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NEW SHAPE FOR THE SCREW DIAGRAM

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ABSTRACT

Although it may be expected that, with increase in internal pressure the bolt load in flanged joint would necessarily increase, this is not always true. This effect was tested experimentally by using strain gauges. Narrow gasket type was used with different materials and dimensions.

This paper deals with the analysis of the obtained results as well as with the theoretical analysis. One of the most interesting effect is that the tensile load increases or decreases as the applied pressure increases. It can be seen that it depends upon gasket material and its dimensions.

New shape for the screw (bolt) diagram has been proposed theoretically as well as prooved experimentally for the case when the bolt stress decreases with the increase of internal pressure.

INTRODUCTION

One of the most important parameter which designer must take into consideration when he deals with the problem of bolted connection, is the change of the tensile force in bolt due to the application of external force. Therefore Wesstrom and Bergh[I]put the theoretical bases for stresses developing in the bolted connection. They used the following equation which shows that the change in bolt force is proportional to the internal pressure.

$$F_{b_2} = F_i + \alpha P$$

Where F_i & F_{b_2} are the bolt forces before and after applying

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the internal pressure (p).

α is a parameter which may be positive or negative.

They proved that the sign of parameter (α) depends on the characteristics of the flanged bolted joint.

Strain gauges were used in [2,3] to measure the change in bolt strains during the increase of external load. It was proved experimentaly that the bolt stress may increase , decrease or remain constant during the increase in internal pressure, depending only on the initial bolt stress.
The same results were obtained in the year 1969 [4].

From the mentioned review, it has been shown that little consideration was paid to the effect of the gasket on bolt stress. Therefore it was decided here in this paper to study experimentally and analytically the change in bolt stress using narrow gaskets with different widthes, thicknesses and materials. Special test rig was designed where bolt stresses were measured using strain gauges. Three materials for narrow gasket were used with different dimensions. In each case the fluid pressure was applied on the bolted connection while stresses were observed untill leakage occurs.

THEORETICAL ANALYSIS

The beam sgo. Fig.1 represents the half cover of the bolted connection, while elements (s s`) and (g g`) represent the screw(bolt) and gasket respectively. The geometrical parameters and the material properties for each part are shown in the structure. The forces on the members (s s`) and (g g`) i.e on the bolt and gasket are (A, B) and are applied in the shown directions. Using Castigliano method (for example), which is based on the deformation energy for determining the displacements of the points (s,`g`), the following equations can be obtained:

$$f_A = \frac{\partial L}{\partial A} \quad , \qquad f_B = \frac{\partial L}{\partial B} \tag{1}$$

where f_A and f_B are the deformations in bolt and gasket, L is total deformation energy of the form structure.

$$L = \frac{A^2 l_1}{2E_1F_1} + \int_0^a \frac{M_1^2\,dx}{2Ej} + \frac{B^2 l_2}{2E_2F_2} + \int_a^l \frac{M_2^2\,dx}{2Ej} \tag{2}$$

$$M_1 = Ax \quad , \qquad M_2 = Ax - B(x-a)$$

$$\frac{\partial M_1}{\partial A} = X \quad , \qquad \frac{\partial M_2}{\partial A} = X \tag{3}$$

$$\frac{\partial M_1}{\partial B} = 0 \quad , \qquad \frac{\partial M_2}{\partial B} = (x-a)$$

$$f_A = \frac{A\,\ell_1}{E_1F_1} + \frac{1}{EJ}\int_0^a M_1 \frac{\partial M_1}{\partial A}\,dx + \frac{1}{EJ}\int_a^\ell M_2 \frac{\partial M_2}{\partial A}\,dx$$

$$= \left(\frac{\ell_1}{E_1F_1} + \frac{\ell^3}{3E\ell}\right)A + \left(\frac{-2L^3+3a\ell^2-a^3}{6EJ}\right)B$$

$$= C_{AA}\cdot A + C_{AB}\cdot B \qquad (4)$$

$$f_B = \frac{B L_2}{E_2F_2} + \frac{1}{EJ}\int_0^a M_1 \frac{\partial M_1}{\partial B}\,dx + \frac{1}{EJ}\int_a^\ell M_2 \frac{\partial M_2}{\partial B}\,dx$$

$$= \left(\frac{-2\ell^3+3a\ell^2-a^3}{6EJ}\right)A + \left(\frac{L_2}{E_2F_2} + \frac{b^3}{3EJ}\right)B$$

$$= C_{AB}\cdot A + C_{BB}\cdot B \qquad (5)$$

Taking into consideration the deformation conditions, the next equation can be written:

$$\lambda = f_A + f_B$$

$$\lambda = C_{AA}\cdot A + C_{BB}\cdot B + C_{BA}\cdot A + C_{BB}\cdot B$$

$$= C_A\cdot A + C_B\cdot B = \bar{f}_A + \bar{f}_B \qquad (6)$$

where $C_A = C_{AA} + C_{BA}$ and

$C_B = C_{AB} + C_{BB}$

Then the next coefficients are obtained :

$$C_A = \frac{\ell_1}{E_1F_1} + \frac{a(\ell^2-a^2)}{2EJ} \qquad (7)$$

$$C_B = \frac{\ell_2}{E_2F_2} - \frac{ab^2}{2EJ}$$

Equation (6) is typical for initialy tightening systems for example as shown in Fig. 2 where:

$$f_A = \frac{A\ell}{E_1 F_1} \quad , \qquad f_B = \frac{B\ell}{E_2 F_2}$$

$$f_A = C_A \cdot A \quad , \qquad f_B = C_B \cdot B \qquad (8)$$

where $C_A = \frac{\ell}{E_1 F_1}$, $C_B = \frac{\ell}{E_2 F_2}$

Then the next equation can be used

$$\lambda = f_A + f_B = C_A \cdot A + C_B \cdot B \qquad (9)$$

where $C_A > 0$ and $C_B > 0$

Considering the change in the stiffness of the elements of the last structure then, when C_A changes to $\overline{C}_A$, $\overline{C}_A > C_A$ and C_B to $\overline{C}_B < C_B$ the slopes of the characteristic elements change from the full lines to the dotted one Fig. 3. The extreme case can be obtained when $C_B = 0$ (i.e the stiffness $S = \frac{1}{C_B} \rightarrow \infty$), the bolt diagram changes from Fig. 3 to Fig. 4.
In the case under study the coefficient C_B could be positive as well as negative values, which is impossible in the typical initialy tighting systems. That happens when

$\frac{l_2}{E_2 F_2} < \frac{a b^2}{2E\gamma}$ as shown in equation (7), then $C_B < 0$.
For such situation the bolt diagram changes its shape to Fig. 5 In comparison to Fig.4, where the gasket is ideal stiff element, the element represented by the right line in Fig.5 will be known as a "over stiff". In fact, it is not the behaviour of the gasket alone but it is the property of the bolted system as a whole, which may be expressed by the value obtained from equations (5),(6). This means that the over stiffness as a physical property of materials does not exist. The use of the nomogram with over stiff member, $C_B < 0$, is shown in Fig.6. The coordinate of point (O_o) give the value of initial forces in bolts and gasket, where $A_o = B_o$, and the value of displacement (i.e the algebraic sum) represents the geometrical tightening (λ) of the system.
The value of the applied external force P is represented by the vertical distance between the two characteristics , where P = A - B. It can be seen that the force in both elements, the gasket and the bolt, decreases. It means that, the decrease of the compression in the gasket in this case is much

bigger than in the case with $C_B > 0$ Fig. 3; i.e it has big effect in leakage due to the applied fluid pressure.

The discussed effects are in a good agreement with the experimental results as well as [2,3,4] .

THE EXPERIMENTAL INVESTIGATION

Fig. 7. shows the used test rig which consists of cylinder (1) with welded flange (2). The cover, narrow gasket and the tested bolts are shown in the same figure. The pressure of the fluid is obtained by using oil gear pump connected with electrical motor. This pressure is measured by means of manometer, and the bolt strains are measured by using electrical strain gauges. Twelve bolts were used in this experimental work, the strain of six bolts were measured. Two opposite strain gauges were cemented on the shank of the tested bolts as shown in Fig. 8. By means of this method the average of the tensile force as well as the bending of the bolts can be measured.

Three materials of the narrow gasket were used, they are rubber, copper, and aluminium. Different thicknesses and widthes of the narrow gasket were used. The inner diameter of all gaskets was fixed and equal to the inner diameter of the cylinder. The tightening moment of the bolt was measured by means of calibrated torque wrench. Measurement was made with different initial tighting moment.

RESULTS AND DISCUSSION

Series of experiments has been carried out using several types of materials for the narrow gasket with different dimensions. The average tensile strains (ε_t) were measured and plotted in Fig. 9. which shows the behaviour of the bolts during increasing the internal pressure. During the application of the tightening torques, all technical advices which mentioned in [3,4] has been considered. From the curves in Fig. 9.,it can be seen clearly that the tensile force in bolts increases or decreases gradually due to the increase of internal pressure. It depends upon the gasket material and its dimensions. The obtained results agree with what were given in [2,3,4]. In trying to draw the bolt(screw) diagram from the obtained results. Two cases can be obtained:-

<u>Case I</u> -

Where the bolt stress increases as the internal pressure increase. In this case the line OA is easily drawn from the data obtained experimently. The line A B``C``O` can be determined by drawing horizontal lines b b`, c c`, d d`represent the bolt forces due to the change in fluid pressure. From the intersection points B`, C` and d` vertical lines can be drawn. Points B``, C``, and d`` can be plotted so that the vertical distances B`B`` and C`C`` and d`d`` represent the applied pressures. The straight line passes through points A B``C``d``O` represents the gasket bahaviour with a slope $C_B > 0$ as in Figs. 3,10,11

Case II -
Where the bolt stress decreases as the internal pressure increases. The same steps which were mentioned above is followed and the obtained diagrams are shown in Figs. 5 ,12,13,14,15. In this case the slope of the line o`A` changes to get $C_B < 0$ as in fig.5.

Figure 16 shows the whole cases which can be met, where seven bolt diagrams are drawen with different slopes but with constant force due to the initial tightening moment and with constant ($\lambda = O_A O_B$). A constant internal pressure (P) has been applied in each bolt diagram and the corresponding change in bolt and gasket forces can be determined. The dotted curve ($b_1 b_2 b_3 b_4 b_5 A_6 b_7$) represents the change in the bolt force for different cases,while the dotted curve ($c_1 A_2 c_3 c_4 c_5 c_6 c_7$) represents the change in the gasket force.

All cases may happen except case (I) where $C_A < 0$, because it is impossible as mentioned in eq.7. It can be said that,the increase rate of the bolt force due to the constant internal pressure (P) decreases gradually as point (A_2) moves in the direction of point (A_7). At point (A_6) the bolt force change equal to zero,while it equals to a negative value at point (A_7) where $C_B < 0$.

CONCLUSIONS

The main conclusion of this paper indicates that the change in bolt stress depends on gasket materials, and its dimensions. The bolt stress increases or decreases with the increase of the internal pressure.

New shape for the bolt diagram can be obtained in the case when the bolt force decreases with the increase of the internal pressure.

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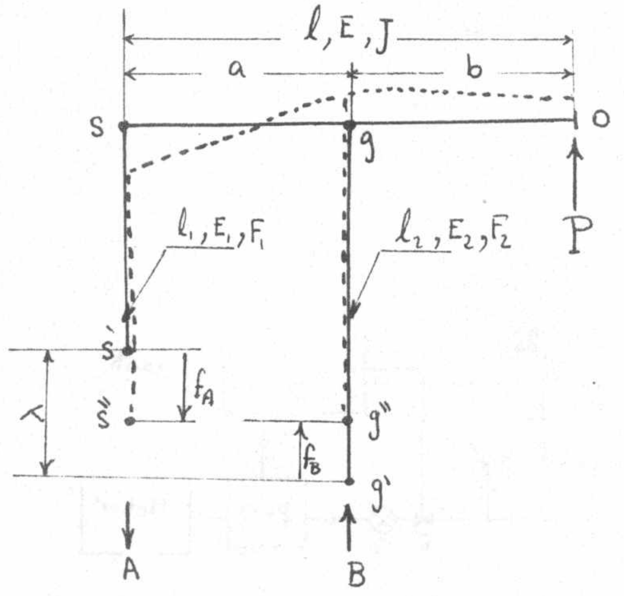


Fig.1.Forces and deformations in narrow gasket.

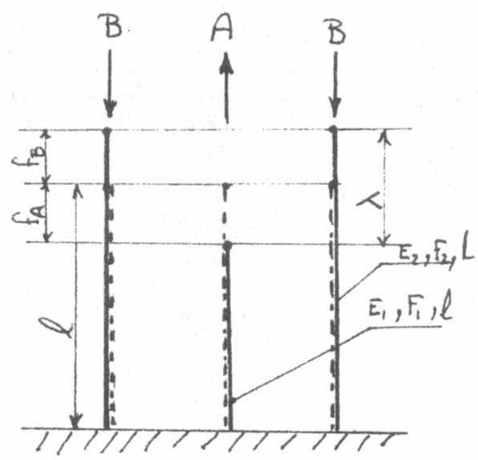


Fig.2.Typical system.

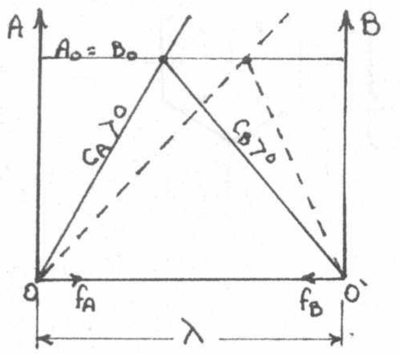


Fig.3.Screw diagram with different stiffnesses.

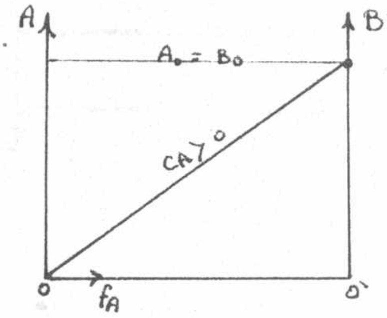


Fig.4.Screw diagram with ideal stiff gasket.

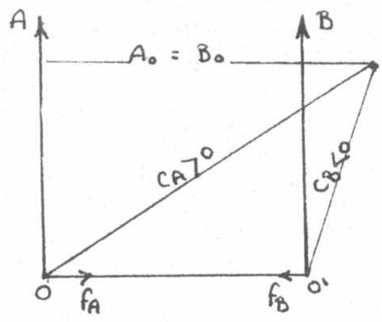


Fig.5.Screw diagram for nontypical system.

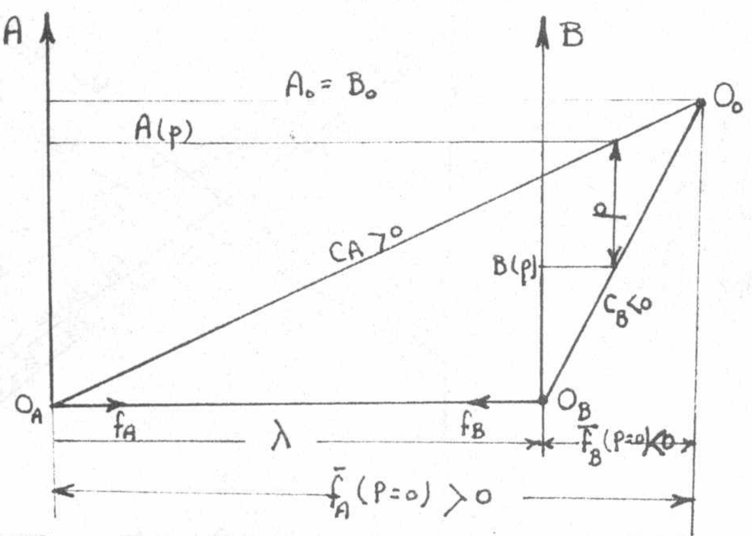


Fig.6.The screw diagram using over stiff gasket.

Fig.7.The test rig,where:-
I.steel cylinder,
2.welded flange,
3.narrow gasket,
4.cover,
5.tested bolts,
6.return valve,
7.nunreturn valve,
8.manometer.

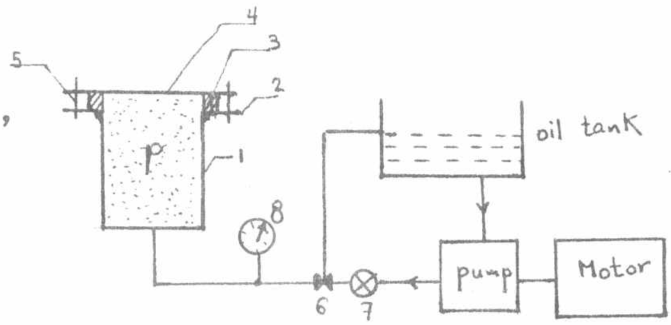


Fig.8.The tested bolt with two opposite strain gauges.

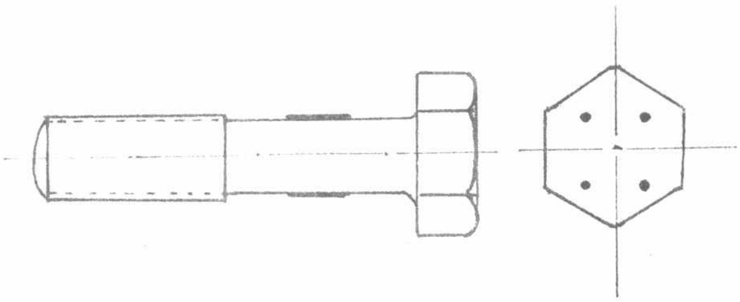
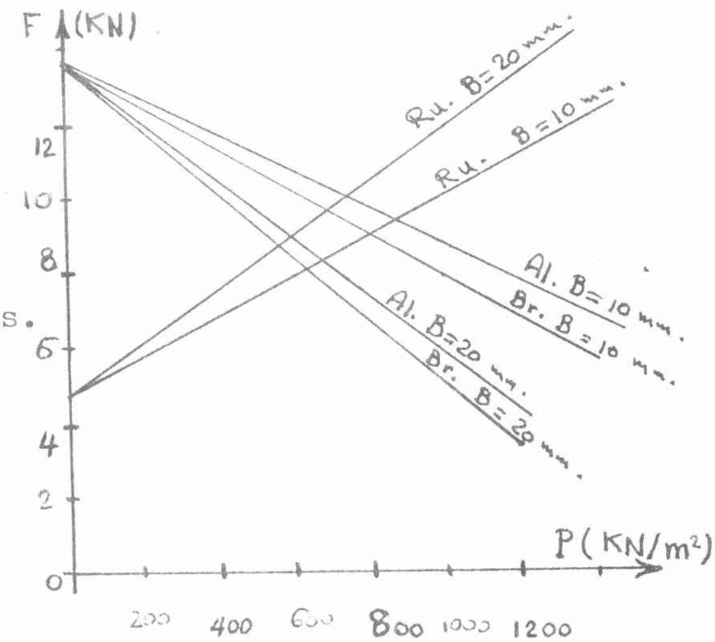


Fig.9.The relationships between the internal pressure (P) and the tensile force in bolts for different gasket materials and dimensions.



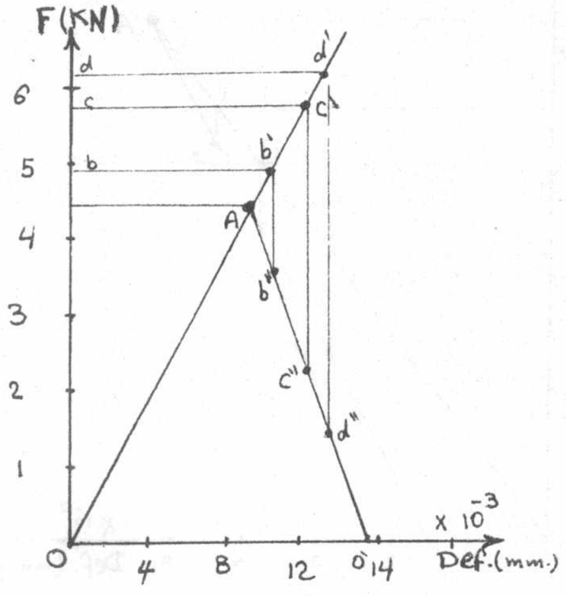


Fig.10.The screw diagram,case I where $C_B > 0$,for rubber gasket with width W= 20 mm.,thickness t = 3 mm.

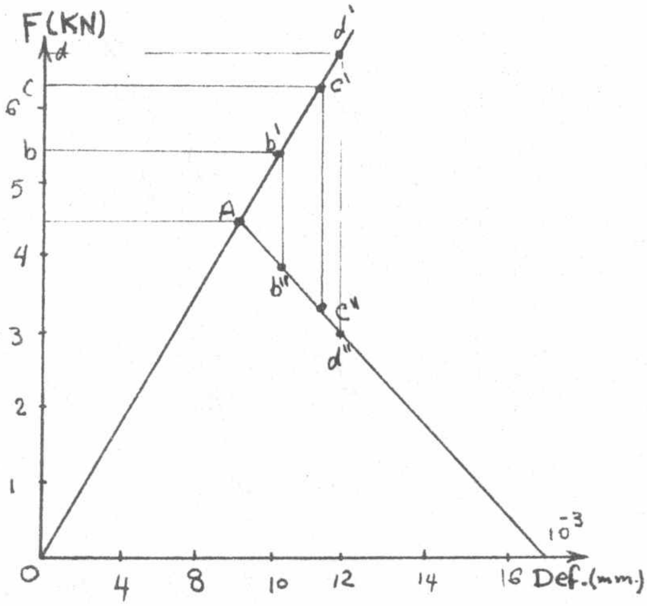


Fig.11.The screw diagram,case I where $C_B > 0$,for rubber gasket with W = 10 mm., t = 3 mm.

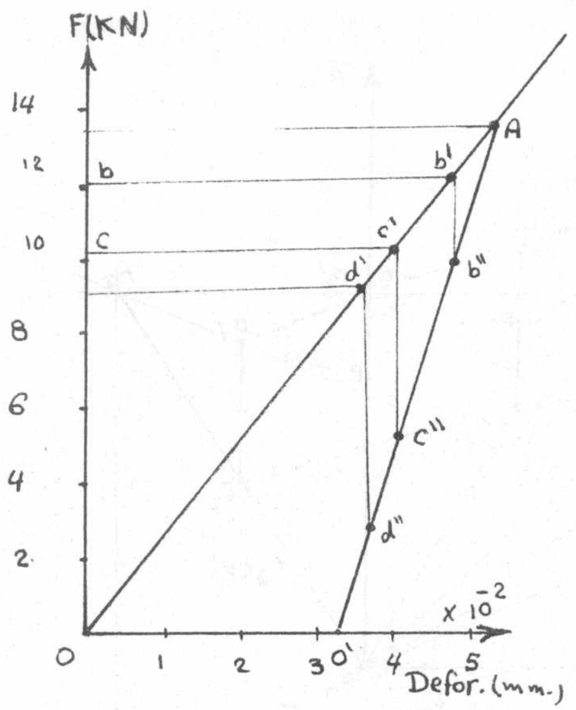


Fig.12.The screw diagram,case II,where $C_B < 0$,for brass gasket with W= 20 mm., t = 3 mm.

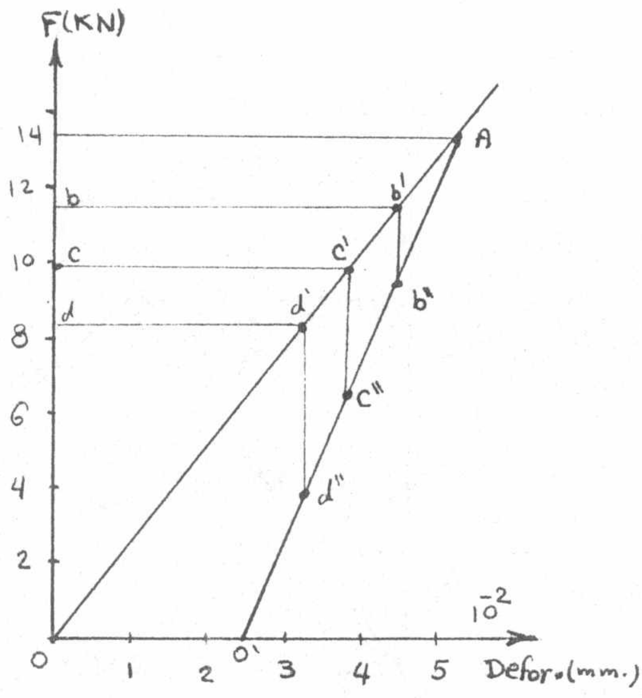


Fig.13.The screw diagram,case II,where $C_B < 0$,for brass gasket with W = 10 mm., t = 3 mm.

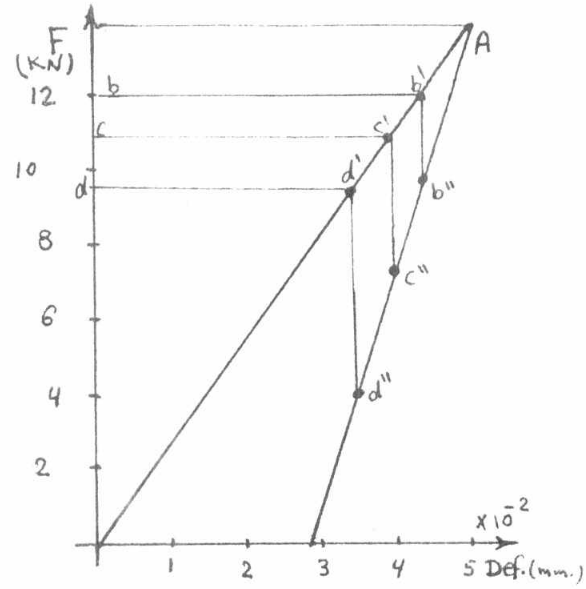


Fig.14.The screw diagram, case II, where $C_B < 0$, for aluminum gasket with W=2o mm., t=3 mm.

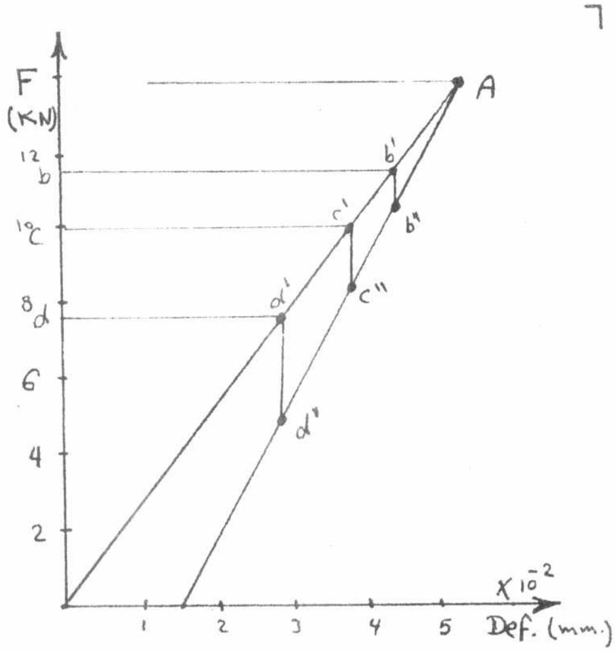


Fig.15.The screw diagram, case II, where $C_B < 0$, for aluminum gasket with W=IO mm., t=3mm.

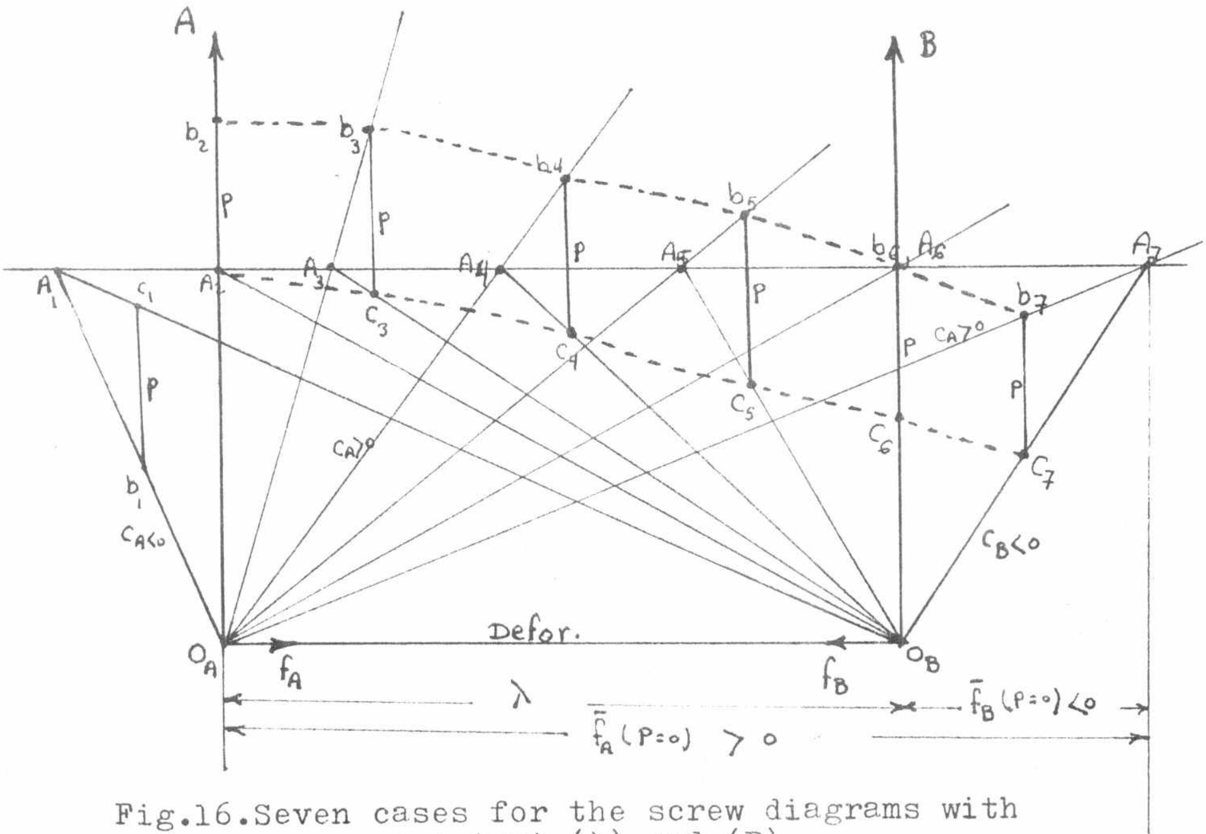


Fig.16.Seven cases for the screw diagrams with constant (λ) and (P).