain bolt tensile stiffness "Z".

Obviously the bolt tensile stiffness Z depends on the geometry and material of bolts, washers and nuts; from a practical point of view it can be considered as the static equivalent of the general mechanical impedance.

All loads are presented as total loads (not load intensities).

Three basic types of models investigated are: Bolted Model, Boltless Model and Flange Model, Fig. 1.

a) Bolted Model made of steel ($E_1 = 2.1 \times 10^6$ kg cm^{-2} = 30×10^6 lb. in^{-2}) Fig. 1a, was an idealized version of the typical industrial connections; the tolerances of machining and of assembling were much higher than required by industrial standards. The roughness of contacting surfaces was 0.2 micron. The bolt tensile stiffness was close to that of the bolts used in practice. The model was loaded internally by oil pressure varying from 0 to 300 psi.

b) Boltless Model made of epoxy resin ($E_e = 37 \times 10^3$ kg cm^{-2} = 525×10^3 lb. in^{-2}), Fig. 1b, represented the mathematical model commonly used in theoretical analysis. The bolt circle force has been replaced by a bolt ring force 12.5 mm (0.5 in) wide. Incidentally, such connections occur in engineering practice.

The equivalent tensile stiffness of the bolt simulating systems corresponded roughly to the tensile stiffness of industrial connections where the bolt assembly force is set at 10% of the allowable load. The model was loaded internally by air pressure varying within the limits 0 to 15 psi (this corresponds to a pressure from 0 to 855 psi in the Bolted Model) or by the axial shell force, as in the Flange Model.

c) Flange Model made of plexiglas ($E_p = 32 \times 10^3$ kg cm^{-2} = 455×10^3 lb. in^{-2}), Fig. 1c, represented flanges not connected with shells. Flange Model was loaded by the ring bolt force through an equivalent bolt tensile stiffness as Boltless Model, and by the axial shell force.

More details on the design of experiments and on the measurement technique are given in paper [1]. Bolt assembly forces and bolt working forces for the Bolted Model have been measured by internal gages located close to the bolt axis as described in paper [6].

3. Experimental Results

This paper presents only the part of the results which characterize the elastic response of the bolt assembly and which characterize the dependence of the behavior of the face-to-face bolted and boltless flanged connections on the bolt tensile stiffness and on the bolt preload.

3.1 Elastic Response of Bolt System to Axial Loads

The bolt system consisted of twelve single bolt systems represented in Fig. 2a. Elastic response of such a system to bearing forces between the inner surfaces of washers and the outer surfaces of flanges can be conveniently characterized by the change with the axial load P of the distance ℓ_o between the inner washer surfaces, or by the change with the load of the overall length L_o. The first measure, $\ell_o = \ell_o(P)$ is more convenient for determination of the interaction between the flanges and the bolts; the second, $L = L(P)$ is more convenient for determination of the bolt load.

It is often assumed that the function

$$\ell_o = \ell_o(P) \quad \text{or} \quad \Delta\ell_o = \Delta\ell_o(\ell_o, P)$$

is linear with respect to P and is compatible with the definition of the bolt tensile stiffness, Z,

$$Z = \frac{P}{\Delta\ell_o} = \frac{AE}{\ell_o} \tag{2}$$

when ℓ_o is replaced by the so-called effective strain length of the bolt or effective bolt length, ℓ_e. The effective bolt length according to [4] is:

$$\ell_e = \ell_o + 2t_w + 0.5\, d_b \tag{3a}$$

The mechanical model of this formula is represented by Fig. 2b:

$$\Delta\ell_e = \Delta\ell_o = \frac{P}{Z_o} \qquad \text{where } Z_o = \text{const.} \tag{3b}$$

A better mechanical model of the actual bolt tensile stiffness is given in Fig. 2c; the corresponding formula for change of the distance ℓ_o is:

$$d(\Delta\ell_o) = \left[\frac{1}{Z_1} + \frac{2}{Z_2(P)} + \frac{1}{Z_3(P)} + \frac{1}{Z_4(P)}\right] dP \tag{4}$$

where Z_1 = const. and $Z_2(P)$, $Z_3(P)$, $Z_4(P)$ are nonlinear functions of P; values of Z_2, Z_3 and Z_4 increase with increasing P to a certain maximum value, $Z_4(P)$ depends strongly on the geometry of the nut.

The change with axial load of the overall bolt length can be approximated by a mechanical model represented on Fig. 2d; the corresponding formula is:

$$d(\Delta L_o) = \left[\frac{1}{Z_5} + \frac{1}{Z_6(P)}\right] dP \tag{5}$$

where Z_5 = const. and $Z_6(P)$ is a nonlinear function of P. Stiffness $Z_6(P)$ decreases with increasing P, to a certain minimum value.

The experimental results, presented in Fig. 3 confirmed the above analysis. Fig. 3 presents the calculated and the experimentally determined elongations of the system of 12 single bolt subsystems as in Fig. 2a, and of several bolt stimulating systems as a function of the total bolt force. The results for plastic models were recalculated for steel, by means of known model law (1).

Four characteristic bolt force levels are indicated on Fig. 3: bolt force corresponding to the rate pressure 300 psi; bolt force corresponding to the pressure 720 psi; and bolt forces equal to 50% and to 100% of the allowable bolt force, corresponding to the allowable bolt stress of 50,000 psi for twelve bolts made of steel type A-540 B 23 of one-inch diameter [3].

Curve (a) shows the variation of elongation with axial load for a prismatic bar having a cross-sectional area equal to the total cross-sectional area of twelve bolts, and the length equal to the distance between the washer inner surfaces, $(\ell_o)_a = 3.39$ in. The slope of the curve (a) represents the bar tensile stiffness, Z_a.

Curve (b) presents the elongation of a one-inch diameter bolt, threaded full length; the distance between the bolt head and the nut was equal to 3.64 inch (9.55 cm); the effective bolt length, ℓ_e, was calculated according to elementary formula (3): $\ell_e = (\ell_o)_b = 4.06$ inch. The bolt core diameter (0.84 in) has been taken as the bolt diameter.

The slope of the curve (b) is constant, and equal to the bolt tensile stiffness:

$$Z_b = \frac{n\ E\ A}{\ell_e} = 49.\ \times 10^6\ \text{lb in}^{-1} =$$

$$= 8.75 \times 10^6\ \text{kg cm}^{-1} \qquad (6)$$

According to this formula Z_b does not depend on the load level. Z_b is equal to about 85% of the Z_a.

Curve (c_1) presents the elastic response to axial tensile load of the system of 12 industrial bolts commonly used in such flanged connections, for $(\ell_o)_c = 3.39$ inch. Each single bolt system consists of the 1 inch bolt, two washers and one heavy series nut. As expected, the relation between the bolt force and bolt elongation is strongly nonlinear for low bolt forces. The slope of the curve representing the stiffness for the given force,

$$Z_c = \frac{dP}{d(\Delta\ell)} = Z_c(P) \qquad (7)$$

is initially very small but approaches the slope of the (b) curve when the bolt force exceeds 50% of the allowable bolt force. The slope of curve (c) approaches about 55% of the slope of the curves (a) and (b) only at very high loads, close to the allowable loads. However, the corresponding elongation is several times greater. For low bolt forces, corresponding to the rated pressure, the actual bolt tensile stiffness is only about 30% of the calculated value or less.

The slope of the curve (c_2) presents the stiffness of the bolt system used in the Bolted Model. Bolt shafts are cylindrical and 2.25 in. long. In spite of a 0.22 inch diameter and 2 inch long hole in the bolt axis, the tensile stiffness of these bolts is slightly higher than that of the industrial bolts.

Curve (d) presents the actual variation of the overall bolt length with the axial load. As expected, this relation is slightly nonlinear.

The slopes of the curves (e_1), (e_2), (e_3) present the tensile stiffness of the bolt simulating system for the Boltless Model. The slope of the (e) curves is close to the slope of the (c_1) and (c_2) curves when the bolt force corresponds to the rated pressure.

The slopes of the curves (f) and (g) present two tensile stiffnesses of the bolt simulating systems used to investigate the response of the Flange Models.

3.2 Influence of the Bolt Tensile Stiffness and the Bolt Assembly Forces on Response of Flanges not Connected with Shells

The influence of bolt tensile stiffness on the functional relationships between the bolt working forces P_b, the interface contact force, $R = P_b - P_s$, and the axial shell forces, P_s, is illustrated by Fig. 4 for several bolt assembly forces.

Results show that both the bolt working forces and the interface contact force increase with increasing bolt tensile stiffnesses.

The nonlinear relations between the axial shell force and either the bolt working forces or the contact force is clearly indicated. Increasing bolt preload causes an increase in the linear range of the dependence of P_b and R on the axial shell force. In the whole range of the axial shell force and the allowable bolt forces, the change of the interface contact force is strongly nonlinear for given values of bolt stiffnesses: the value of this force at first decreases with increasing axial shell force and then increases. Increasing bolt stiffness increases this nonlinear dependence especially when the bolt assembly forces are low.

Figs. 5a and 5b present schematically the dependence of the flange deflection in the axial direction on the bolt stiffness and the bolt preloading. The figures are self-explanatory. It follows clearly that the part of the deflection of the flanges due to the axial shell force decreases with increasing Z and P_{bA}; consequently, the rotation of the inner flange boundary decreases too.

One of the major parameters describing the behavior of the face-to-face flanged connection is the interface contact area. The typical dependence of the contact area on the bolt preload and the axial shell force is presented in Fig. 6. It follows clearly that when the bolt assembly force increases, the centre of the interface contact area approaches the bolt hole circle. The influence of the assembly forces on the location of the centre of the contact area is greater than the influence of the axial shell force, i.e., the influence of the internal pressure.

3.3 Influence of Bolt Assembly Force on Response of Flanges Connected with Shells

One of the most important problems of the stress analysis in flanged connection, namely the correlation between behavior of bolted and boltless face-to-face flanged connections, and the influence of the bolt assembly forces on real structures, can be discussed on the basis of results presented in Figs. 7a and 7b.

Fig. 7a presents the relation for the Boltless Model, namely the dependence of the working bolt force on the internal pressure for several bolt assembly forces. It follows that the variation of the bolt working forces with the internal pressure is negligible when the bolt assembly force is above a certain level, which depends on the value of the internal pressure.

Fig. 7b presents the same relation for the Bolted Model and the relation for the resultant of interface pressure. These relations are qualitatively identical with the relations for the Boltless Model; however, there are some insignificant quantitative differences. These differences are due to the following two reasons.

In the Bolted Model, the variation of the axial shell force is proportional to the internal pressure because an oil ring prevents the oil from penetrating between the flange faces. In the Boltless Model, the air can penetrate the space between flanges which results from the deflection of flanges; consequently, the total axial force resulting from the internal pressure increases faster than the internal pressure.

The Boltless Model is loaded by a ring bolt force 0.5 in. wide; this means that the load acts on 20% of the flange width. In the Bolted Model the bolt force is transmitted to the flanges by 2-inch diameter washers, loaded by bolt heads and nuts; in this model the load acts on 75% of the flange width.

The influence of the bolt preload on the axial deformation of the flanges of the Boltless

Model is illustrated by Fig. 8. The state of deformation is also influenced by the shell connected with flange.

The bolt preload produces certain radial deformations as shown in Fig. 9. Radial displacements of the flange depend nonlinearly on the bolt preload.

One of the factors characterizing the state of deformation of the flanges under bolt preload is the axial displacement of the bolt circle, Fig. 10.

4. Conclusions

The results show that the bolt tensile stiffness and the bolt preloading during assembling of the bolted flanged connections are major design parameters. The influence of these parameters on the response of the connection is particularly significant when the connection is loaded by the rated pressure and by an axial shell force, as well.

The influence of the pressure in the flanged connection of the axial shell forces on the flange deflection, on the variation of the bolt working forces, and on the location of the interface contact area decreases significantly when the bolt tensile stiffness and the bolt preload increases.

At high Z and P_{bA} values the contact area is practically centered around the bolt hole circle, for the whole range of pressure; thus the probability of leaking is lowered.

Variation of the flange deformation and of the bolt working forces with varying internal pressure and varying axial shell force decreases with increasing bolt stiffness and bolt preloading. It seems to follow from this fact that it should be possible to increase the service life of the metal-to-metal connections by increasing the bolt preloading and the bolt stiffness.

Response of flanged connections with flat gaskets depends obviously on the visco-elastic response to loads of gasket material. Some new data on creep of gasket materials are presented in [7].

A certain constraint for the upper limit of the bolt tensile stiffness and the bolt preloading seems to be represented by the acceptable maximum value of the flange rotation produced by bolt preload which in turn depends on the bending stiffness of the flanges: the results show that the flange deformation produced by bolt preload produces a distinct, non-negligible, stress state in the connection flange-shell. This assembly stress state in flange and shell could be detrimental or could be helpful from the point of view of the service life of the connection, depending on the signs and magnitudes of the assembly stresses induced by service loads.

The results show that the bolt tensile stiffness as calculated from the elementary formula is a rather poor measure of the bolt behavior, especially at low values of the bolt assembly forces. The so-called effective bolt length as defined by the linear formula for the bolt stiffness is a strongly nonlinear function of the bolt load. However, when bolt preloading is above 50% of the allowable bolt load for high strength steel, the bolt tensile stiffness approaches 50% of the calculated stiffness and varies insignificantly with the bolt load.

Influence of the bolt preloading and the bolt stiffness on the response of the bolted steel model, the boltless epoxy model and the plexiglas flange model is qualitatively the same. To a certain extent, this fact justifies the present tendencies to predict the behavior of a bolted flanged connection on the basis of the known behavior of a boltless connection.

The response of the connection is strongly nonlinear as already observed by other authors [8]. This fact limits applicability of various model techniques requiring excessive deformation, such as the photoelastic freezing techniques. The results of the model tests of bolted flanged connections are linearly transferrable to the prototype only when the strains in the model and object are identical. Deviations from this principle introduce errors of unknown magnitude and sign.

In summarizing, the following particular conclusion could be formulated:

1. Bolt tensile stiffness related to the distance ℓ_o between the washers is a strongly nonlinear function of the bolt load.
2. The length of the threaded part of the bolt should be kept to a minimum, to increase the bolt tensile stiffness.
3. Design Codes should prescribe the minumum values of the bolt assembly forces; for instance in the range from 50% to 90% of the allowable bolt force depending on the bending stresses in bolts and the presence of a gasket.
4. Model tests of such systems as bolted flanged connections should be performed at identical strains; this severely limits the applicability of freezing techniques.
5. Bolt tensile stiffness, i.e., the equivalent stiffness of the bolt load simulating system in any model test or prototype test, should correspond closely to the bolt tensile stiffness in the working conditions of the connection, otherwise the results are not transferrable.

Acknowledgement

The results and conclusion presented have been derived from data obtained during research work on behavior of bolted flanged connections, initiated by the Subcommittee on Bolted Flanged Connections, Pressure Vessel Research Committee and continued after completion of initial project. The initial part of the experimental research was sponsored by the P.V.R.C., the next part by the Department of Civil Engineering of the University of Waterloo, and by the National Research Council of Canada, under Research Grant No. A-2939. The opinions, findings and conclusions expressed in this paper are those of the authors and not necessarily those of the sponsors.

Bibliography

[1] Pindera, J. T. and Sze, Y., "Studies of Physical and Mathematical Models of Some Flanged Connections", Fourth International Conference on Stress Analysis, Cambridge (England), April 6-10, 1970; Proceedings, pp. 396-408. The Institution of Mechancial Engineers. London.

[2] Essenburg, F., Gulati, S. T., "On the Contact of Two Axisymmetric Plates". Journal of Applied Mechanics, June 1966.

[3] A. S. M. E. Boiler and Pressure Vessel Code, Section VIII, Division 1 and Division 2, 1968 Edition.

[4] A. S. M. E. Boiler and Pressure Vessel Code, Section VIII; Winter 1968 Addenda, December 31, 1968.

[5] Pindera, J. T., "Physical Basis of Modern Photoelasticity Techniques", Beitrage zur Spannungs-und Dehnungsanalyse, Vol. V, 1968, Akademie-Veilag, Berlin.

[6] Pindera, J. T., "A New Type of Bolt Force Gage", Pomiary, Automatyka, Kontrola, No. 4, 1960.

[7] Pindera, J. T., and Sze, Y., "Creep of Some Gasket Materials", Transactions of the CSME, Vol. 1, No. 2, June 1972.

[8] Munse, W. H., Petersen, K. S., and Chesson, E., Jr., "Strength of Rivets and Bolts in Tension", Journal of the Structural Division, Procs. of the A.S.C.E. March 1959, pp. 7-28.

[9] Pindera, J. T., and Sze, Y., "Response to Loads of Flat-Faced Flanged Connections and Reliability of Some Design Methods", Transactions of the CSME, Vol. 1, No. 1, Mar. 1972.

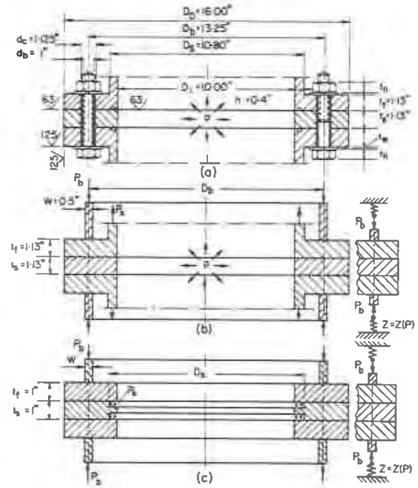

Figure 1: Shape and dimensions of the flanged connections investigated.

(a) Bolted Model (made of steel): simplified industrial prototype of 10 inch diameter ASA 300 psi rated flanges (very close maching tolerances) with 12 bolts of 1 inch thread diameter. Model is loaded by the bolt forces and by internal oil pressure. Left bolt: industrial connection. Right bolt: as used in tests.

(b) Boltless Model (made of epoxy resin): simplified boltless prototype. All dimensions as in first model with exception of the bolt holes. Bolt action is replaced by a bolt ring force P_b, acting on the bolt circle, through an equivalent static bolt tensile stiffness Z. Other loads are the axial shell force P_s and the internal air pressure, p.

(c) Flange Model (made of plexiglas): simplified model of boltless flanges not connected with shells. Models are loaded by the bolt ring force and by the axial shell force.

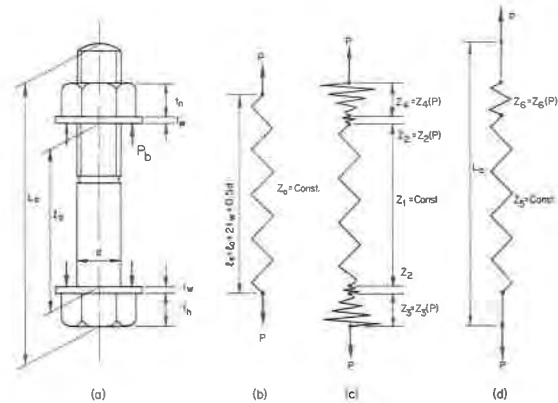

Figure 2: Geometry and elastic characteristics of the system consisting of 12 bolts, 1 inch diameter, with washers and nuts.

(a) Geometry of a bolt

(b) Elastic characteristic as assumed in [4]

(c) Actual elastic characteristic regarding ℓ_o

(d) Actual elastic characteristic regarding L_o

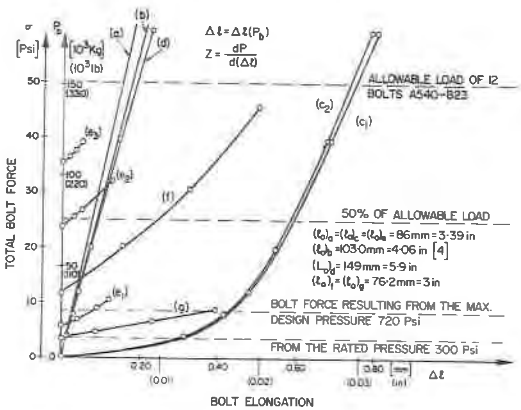

Figure 3: Elongation of the system of 12 bolts 1 inch diameter and of several bolt simulating systems as a function of bolt force.

(a) Elongation calculated for a prismatic bar in tension: bar cross-section area is equal to the total bolt core cross-section area: bar length is equal to the distance between the inner washer surfaces

(b) Elongation calculated according to [4], related to "ℓ_e"

(c) Actual variation of the distance between washers (denoted by ℓ_o); c_1 – measured for standard bolt with heavy series nut and two washers (USAS B18.2-1965); c_2 – measured for bolts used in the Bolted Model.

(d) Actual variation of the overall bolt length (denoted by L_o)

($e_{1,2,3}$) Elongation (separation) measured for a bolt simulating system acting on the Boltless Model for three levels of bolt preload.

(f,g) Elongations (separation) measured for two different bolt simulating systems acting on the Flange Model.

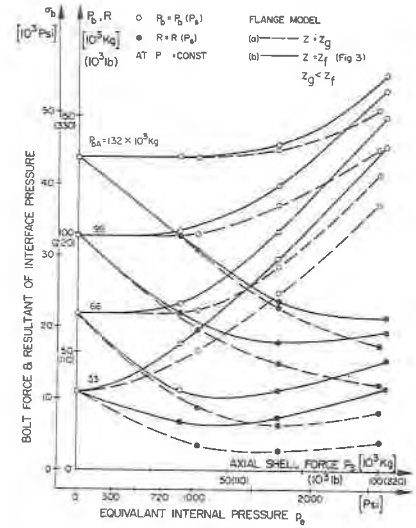

Figure 4: Influence of the bolt tensile stiffness Z on the variation of the bolt working forces P_b and the resultants of interface pressure R with the axial shell force P_s in Flange Model, for several bolt assembly forces P_{bA}.

G 2,

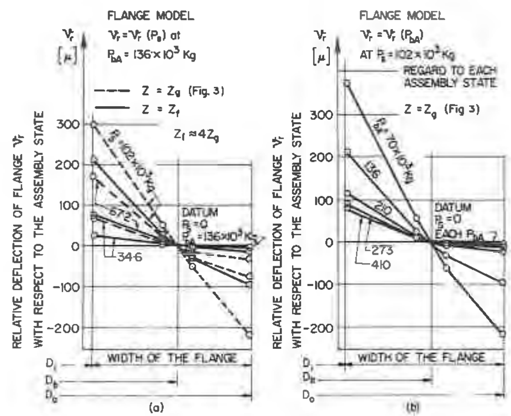

Figure 5: Typical relative deflections of the flanges of the Flange Model under the influence of various parameters.

(a) Influence of bolt tensile stiffness Z on the deflection of flanges, for several axial shell forces P_s, at a constant bolt assembly force $P_{bA} = 136 \times 10^3$ kg.

(b) Influence of bolt assembly forces P_{bA} on the deflection of flanges at constant axial shell force $P_s = 102 \times 10^3$ kg.

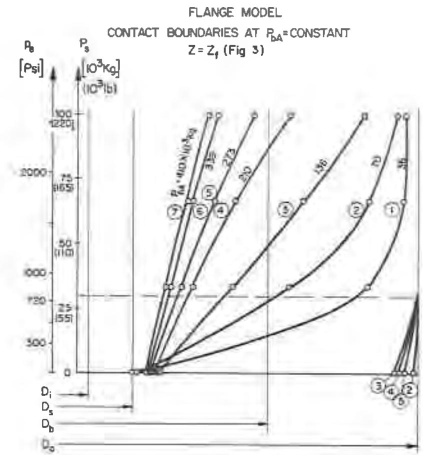

Figure 6: Influence of the bolt assembly force P_{bA} on the variation of the interface contact area with the axial shell force P_s, in the Flange Model, at bolt tensile stiffness $Z = Z_f$ (Figure 4).

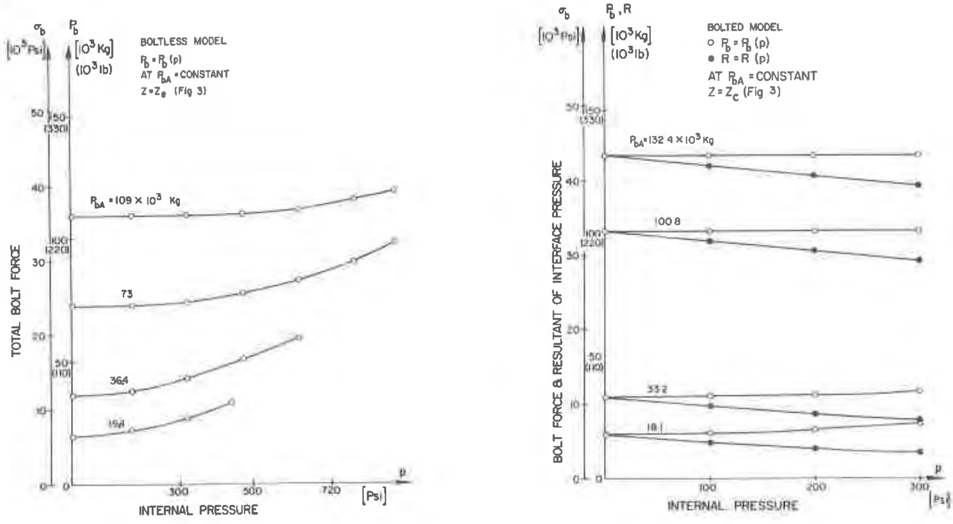

Figure 7: Influence of the bolt assembly force P_{bA} on the variation of the bolt working forces P_b and the resultant of the internal pressure R with the internal pressure p.

(a) Boltless Model $Z = Z_e$ (Figure 4)

(b) Bolted Model $Z = Z_c$

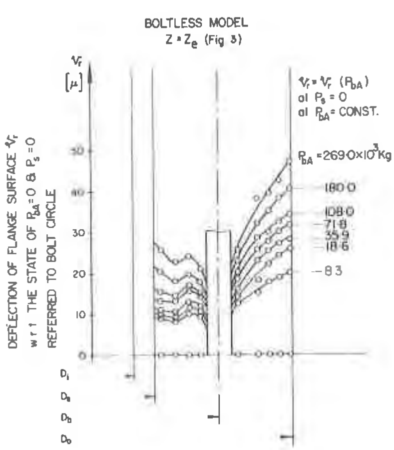

Figure 8: Axial deflection of the flanges surface of the Boltless Model with respect to the bolt circle, under influence of the bolt preloading.

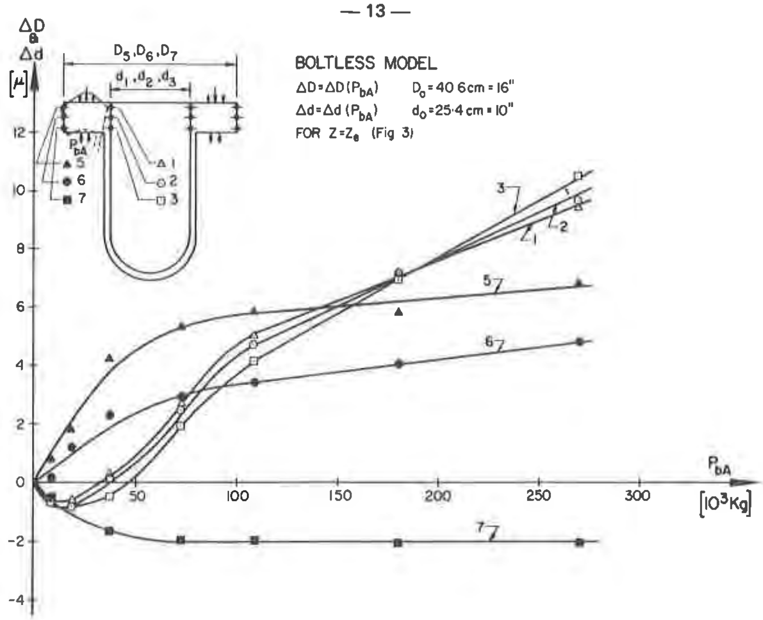

Figure 9: Change of the inner and outer diameter of the flange produced by bolt preload in Boltless Model.

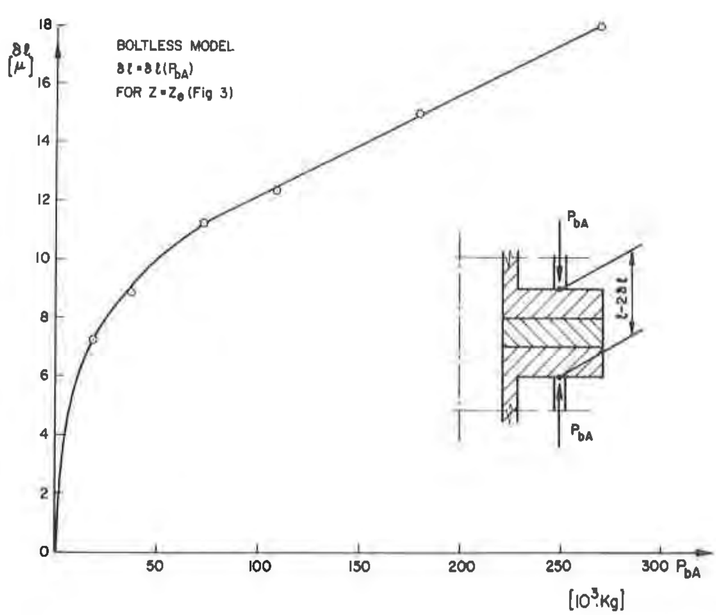

Figure 10: Displacement of the bolt circle $\delta\ell$ produced by bolt preload in Boltless Model.