EXPERIMENTALLY AND NUMERICALLY VERIFIED ANALYTICAL MODEL FOR PREDICTION OF PROCESS FORCE IN NOVEL METHOD FOR PIPE-TO-FLANGE JOINING

[1]JAN HENRIKSEN, [2]MICHAEL RYGAARD HANSEN, [3]FREDRIK CHRISTOPHER THRANE

[1,2,3]University of Agder, University of Agder, ICR Integrity Ltd,
E-mail: [1]jan.henriksen@uia.no, [2]michael.r.hansen@uia.no, [3]fredrik.thrane@icr-world.com

Abstract - The process force during the joining of a pipe and flange by cold forming using a specially designed hydraulic tool is investigated. Assuming a perfectly plastic material behavior, an analytical model is developed based on general plasticity. The method is restricted to basic and easily obtainable material and geometrical data. Results obtained from an experimentally verified numerical axisymmetric FE simulation of the connection process is utilized for comparison. For a perfectly plastic model, the results given by the analytical model are in excellent accordance with simulated results with respect to maximum actuation force. When introducing strain hardening without access to true stress-strain data the accordance remains good, however, less accurate. Overall, it is concluded that the developed analytical model is a highly useful tool to estimate the maximum actuation force required to connect the pipe to the flange.

Index Terms - Flange, Cold forming, FE simulation

I. INTRODUCTION

To avoid the safety issues associated with the use of welding in offshore settings, a novel method has been developed based on plastic radial expansion that allow pipes and flanges to be connected. The method was originally presented by [2]-[4]. In those papers, the connecting process was described in detail and subjected to a study that showed the usefulness of finite element (FE) analysis when estimating deformations and forces during and after the cold forming. The process set-up for this new method is illustrated in Fig. 1. The process tool is segmented and positioned inside the pipe that, in turn, is positioned inside a modified flange with two machined grooves.

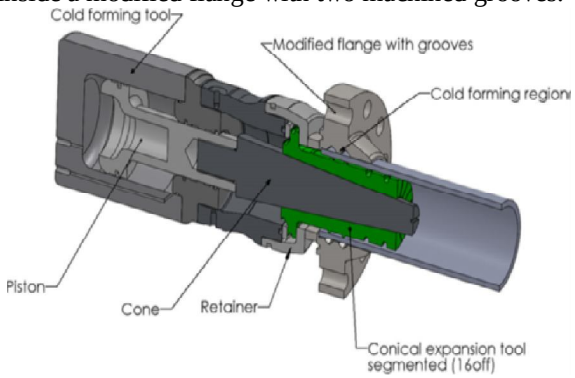


Figure 1 Main components in connecting process

The segments will expand radially when the cone is pushed forward by the hydraulically actuated piston. Each segment has two ridges aligned with the grooves of the flange. During the expansion, the ridges will force the pipe to deform into the grooves. For more details, refer to [3]. Due to the complex nature of the connection process numerical simulation models is generally preferred to capture the effect of complex part geometry, nonlinear material behavior and complex contact condition. Reference [3] and [4] analyzed both the process parameters and axial strength of the connection by use of numerical simulations. Reference [1] used a three-dimensional FE model to evaluate the novel technology with emphasis on both the connecting process and the pipe pull out (PPO) capacity.The focus of the present paper is whether the novel cold forming process can be satisfactorily predicted by an analytical model. This will provide the user with a first prediction of the required tool pressure without setting up a complex and time consuming numerical simulation or, even more demanding, experimental testing.

II. REFERENCE SYSTEM

To verify the analytical model a 6-inch schedule 80 pipe-flange connection is considered. This connection is a typical connection size and has, previously, been subjected to both experimental and numerical studies in [3] and [4] that clearly shows the ability of numerical finite element models to accurately predict the required process force. Therefore, an experimentally verified numerical model will be used as reference for the analytical model since the numerical model also allows for the investigation of an elastic perfectly plastic material behavior. The material quality for the pipe/flange is X52Q/ A694 F52 with yield stress and Ultimate tensile strength respectively of 435MPa/360MPa and 539MPa/455MPa. To serve as an input to the numerical simulations, uni-axial tensile testing was performed for the pipe and flange material to determine the plastic strain/true stress curves. The material testing also indicated plastic isotropy for both the pipe material as well as the flange material. Two models were built up for verification purpose of the analytical model; one with an elastic-perfectly plastic material model and one utilizing stress-strain curves found by tensile testing. The results from the tensile testing experiments were corrected for triaxial stress

condition in the necking zone of the test specimen.

III. ANALYTICAL MODEL

Initially, an elastic-ideally plastic material model is considered for the pipe. Further the flange is assumed to be rigid. However, only the plastic deformation of the pipe is included in the model. Elastic deformation of the flange is anticipated to have minor effect on the deformation work. The deformation energy for the pipe may be divided into the below given contributions.

- Circumferential plastic deformation of the pipe.
- Plastic hinges to approximate bending of the pipe in the flange groove areas.
- Longitudinal plastic deformation of the pipe.
- Thickness reduction of the pipe in the segment ridge area.

This is further presented in Fig. 2 where the plastic deformation in circumferential direction is indicated to be out of the plane of the figure. To avoid including the deformation energy associated with the local thickness reduction, the other deformation energies are computed based on the locally deformed pipe section. The deformation of the pipe in the flange groove area is approximated by the model presented in Fig. 2 and in Fig. 3. The radius of the pipe, R, is measured from the axial centerline to the pipe wall center for the undeformed pipe. The radial travel of the segment tool and the local contact deformation are denoted by d_T and d_C, respectively. This yields a reduced thickness at the maximum deflection:

$$t_C = t - d_C \tag{1}$$

The plastic hinges are positioned in the centerline of the deformed wall thickness and therefore the radial deformation of the center hinge and plastic hinge rotation are:

$$d_{ch} = d_T + \frac{t_C}{2} - \frac{t}{2} = d_T - \frac{d_C}{2} \tag{2}$$

$$\theta = tg^{-1}\left[\frac{2 \cdot d_{ch}}{b}\right] \tag{3}$$

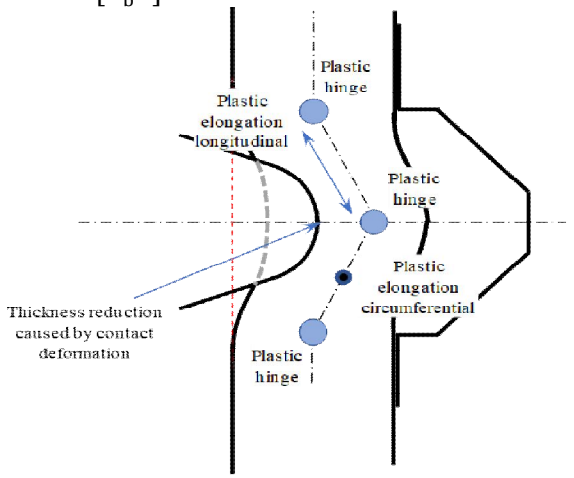


Figure 2 Deformation model of a single pipe flange connection

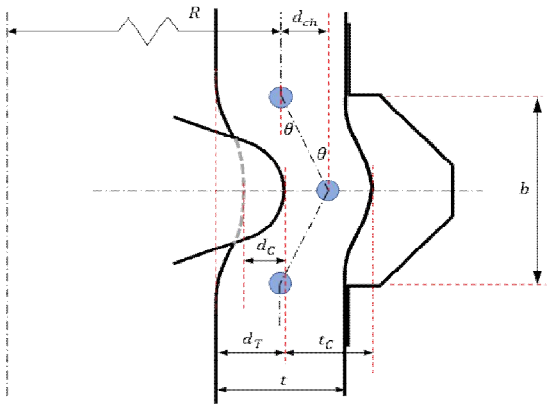


Figure 3 Geometric model parameters associated with plastic hinges.

The force required to bend the locally deformed pipe is obtained from the equilibrium of the external and internal energy. The external energy is given by the radial force supplied by the segment tool multiplied by the radial travel of the center hinge:

$$E_{ext} = F_r \cdot d_{ch} \tag{4}$$

The internal energy is the sum of energy required to deform the plastic hinges and extend the conical section in both circumferential and longitudinal direction. The internal energy can be formulated as the sum of the plastic energy required to deform the plastic hinges E_h, the plastic energy required for the circumferential elongation E_c, and the plastic energy required for the longitudinal elongation E_l:

$$E_{int} = n \cdot (E_c + E_l + E_h) = 2 \cdot (E_c + E_l + E_h) \tag{5}$$

In (5) n=2 is the number of grooves in the flange. The internal energy associated with the local deformation is not included because the external energy is based on the bending of a pipe section with reduced thickness.

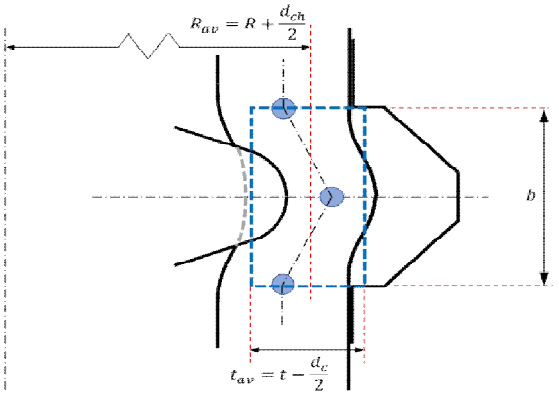


Figure 4 Geometric model parameters associated with radial and longitudinal plastic elongation.

The strain energy required to extend the radius of the deformed region assuming a simplified rectangular cross section, see Fig. 4, is given by:

$$E_c = \Delta_c \cdot A_c \cdot S_Y \tag{6}$$

In (6) Δ_c is the circumferential increase, S_Y is the yield stress and A_c is the area of the simplified cross section defined by an average radius and an average thickness:

$$\Delta_c = 2 \cdot \pi \cdot R_{av} - 2 \cdot \pi \cdot R = \pi \cdot d_{ch} \quad (7)$$

$$A_c = t_{av} \cdot b \quad (8)$$

Similarly, the strain energy required to extend the deformed region in the longitudinal direction is given by:

$$E_l = 2 \cdot \Delta_l \cdot A_l \cdot S_Y \quad (9)$$

In (9) Δ_l is the longitudinal increase obtained between end hinge and center hinge, and A_l is the area of a simplified ring cross section with the same thickness as A_c:

$$\Delta_l = \sqrt{\frac{b^2}{4} + d_{ch}^2} - \frac{b}{2} \quad (10)$$

$$A_l = 2 \cdot \pi \cdot \left(R + \frac{d_{ch}}{2}\right) \cdot \left(t - \frac{d_c}{2}\right) \quad (11)$$

Assuming ideal plastic material model and a two-dimensional sheet approximation the plastic moment, see Fig. 5, can be computed as:

$$M_p^{(eh)} = \frac{\pi \cdot R \cdot t^2}{2} \cdot S_Y \quad (12)$$

$$M_p^{(ch)} = \frac{\pi \cdot (R + d_{ch}) \cdot t_C^2}{2} \cdot S_Y \quad (13)$$

The energy required to deform a plastic hinge is the plastic moment, M_p, multiplied with the bending angle denoted θ. It should be noted that θ is the rotation of the end hinges whereas the total rotation of the center hinge is 2·θ.It is seen from experiments and numerical analyses that the second plastic end hinge of the groove close to the pipe end is only partially rotated. Therefore, we can introduce two extreme values of the plastic hinge energy of a groove connection.The maximum corresponds to full rotation of both end hinges, $E_h^{(max)}$, and the minimum, $E_h^{(min)}$, corres-pond to full rotation of only a single end hinge.

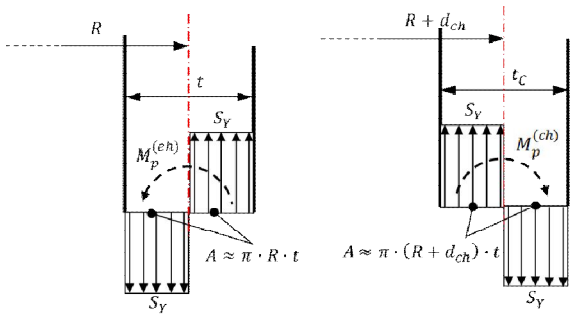


Figure 5 Plastic moment in end hinge (left) and center hinge (right).

$$E_h^{(max)} = \pi \cdot (R \cdot t^2 + (R + d_{ch}) \cdot t_C^2) \cdot S_Y \cdot \theta \quad (14)$$

$$E_h^{(min)} = \pi \cdot (0.5 \cdot R \cdot t^2 + (R + d_{ch}) \cdot t_C^2) \cdot S_Y \cdot \theta \quad (15)$$

It is reasonable to assume that the actual value will lie between these values. Setting the external energy equal to the internal energy by combining (4) and (5) yields an analytical expression for the required radial force for a two-groove connection:

$$F_{r,max}^{(ph)} = \frac{2 \cdot \left(E_C + E_l + E_h^{(max)}\right)}{d_{ch}} \quad (16)$$

$$F_{r,min}^{(ph)} = \frac{2 \cdot (E_C + E_l) + E_h^{(max)} + E_h^{(min)}}{d_{ch}} \quad (17)$$

The radial force is a non-linear function of the yield stress, the basic geometry of the pipe flange connection, (R,b,t), the prescribed radial tool segment travel, d_T, and the local contact deformation, d_C. Of these values, the local contact deformation is easily the most elusive.

IV.COMPARISON

The radial forces obtained from (16) and (17) are compared with results from the FEM modelassuming perfectly plastic material behavior described earlier. The local contact deformation of the analytical model is simply adopted from the FEM analysis where a local contact deformation of d_C=3.425 mm for a segment expansion of d_T=5 mm was observed. This yields a pipe wall thickness reduction from t=11.25 mm to t_C=7.825 mm. Introducing the local deformation ratio:

$$\mu = \frac{d_C}{d_T} \quad (18)$$

this corresponds to μ=0.685. Fig. 6 presents the simulated segment expansion force obtained for the elastic perfectly plastic model. Fig. 7 also includes the simulated segment expansion force utilizing the experimentally obtained stress-strain curves. The required radial force is obtained from the simulation results for the full segment expansion that corresponds to point D in Fig. 6 and Fig. 7.

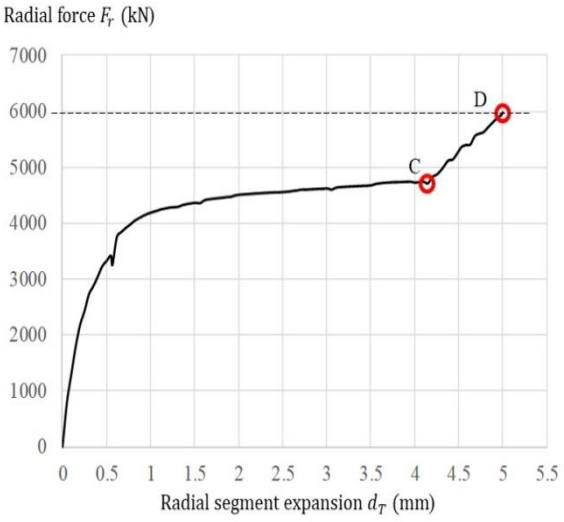


Figure 6 Radial force vs. radial segment expansion for perfectly plastic FEM model.

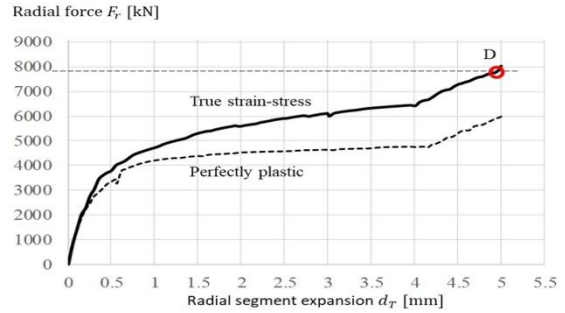


Figure 7 Numerical and analytical computations of radial force.

In Table I the required radial force is listed for different models. There is a good correlation between the results from the analytical model, equations (16) and (17), and the numerical values when pipe wall thickness reduction is included. In fact, the numerical result lies almost exactly on the average value of the maximum and minimum values of the interval which must be considered a coincidence, however, it is safe to conclude that the interval of the analytical model is valid for this situation in which the second plastic end hinge of the groove close to the pipe end has a rotation between 0° and the full rotation, θ=19°. Also, an assumption of no pipe wall thickness reduction, μ=0, clearly overestimates the segment expansion force interval and represents an overly conservative computation of the radial force.Strain hardening should be included. However, plastic strain-true stress material data is generally not available, and the effect of strain hardening is proposed to be based on the classical one-axis tensile information where a strain hardening ratio ζ equal to the ultimate tensile strength divided by the yield stress is proposed. Hence, the radial segment force adjusted for strain hardening is given in (19).

$$F_{r,a*}^{(ph)} = \zeta \cdot F_{r,a}^{(ph)} \; a = \begin{cases} min \\ max \end{cases} \qquad (19)$$

In Table II the required radial force is listed for the FEM model using the true stress-strain curve and (19). The interval predicted by the analytical model contains the result from the numerical model, however, the fit is less convincing as compared to the elastic-perfect plastic model which is not surprising because of the simplified expression for strain hardening.

Table I Radial force for different perfectly plastic material models

Model	μ [−]	Radial force [kN]
FEM, perfectly plastic	-	5990
$F_{r,max}^{(ph)}$	0	9030
$F_{r,min}^{(ph)}$	0	8340
$F_{r,max}^{(ph)}$	0.685	6410
$F_{r,min}^{(ph)}$	0.685	5680

Table II Radial force for different models including strain hardening

Model	μ [−]	Radial force [kN]
FEM, true stress strain	-	7780
$F_{r,max*}^{(ph)}$	0.685	7950
$F_{r,min*}^{(ph)}$	0.685	7040

In general, it is concluded that the analytical model as presented in (19) can predict an interval that contains the value obtained by a realistic FEM model if the local contact deformation can be estimated.

CONCLUSION

An analytical expression of the radial force required to deform a pipe into machined grooves in a modified standard flange is presented in this paper. The main criteria for the analytical expression is to be independent of the material flow curve but rather to rely on the material properties normally given in a material certificate which is the yield stress and ultimate tensile strength. In this way, the analytical expression allows the radial force to be estimated without use of skill intensive numerical simulations or advanced material testing. Comparing the results from the analytical expression assuming perfectly plastic material behavior with results obtained from the numerical simulation the accuracy is so high that it is only limited by the reliability of the input data. As expected the accuracy is poorer when including the simplified expression for strain hardening. However, the results from the analytical model are still considered to be in good accordance with simulated values.

In conclusion, the proposed analytical model must be considered an excellent combination of simplicity and accuracy when estimating the required tool radial force to deform the pipe into the grooves without setting up a complex and time consuming numerical simulation. Further work is to include the local contact deformation of the pipe adjacent to the segment ridges in the analytical model. In addition, the accuracy of the analytical expression will be tested out for different pipe-flange dimensions and configurations.

ACKNOWLEDGEMENT

This research was supported by the Research Council of Norway, Grant No. ES468846 User-driven innovation project (PETROMAKS) and Quickflange Technology AS. Both are gratefully acknowledged for funding of the project.

REFERENCES

[1] Barsoum, A.M. Khalaf,"Evaluation of a Pipe-Flange ConnectionMethod Based on Cold Work", ASME J. Pressure Vessel Technol., 137(6), p.061407-(1-8), December2015

[2] J. Henriksen, M.R. Hansen, «FEM Simulation of Cold Deforming Pipe Into Flange", 26th Nordic Seminar on Computational Mechanics, pp. 93-96, October 2013

[3] J. Henriksen, H.O. Nordhagen, H.N Hoang, M.R. Hansen, and F.C. Thrane, «Numerical and experimental verification of new method for connecting pipe to flange by cold forming" Journal of Materials Processing Technology 220, pp. 215-223, June 2015

[4] J. Henriksen, M.R. Hansen, and F.C. Thrane, «Axial Load Capacity of ColdFormed Pipe Flange Connection", Journal of Pressure Vessel Technology, Vol. 139, October 2017/051201, October 2017

★★★