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Bolted Joints Analysis Methods and Evaluation

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Abstract: *Calculation of accurate bolt forces is the primary requirement in many industries. All the theoretical calculations for bolt forces, includes many significant assumption based on idealized mechanical models. In this paper two models of flange joints were taken and analyzed forpretensionvariation due to internal temperature changes, And for theForces induced in the bolt due to the combined effect of external forces applied and internal temperature change. The results were utilized to gain insight into joint softening that arises from gradual, nonlinear opening of flange gap under external tension. Later these results were compared with the theoretical calculations, and our models allow relaxations for many assumptions in theoretical calculations.*

Keywords: *Bolt force,Heel gap, joint separation load, Pretension, Prying factor.*

I. Introduction

Industrial applications, structures are often uses the flange joints. The main objective of flange joint design is to provide adequate joint strength and stiffness and to minimize the fluctuating stresses induced in the bolted joint due to the thermal loads and external loads. Bolted joints are frequently analyzed using hand formulas that include many significant assumptions. In this paper, we evaluate several commonly used formulas by comparing their predictions to those of detailed finite element models. Our models allow the relaxation of many assumptions and enable a rational appraisal of these very important and widely used formulas. In addition, the FE results provide an understanding of the mechanics, which is just as important as the ability to make accurate predictions in specific cases.

The strength of the bolted joint is determined by analysis. Once the analysis is done it is clear that whether it is reaching the requirements or not. Based on the analysis results the changes have to be done in the design to increase the strength or stiffness of the joint. For the purpose of improvements to be done in the joint design, the bolt loads must be accurately calculated for the realistic design and service conditions.In this study, we analyze a bolt circle joining two flanges, to predict bolt load and joint stiffness for loads ranging up to the joint failure load. Thermal expansion is included. We compare the results from both analyses to those obtained from commonly used formulas

Using a detailed FE analysis for testing has advantages and disadvantages. The disadvantages are that the analysis may lack, or inadequately resolve, some significant effect present in the real hardware. The advantages are that the boundary conditions, material properties, geometry and loads can be precisely controlled, interesting quantities that may be impractical to measure can be recovered easily, and it is less expensive to obtain the insight that comes from observing a very large number of cases than it would be if testing were the sole approach. .

This section has two purposes:

1) to assess whether popular hand-calculation formulas for bolt and flange stiffness can be used to accurately predict the bolt load change due to thermal expansion, and
2) To gain insight into the joint mechanics when loads and displacements are perfectly axisymmetric, so that the lessons can be applied to the more realistic joint design shown.

The second purpose is arguably the more important, because bolt load changes due to thermal expansion are often only 10% or less of the total bolt load, which is of the order of typical pretension uncertainty. We study here a single, unconstrained joint consisting of a steel bolt-nut-washer set clamping two L shaped flanges of aluminum.

II. Nomenclature

Most symbols used in this document are defined below. In general, material and geometric properties are subscripted f for flange, b for bolt and nut, w for washer and m for the clamped members (flanges and washers) as a set.

P_{bolt}= bolt force,
P_{ini}= pretension given to the bolt,
P_{sep}= separation load of the joint,
Φ = joint stiffness ratio,
P_{ini}= initial pretension given to the bolt,
ΔP = change in the pretension,
k = stiffness,
δ = change in length,
ΔT = Change in the temperature,
f_{pyr} = Prying force
α = coefficient of thermal expansion
l = length,
A = cross section area,
E = young's modulus,
θ =compression frustumhalf-angle,
d = diameter of the bolt,
e_b =hole edge distance,
D = maximum diameter ofcompression frustum

III. Loads on the Bolted joint

A bolted joint is an assembly of bolt, nut, washers and flanges. In order to get the adequate joint strength pretension is to be applied on the bolt. Pretension compresses the washers, and flanges up to some extent. At the same time the pretension causes an elongation in the bolt. Pretension load is applied to the connection by stretching the fastener to a certain torque value. Torque is the turning moment of fastener or nut. Due to the many variables associated with the torque, a safety factor is calculated in determining torque value which will produce a pretension load lower than the yield point of that fastener.

Torque meter can be used to measure bolt tension. High pretension tension helps to keep joint tight, and increases the strength of a joint, and generates friction between parts to resist shear and improves the fatigue resistance of bolted connections. Generally 75% of proof strength is applied as the pretension. As a rule of thumb, the pretension should exceed the maximum load by 15% or so.

A realistic bolted joint, is subjected to thermal loads, external tensile or compressive forces. Bolted joints may be exposed to temperature changes of hundreds of degrees, as well as external loads. A change in temperature after fastener installation can induce significant stress when the bolt has a different coefficient of thermal expansion from the flanges.

The design considered here is bolt, nut and washers set made of structural steel, joining two flanges, the flanges were mounted or welded on shells circumference. However, first, the basic stiffness relationships between the fastener and the clamped material must be understood. The offset distance from the bolt centerline to the shell creates prying that obscures these basic relationships. External applied or constraint loads transmitted to the joint through the shells act at a distance from the bolt circle, tending to pry open the flanges rather than directly lifting them off one another. Even if the external load is aligned with the bolt axis using a fitting, the flanges will peel apart instead of gapping all at once, unless the fitting and flanges are unusually rigid relative to the bolt.

IV. Theoretical formulae

Most of theoretical analysis starts from an equation[1] of the form

$$P_{bolt} = \Phi P_{ext} + P_{ini} + \Delta P \qquad (1)$$

From the above equation (1) in the absence of external load the change in bolt length (in the grip) is equal to the sum of change in the lengths of washers and flanges. This leads to

$$\delta_b = \alpha_b l_b \Delta T + \frac{l_b}{E_b A_b}\Delta P \qquad (2)$$

$$\delta_f = \alpha_f l_f \Delta T - \frac{l_f}{E_f A_f}\Delta P \qquad (3)$$

$$\delta_w = \alpha_w l_w \Delta T - \frac{l_w}{E_w A_w}\Delta P \qquad (4)$$

From the equation (2), (3) and (4)

$$\delta_b = \delta_w + \delta_f$$

On solving the above

$$\Delta P = \frac{(2\alpha_f l_f + 2\alpha_w l_w - \alpha_b l_b)\Delta T}{\frac{l_b}{E_b A_b} + 2\frac{l_f}{E_f A_f} + 2\frac{l_w}{E_w A_w}} \qquad (5)$$

In the equation (1) Φ is the joint stiffness ratio. For the calculation of the stiffness several methods were proposed till now. Out of that we use two methods those are:

4.1 Method (1)

This method is from mechanical design textbook of Shigley:

I) Stiffness of any member[2] is:

$$k = \frac{AE}{l} \qquad (6)$$

II) Stiffness of a member in a bolted joint based on simple approach using fixed cone angle is[2]

$$k = \frac{\pi E_f d \tan\theta}{ln \frac{(l_f \tan\theta \ +D-d)(D+d)}{(l_f \tan\theta \ +D+d)(D-d)}} \qquad (7)$$

Using the equations (3) and (4)

$$A_f = \frac{\pi l_f d \tan\theta}{2 ln \frac{(l_f \tan\theta \ +D-d)(D+d)}{(l_f \tan\theta \ +D+d)(D-d)}} \qquad (8)$$

4.2. Method (2)

This method is that of Juvinall, this is also based on compression frustum

$$A_f = \frac{\pi}{4}\left[\left(\frac{3d_b \tan 30°}{2}\right)^2 - d_b^2\right] \quad (9)$$

$$k = \frac{\pi E}{4l}\left[\left(\frac{3d_b \tan 30°}{2}\right)^2 - d_b^2\right] \qquad (10)$$

The equation (1) is applicable when the line of action of forces coincides with the axis of the bolt. In this work the external load acting on the flanges. The line of action of forces is not coinciding with the axis of the bolt. Some prying is induced in the joint. For this purpose we modify the equation (1) by inserting the prying factor in the equation. Prying factor is a multiplier on the bolt force due to the offset bolt centerline.

$$P_{bolt} = \Phi f_{pry} P_{ext} + P_{ini} + \Delta P$$

$$= \frac{k_b}{k_b + k_f + \frac{2\, k_b\, k_f}{k_w}} f_{pry} P_{ext} + P_{ini} + \Delta P \qquad (11)$$

$$f_{pyr} = \frac{\frac{t_f}{2} + w_b}{e_b} \qquad (12)$$

In actual joint, joint separation occurs gradually as the flanges peel apart starting at the heel. But in the theoretical calculations separation occurs at all once.

$$P_{sep} = \frac{(P_{ini} + \Delta P)(k_b + k_f + \frac{2\, k_b\, k_f}{k_w})}{f_{pyr}\,(k_f + \frac{2\, k_b\, k_f}{k_w})} \qquad (13)$$

Once the theoretical value of bolt force reaches the P_{sep} , thus the bolt force is

$$P_{bolt} = \frac{k_b}{k_b + k_f + \frac{2\, k_b\, k_f}{k_w}} f_{pry} P_{ext} + P_{ini} + \Delta P, \qquad P_{ext} \le P_{sep}$$

$$P_{bolt} = f_{pry} P_{ext}, P_{ext} \ge P_{sep} \qquad (14)$$

V. Flange joint model

Actual bolted joints are many in types; here we are taking one common type of joint design is a flange connection between two cylindrical shells or housing. There are numbers on the circumference of cylinder in a cylindrical shell assembly. For the

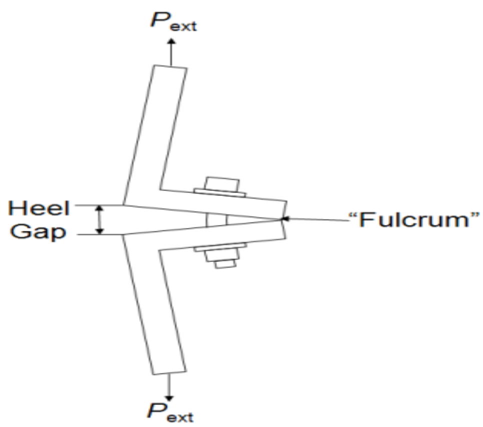

Figure (1): two dimensional model of flange joint assembly.

Analysis purposes it is sufficient to model one flange joint with carefully chosen boundary conditions. Here we have taken standard M10 bolt of class 9.8, and nut with pitch 1.5mm. Flanges are of L shape. The bolt, nut and washers are made of structural steel, and flanges are made of aluminum.

Table (1): material properties

Material	E	α	ϑ
Structural steel	2×10^{11} Pa	1.2×10^{-5} /°c	.3
Aluminum	0.71×10^{11} Pa	2.3×10^{-5} /°c	.33

Bolt and nut are modeled as per Metric Threads. The bolt belongs to class 9.8 the proof strength of bolt is 650 MPa, Yield strength is 720 MPa, and the tensile strength is 900 MPa.

VI. Finite element analysis procedure

The detailed finite element analysis for a bolted joint presented is exemplified in the following phases:

• The first phase is modeling the joint using CREO software. The model geometry was generated using the same software and then imported as a neutral file in ANSYS WORKBENCH. Geometric details, such as chamfers, radii of connection have only a local influence on behavior of the structure.

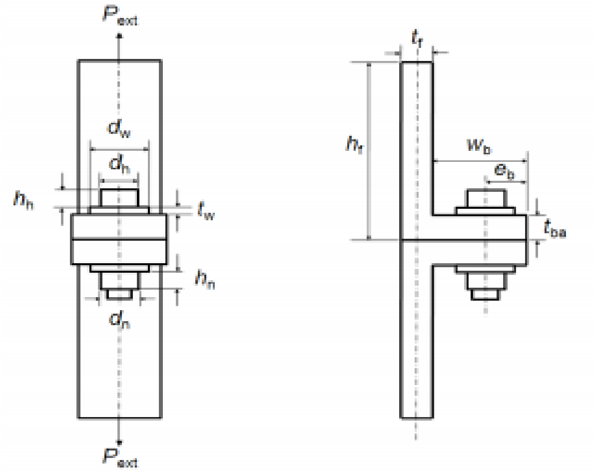

Figure (2): dimensions of the flange joint:

t_f	12.7mm	h_n	8.4mm
t_w	2.8mm	t_{ba}	12.7mm
w_b	31.3mm	h_f	100mm
w_b	31.3mm	e_b	15.65mm
h_h	6.85mm	d_b	10mm

.• Next, the prepared geometric structure is reproduced by finite elements. The finite elements are connected by nodes that make up the complete finite element mesh. Each element type contains information on its degree-of-freedom set (e.g. translational, rotational, thermal), its material properties and its spatial orientation (1D-, 2D-, 3D-element types). The mesh was controlled in order to obtain a fine and good quality mapped mesh. The assembly had 23519 nodes and 12714 elements.

• In order to solve the resulting system equation, boundary and loaded conditions are specified to make the equation solvable. These flanges were given an axial load, bolt was given a pretension of 28275 N, and total joint was subjected to a thermal load 110°c. Pretension is applied in Z axis direction.

• The last phase is interpreting the results.

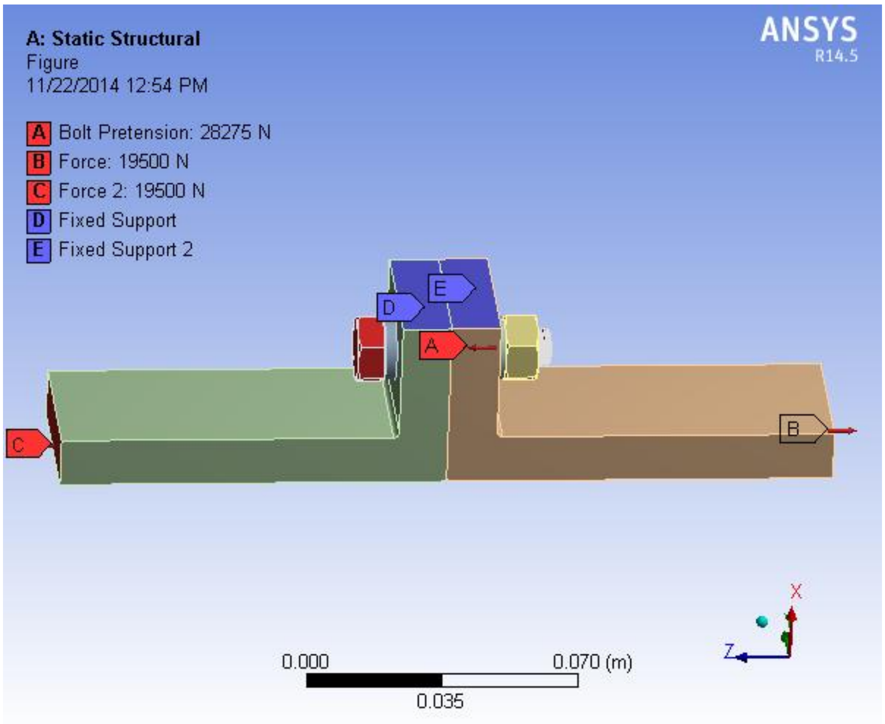

Figure (3): loads and supports given on long flange joint assembly.

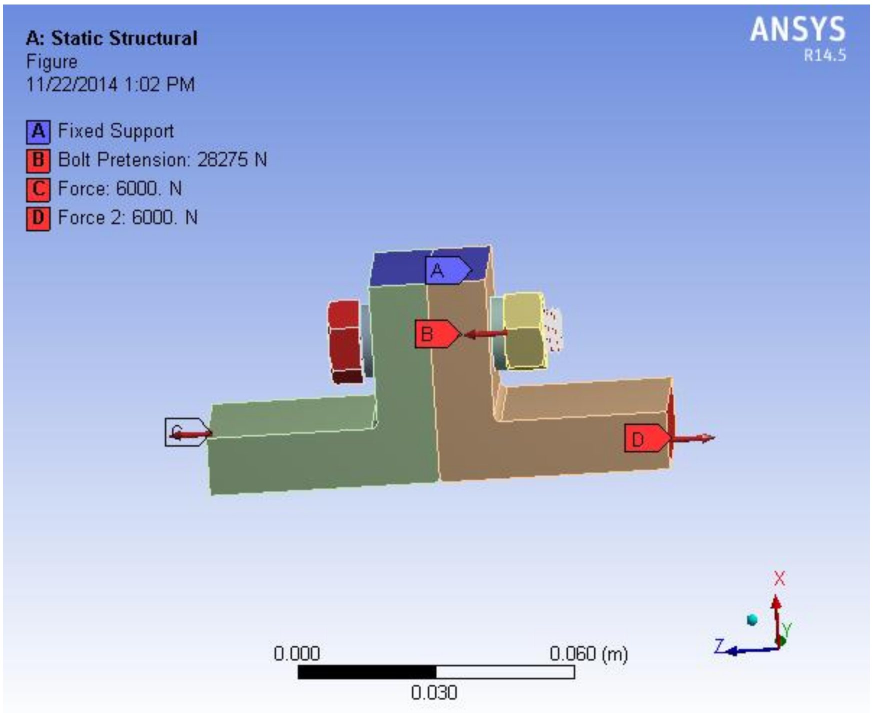

Figure (5): loads and supports given on short flange joint assembly.

VII. Results

Fig.(6) shows that results were plotted for short flange assembly bolted joint. The plot is for bolt forces versus external load for an initial pretension of 28275 N. It is clear from the plot, external load reaches near to the given pretension value, Bolt force reaches the yield strength of bolt. After this bolt forces crosses the elastic limit, that is yield criteria, the joint behavior after yielding is beyond the limitations of this study.

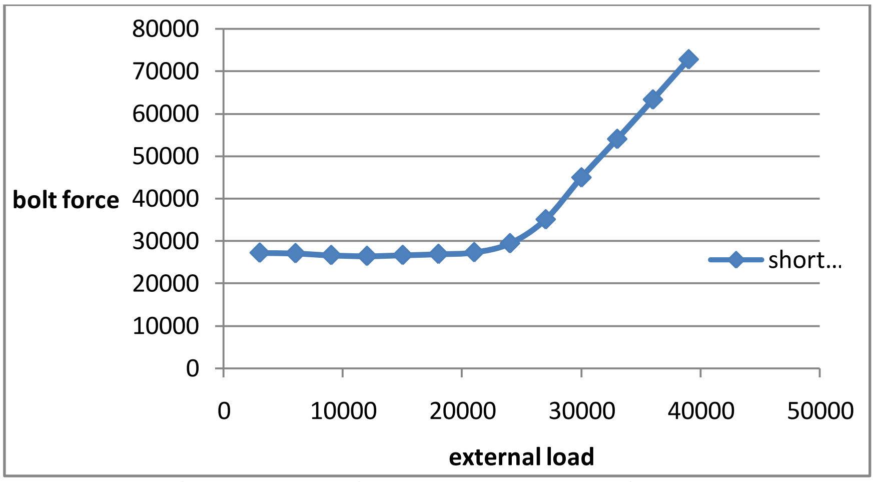

Figure (6): bolt force developed for external load in short flange model assembly

Fig.(7) shows the results of long flange assembly. From the graph it is clear that the bolt forces increases gradually unlikely in the theoretical calculations. In the Fig.(8) the bolt forces of short and long flange were compared for the purpose of better design of joint. From the graph it is very clear that there is no much difference in the bolt forces induced in the short and long flanges.

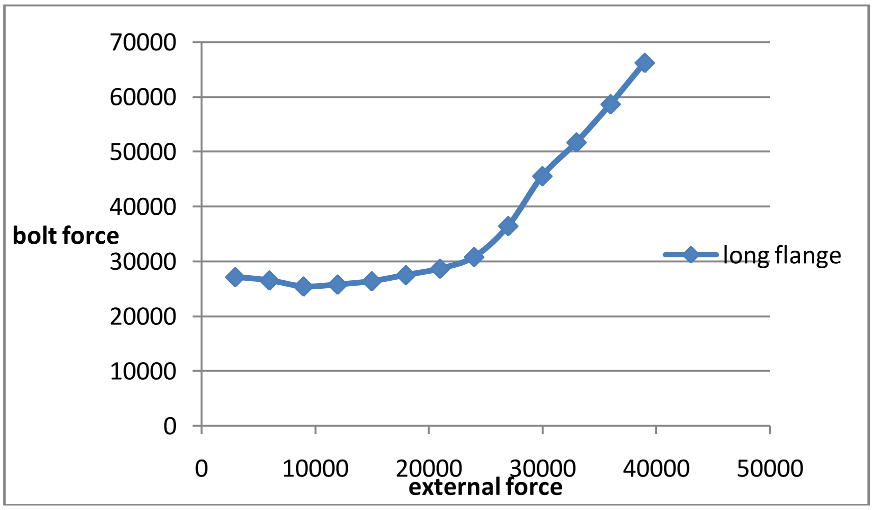

Figure (7): bolt force developed for external load in long flange model assembly**.**

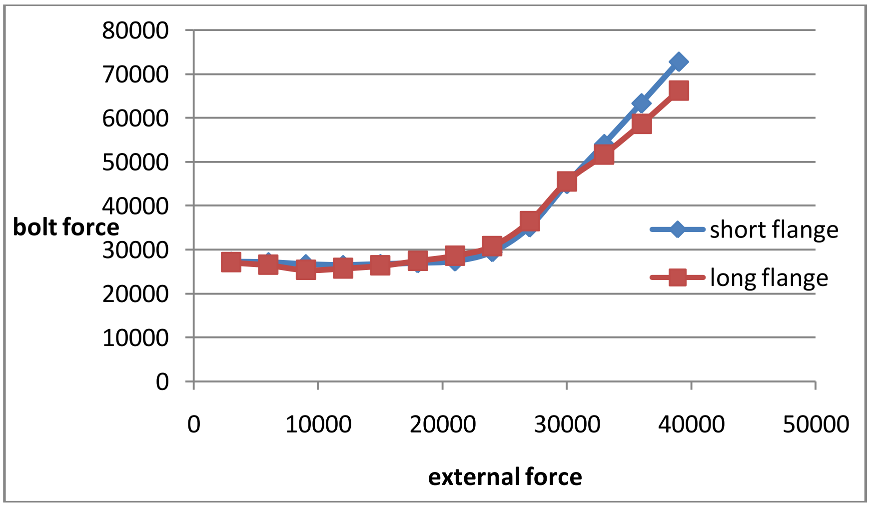

Figure (8): comparison of bolt forces developed in long and short flange assembly models.

The theoretical calculations method (1), and method (2), was used to find bolt forces. Later on the results of theoretical calculations and finite element analysis were compared. The comparison is shown in the fig.(9) and fig.(10) and it was showing that the theoretical calculations estimate a sudden change in the slope of the bolt forces curve. That sudden change in the slope of the curve, which is at the joint separation load of 19000 N. there is a steep increase in the curve with a slope of 2.405. Where as in the finite element analysis results slope of the curve is lesser than the theoretical value. It is observed thatthe theoretical values are 10 to 20% greater than the FEM results.

It is very difficult to measure the pretension applied on bolt, so there may be lot of uncertainties in applying the pretension to the bolt. If any pretension uncertainty exists in the joint, the hand calculations may over predict or lesser estimates the bolt force. This study calculated the bolt forces with hand formulae, for a pretension of ±25% to the actual value. At the same time finite element analysis was also conducted for a pretension of ±25% to the actual value.Both finite element analysis and theoretical calculations were compared. Comparisons were shown in Fig.(11) and Fig.(12). The observations are stating that the increase in the pretension tends to overestimate the bolt force.

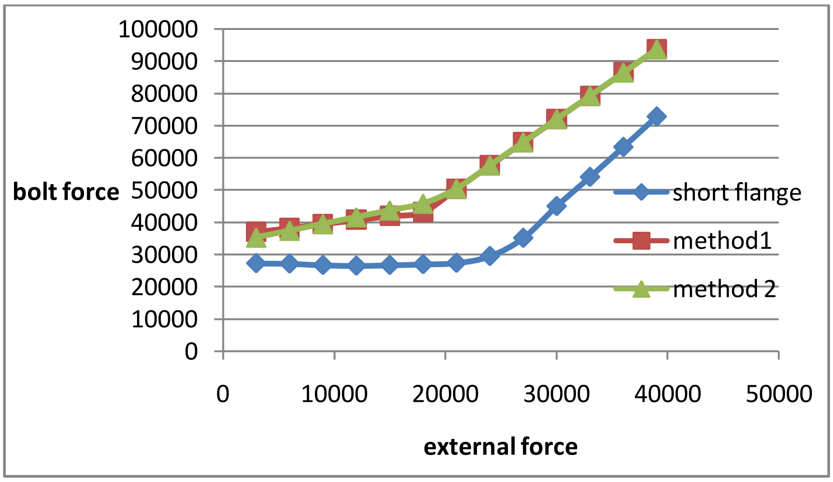

Figure (9): comparison of bolt forces developed in long and short flange assembly models with theoretical calculations method (1) and (2).

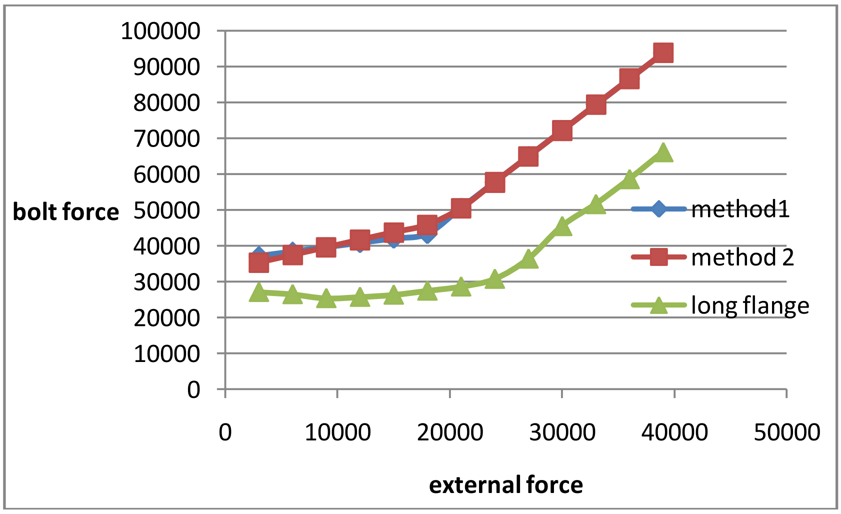

Figure (10): comparison of bolt forces developed in long flange assembly models with theoretical calculations method (1) and (2).

From the observations it was clear that the theoretical values were giving a closer fit to the finite element analysis results if the prying factor is reduced. It was shown in the Fig.(13) that a lesser prying factor value gives a closer fit to the finite element analysis values.

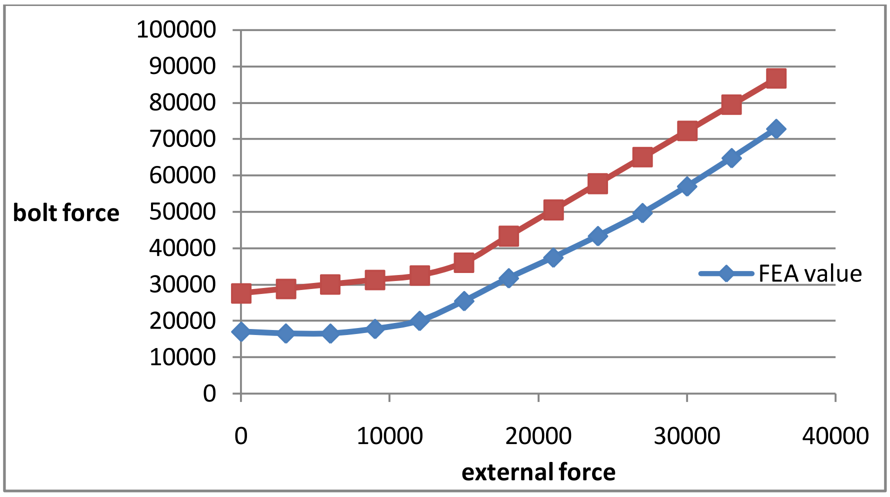

Figure (11): comparison of bolt forces developed in long flange assembly models with method 1 for the pretension value of 20000N.

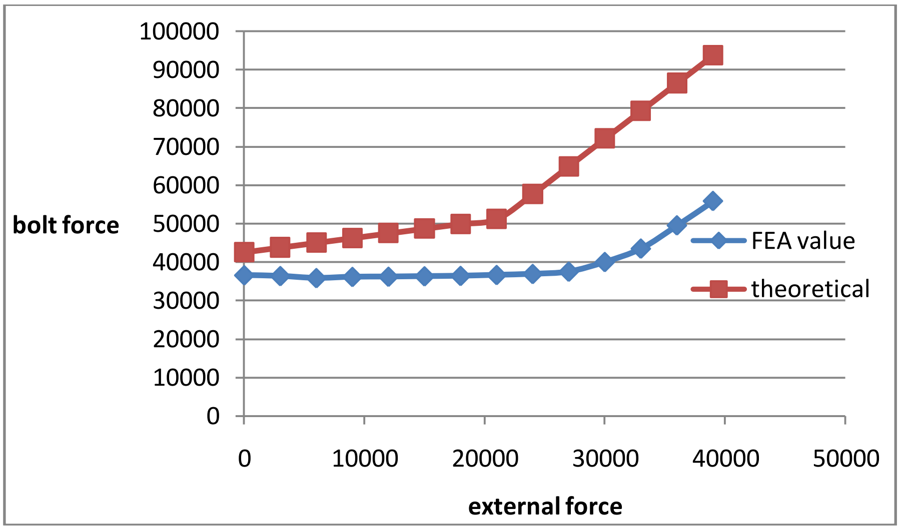

Figure (12): comparison of bolt forces developed in long flange assembly models with method 1 for the pretension value of 35000N.

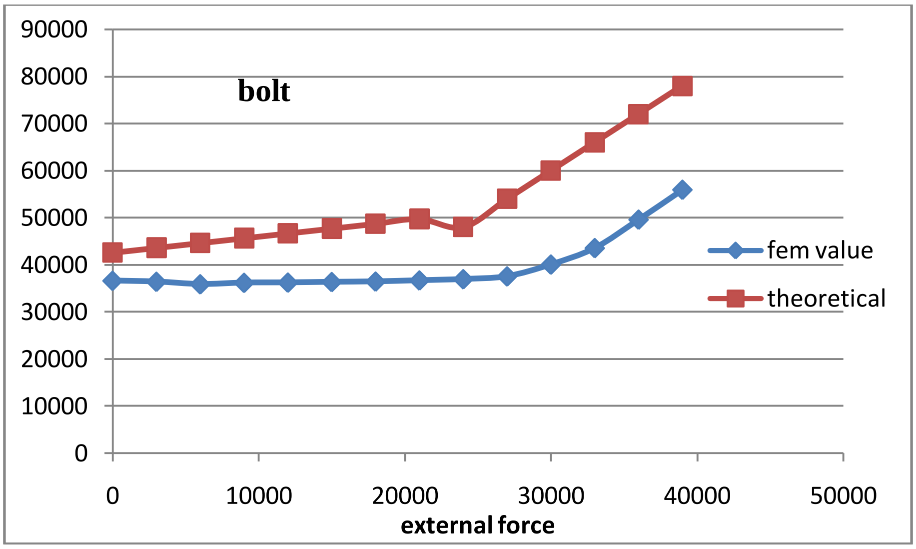

Figure (13): comparison of theoretical values with prying factor 2 to the finite element analysis values.

VIII. Conclusions

1) Theoretical calculations shows that the initial pretension applied is not having any effect on the calculation of change in the pretension due to the temperature change.
2) It is clear from the results plot; increase in the pretension increases the amount of overestimate of bolt forces.
3) Length of the flange does not have any effect on the bolt force estimation either in theoretical calculations or in the finite element analysis results.
4) Decreasing in the prying factor in the theoretical calculations gives a closer fit to the theoretical and experimental values.
5) Stiffness of the total joint is based on the initial geometry in the theoretical approach. But when the joint starts separating stiffness decreases with increase in the heel gap.
6) Because there are many assumptions in the theoretical calculations, theoretical calculations are overestimating the results up to 10-20% of actual values, for the complicated design of flange joints.

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