Modeling of preloaded threaded pipe connections

Jeroen Van Wittenberghe, Patrick De Baets, Wim De Waele
Department of Mechanical Construction and Production, Ghent University
St.-Pietersnieuwstraat 41, 9000 Ghent – Belgium
email: jeroen.vanwittenberghe@ugent.be, patrick.debaets@ugent.be, wim.dewaele@ugent.be

Abstract: **In this paper a modeling method to perform parametric studies on preloaded threaded connections is presented. The method uses a non-linear 2D axisymmetric finite element model, and is illustrated by a parametric study of an API Line Pipe connection. The method was used to quantify the influence of the coefficient of friction, the wall thickness of pin and box and the box recess length of the connection.**

Keywords: **parametric study, threaded pipe connection, finite element model, API Line Pipe**

I. INTRODUCTION

A. Outline

Threaded pipe connections are used to join pipelines as an alternative for welding and in applications where pipes should be frequently coupled and uncoupled. The connections consist of a male and female part, called respectively pin and box. To maintain a sealed and secure connection while being subjected to external variable loads, they are commonly preloaded. This can be done by using conical connections and tightening the pin and box with a specified torque, called 'make-up' torque. This make-up torque is different for every connection type and size. Due to the combination of the preload and external loads, together with the thread geometry, a complex multiaxial stress distribution develops over the connection.

The stress distribution depends on the coupling's geometrical parameters like pipe dimensions and thread type, but is also influenced by the contact interaction properties of the material of the connection. Resulting stress concentrations can initiate fatigue cracks and cause a premature failure of the connection. According to Griffin et al [1] the highest stress concentration under axial load can be expected at the last engaged thread (LET) of the pin.

The influence of the different geometrical and contact parameters on the connector's applicability and service life are not well known. In this study a newly developed program is presented that can be used to perform parametric finite element (FE) studies on threaded pipe connections. After a general introduction on FE modeling of threaded connections, the structure of the program will be presented. In the subsequent sections, the results of a parametric study on a preloaded API Line Pipe threaded connection are discussed.

B. Modeling of Threaded Connections

The simulation of threaded connections is characterized by two non-linearities. Firstly, high local stresses can appear, exceeding the material's yield strength even during the make-up stage. Hence, non-linear elastic-plastic material models are necessary. Secondly, surface interactions together with small sliding between pin and box result in non-linear contact behavior.

Starting from the 1980's the finite element method is used to model threaded connections. Since the computational performance of the computers in that time was rather limited, it was not possible to model full non-linear 3D models. Hence, hybrid models [2, 3] and 2D axisymmetric models [4] were used. With the hybrid modeling techniques, the global load distributions over the threads is analytically calculated and local stresses are determined using 2D axisymmetric finite element models of only a few threads. Results from both techniques were validated by photoelastic experiments. With increasing computing power, the hybrid method was completely abolished to use 2D axisymmetric models with finer meshes. Mid 1990's, more complex material models were introduced to model elastic-plastic material behavior [5-7] and better contact interaction properties were defined to allow sliding of the threads [8].

Nowadays threaded pipe connections are still commonly modeled using 2D axisymmetric finite element models with elastic-plastic material behavior and contact interaction of the threads [1, 9-13]. Despite the vast performance increase of computers over the last decades, full 3D models of threaded couplings still require very long calculation times due to the high number of nodes in the contact analysis. In addition to this, due to their complexity and multiple contacting surfaces, full 3D models tend to be less stable and to diverge from a solution more easily. Performed 3D simulations are generally simplified by using linear elastic material properties and using very coarse meshes [14], resulting in inaccurate stress distribution and useless stress concentration factors. Generally much more precise results can be obtained with 2D axisymmetric models. The disadvantage of the 2D simulations is that they neglect the helical shape of the threads and the runout region. However, it was shown by Chen and Shih [15] in the analysis of bolts and more recent by Zhong [16] for threaded pipe connections, that the results of 2D axisymmetric models are in good agreement with the results of accurate time-consuming full 3D models.

Defining correct contact interactions requires accurate knowledge about the interaction properties, like the coefficient of friction (COF), of the contacting interface. However, a variety of values for the COF are used. In ISO 10407-1 [17] values for the COF are specified between 0.06 and 0.14, mentioning a typical value of 0.08 when thread compounds are used. Guangjie et al [11] even use a lower value of about 0.02. In experimental studies friction values between 0.06 and 0.09

were obtained by Ertas et al [18] for tests on pipeline steel with different thread compounds and Santus et al [13] measured values of 0.15 during torsion tests on full scale threaded connections. As will be shown further, this variation of the coefficient of friction has a significant influence on the thread opening of the connection and correct values for the COF should be used.

II. MODELING STRATEGY

To be able to perform parametric studies to simulate the influence of different parameters on the connection's behavior, a new modeling method was developed using a combination of Matlab® R2008a and Abaqus® 6.8-1. This method was entitled *ThreadGen*© and its structure is illustrated in Fig. 1.

The input of the program consists of the connection's geometrical parameters, loading conditions, material and contact properties together with parameters concerning the numerical analysis like mesh size.

During the first stage of the program, a Matlab® program generates the coordinates of the connection geometry based on the connection input. These coordinates are then processed to generate two Python scripts that can be run by Abaqus® (*Connection.py* and *ConnectionResult.py*). Another Matlab® program generates a script *ConnectionOutput.m* that contains the data necessary to process the generated numerical results.

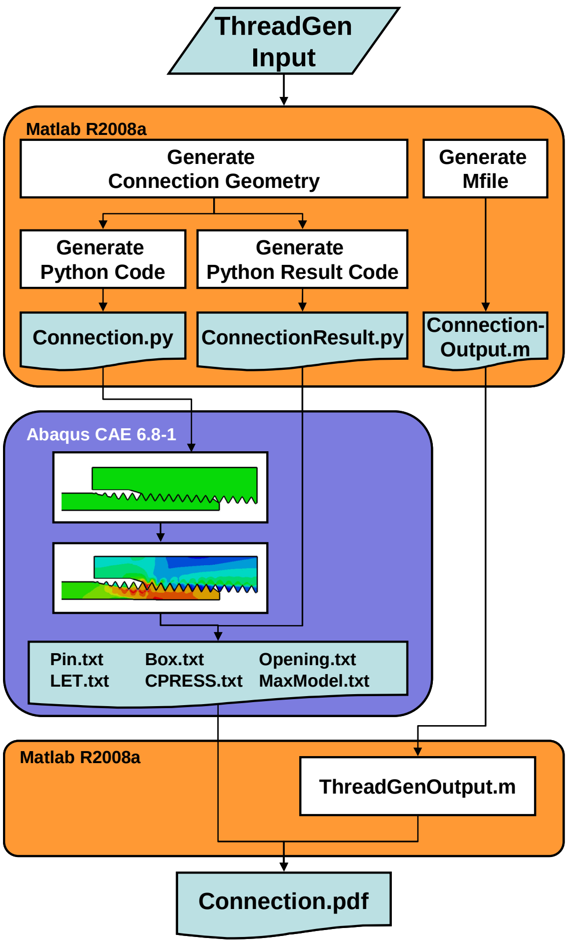

Fig. 1: Structure of the parametric program ThreadGen©

During the second stage, the Python scripts are run in Abaqus®. The script *Connection.py* generates the model geometry together with material, interaction, loading and mesh properties. The model is then analyzed and the second Python script *ConnectionResult.py* is run, which processes the results and selects the relevant data from the finite element simulations and generates specific output as txt-files and images.

In the final stage the output data is processed again by Matlab® to generate a pdf-document summarizing the selected results. This is done by the script *ThreadGenOutput.m* which uses the earlier generated script *ConnectionOutput.m*. The generated results in the txt-files can also be used for further detailed analysis.

When performing parametric studies multiple variations of a connection can be easily simulated by generating a batch of input scripts in the first stage. During the second stage, all connections are then automatically simulated and processed. In the third stage the generated output files are processed automatically.

III. FINITE ELEMENT MODEL

A. Model Geometry

ThreadGen© was used to perform a parametric study on an API Line Pipe connection. The standard connection according to API 5B specifications [19] is shown in Fig. 3 and consists of a female box that connects the threaded ends (called "pin") of two pipes. For the numerical model, only the section in the dashed rectangle is used. The resulting model is shown in Fig. 3. The modeled connection has a nominal size of 4", which corresponds to a pin with outside diameter of 114.3 mm and wall thickness of 6.0 mm. The box has an outside diameter of 132.1 mm and total length of 114.3 mm. The unthreaded pipe body of the pin has a length of 100 mm to eliminate boundary effects when an external tensile stress is applied at its free end.

The script to generate the connection geometry is built in such a way that the geometric parameters like pin diameter and wall thickness, box wall thickness, number of engaged threads and thread dimensions - thread pitch and height - can be easily adjusted. In this way all standard connection sizes and a wide range of modifications can be easily generated.

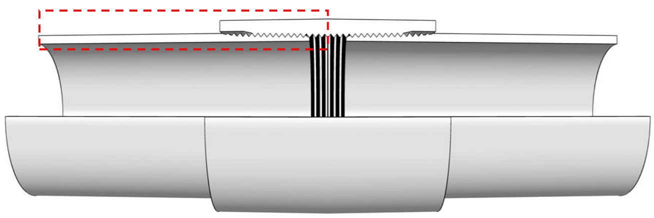

Fig. 2: Section view of an API Line Pipe Connection

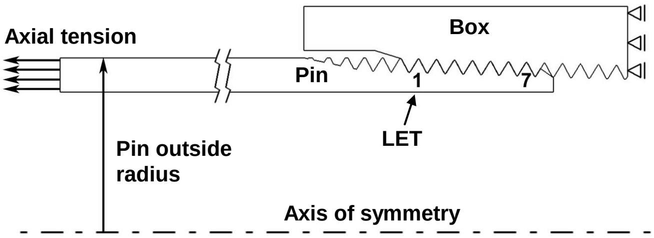

Fig. 3: 2D axisymmetric model of the API Line Pipe connection

The mesh of the FE model was determined through a mesh optimization study and is illustrated in Fig. 4. For the standard 4" connection, the pin consists of 14111 and the box of 5670 linear quadrilateral CAX4R elements. The global mesh size for both pin and box is 1 mm. A finer mesh with seed size 0.2 mm was used in the threads of the box. Since the box is a more rigid component than the pin, the box thread surfaces serve as the master elements in the contact analysis. The mating pin thread surfaces are defined as the slave elements and have an even finer mesh. To be able to study the local stress distribution, the thread roots of the pin are seeded with fifteen elements. The resulting mesh details of the standard API Line Pipe connection are shown in Fig. 5. Note that to avoid sharp edges at the thread crest and root a fillet was applied with a radius of 0.05 mm.

A multi-linear elastic-plastic material model with kinematic hardening for API grade B steel is used. This is the standard material for this type of connection. The model uses a Young's modulus of 208 GPa and a Poisson coefficient of 0.3. The material's yield strength is 241 MPa. The ultimate tensile stress value (true stress) is 521 MPa, the corresponding elongation is 23%. All values correspond to the properties of API steel grade B as specified by API 5L [20].

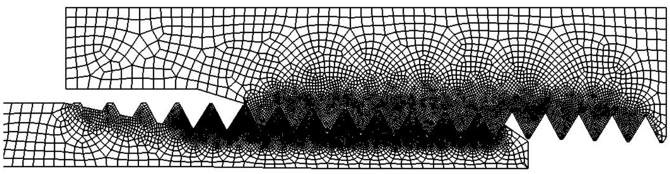

Fig. 4: Mesh of the FE model

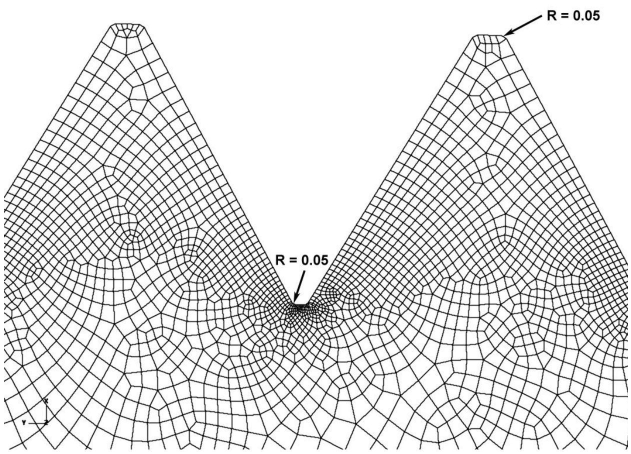

Fig. 5: Detailed mesh around thread root of the pin

B. Analysis of the standard connection

The analysis is carried out in two consecutive steps. In the first step the make-up of the connection is simulated by applying a certain radial overlap between pin and box in the model. This overlap corresponds to the number of make-up turns specified in API spec. 5B. The thread surfaces are then brought into contact using the interference fit option in Abaqus®. During the second step an additional axial tensile stress is applied, as shown in Fig. 3. The magnitude of this stress should be lower than the stress corresponding to the connection's pull-out strength, that can be estimated from the empirical formulas given by Clinedinst [21]. For the considered connection a value of 373 kN is calculated for the pull-out strength. This corresponds to a uniform axial tensile stress of 183 MPa in the pipe body of the pin. Since the thread opening increases drastically at loads near thread pull-out, the calculations tend to diverge from a stable solution. For this reason the applied axial tensile stress is limited to 150 MPa.

The resulting von Mises equivalent stress distribution for both calculated steps is shown in Fig. 6. Note that the stresses in the pin are very high and are close to the material's yield strength even in the make-up stage. When the axial load is applied, the highest stress concentration appears at the root of the last engaged thread of the pin (indicated by the arrow in Fig. 6b). This corresponds to the results obtained by Griffin et al [1] for the analysis of well casing connections and by Dvorkin and Toscano [9] for other API connections.

In Fig. 7 and Fig. 8 the different stress components are shown for the make-up stage and with an additional external load of 150 MPa. From these figures it can be seen that for both load steps, the high von Mises stresses are mainly the result of hoop stresses and axial stresses. Radial and shear stresses are low apart from some local effects around the last engaged thread of the pin.

The acting hoop stresses in the pin have a negative sign, which indicates compressive stresses while the hoop stresses in the box are positive, being tensile stresses. In the thread runout region of the pin, the axial stresses are compressive at the inside wall of the pin and axial tensile stresses appear at the outside. This is the consequence of bending of the pin due to make-up deformation. A similar situation appears at the box recess region. This is the unthreaded extension at the left side of the box. Due to make-up this recess tends to bend causing the axial stress gradient at that location.

The maximum acting von Mises equivalent stress is 425 MPa, which corresponds to a stress concentration factor of 2.83 relative to the applied axial tensile stress of 150 MPa. The stress concentration is mainly composed of axial and hoop tensile stresses. The compressive hoop stress in the pin is reduced due to the axial tensile stress.

The stress concentration at the last engaged thread of the pin is caused by the non uniform load distribution of the axial load over the different threads. This distribution, as a percentage of the total load, is shown in Fig. 9, thread number 1 corresponds to the LET of the pin, as shown in Fig. 3.

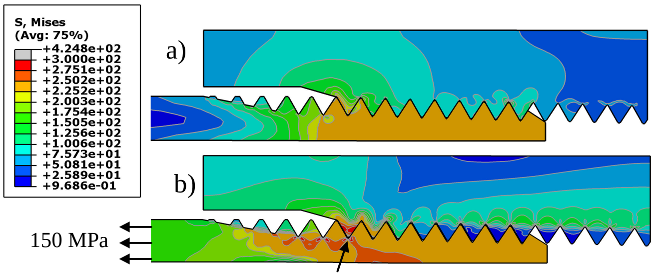

Fig. 6: von Mises stress distribution a) at make-up; b) with an external axial tensile stress of 150 MPa

At an external tensile stress of 100 MPa, the LET carries 47% of the total load. The thread after the LET, which is not fully engaged (thread 0 in Fig. 9), carries a negative load which is a compression caused by the bending of the pin during make-up. When the external stress is increased to 150 MPa, the LET starts to bend, transmitting part of its load to the other threads. Additionally, with this load, the threads will start to slide over each other, creating an opening between the threads and eliminating the compressive load on thread 0. This way the load carried by the LET is reduced to 36% of the total load.

The opening between the threads, however, is highly undesirable since the fluid inside the pipe can find its way out through the created helical path.

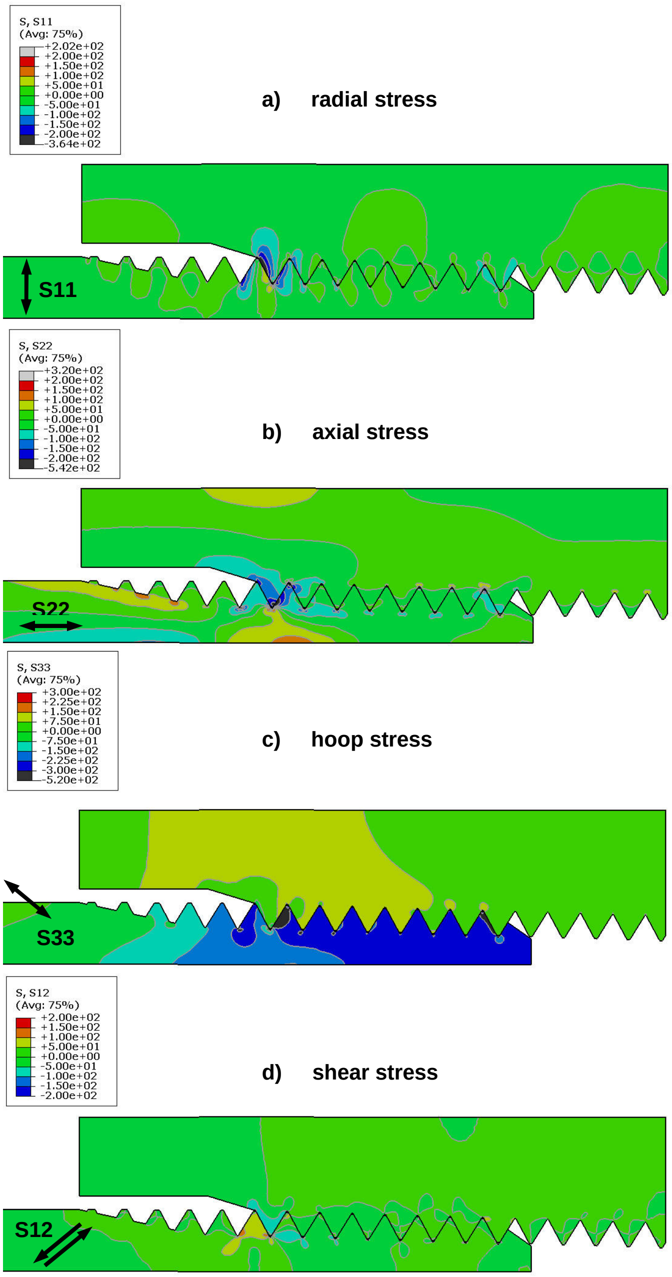

Fig. 7: Stress components at make-up

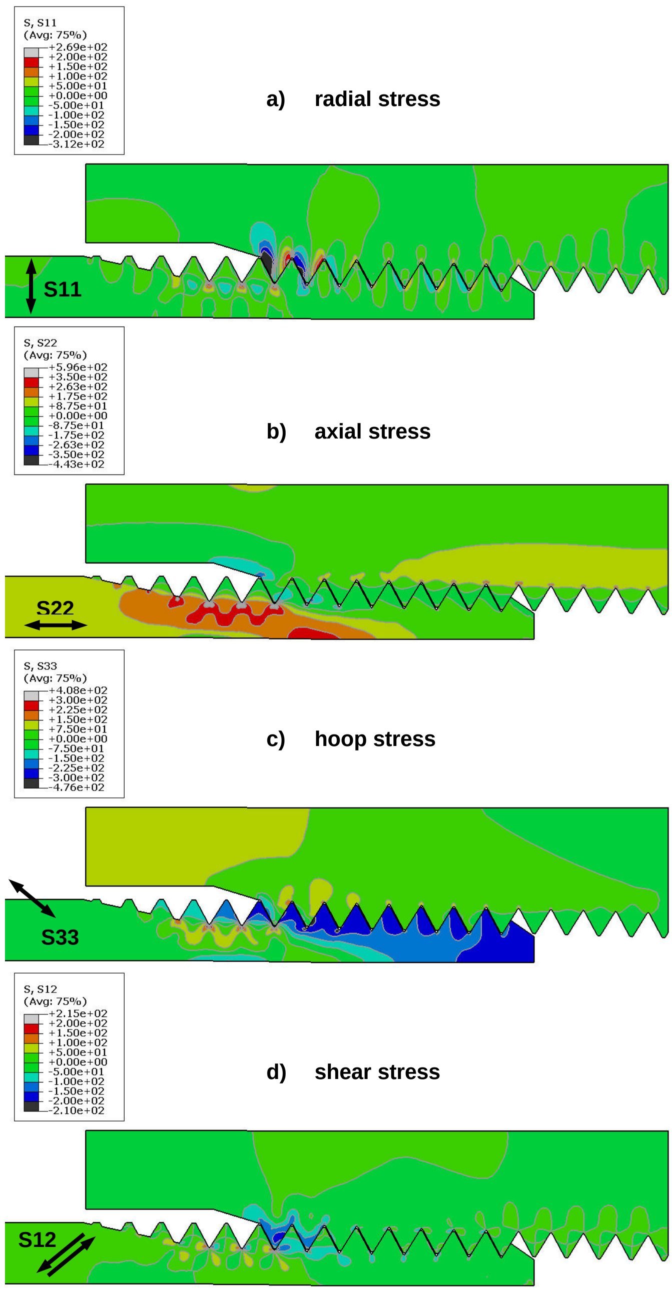

Fig. 8: Stress components with an additional axial load of 150 MPa

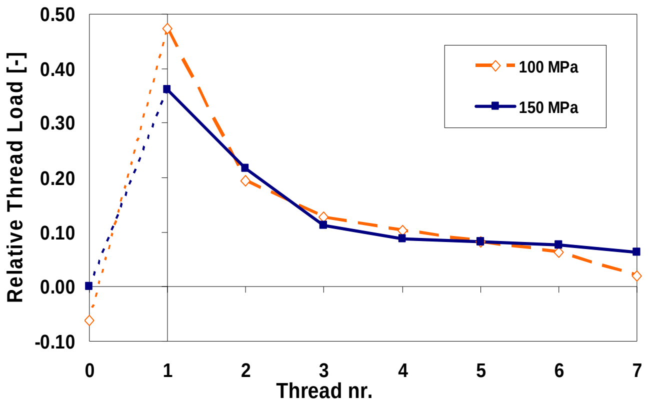

Fig. 9: Thread load distribution (numbering as in Fig. 3)

IV. Parametric Study

A. Performance Parameter

It is known from Newport [3] that changing the hoop stiffness of pin or box affects the load distribution over the threads. The exact correlations and the effect on the overall behavior of the connection, however, remain unknown.

To study the influence of changes in COF, pin and box wall thickness and box recess length, a parametric study was carried out. To evaluate the results of this study, a performance parameter P is introduced, see Eq. 1:

$$P = \frac{1}{O_n} \cdot \frac{1}{TL_n} \tag{1}$$

The performance parameter combines the inverse of the normalized thread opening O_n, as an indication for the sealability of the connection, with the inverse of the normalized thread load at the last engaged thread of the pin TL_n, as a measure for the static and fatigue strength of the connection. This way the parameter encloses the two basic requirements that should be met during its service life.

The normalized opening O_n is defined as the ratio between the value of the opening of the connection with a certain modification and the value of the opening of the standard API Line Pipe connection (0.093 mm) at an external axial stress of 150 MPa and with a COF of μ = 0.12, see Eq. (2):

$$O_n = \frac{O_{modified\_connection}}{0.093mm} \tag{2}$$

As will be shown in Fig. 10, the thread opening starts to increase after a certain external axial stress is exceeded. For this reason the normalized opening O_n is defined at the highest applied external stress of 150 MPa. At this stress, however, the LET is bent and the load distribution over the threads has changed. This is why the normalized thread load TL_n is defined at a lower external axial stress of 100 MPa where the load distribution is more representative for the overall behavior of the connection.

The normalized thread load TL_n is the ratio between the thread load at the LET of the pin of the connection with a certain modification and the standard API connection at an external axial stress of 100 MPa (47%), see Eq. (3).

$$TL_n = \frac{TL_{modified\_connection}}{47\%} \tag{3}$$

The parameters O_n and TL_n are defined relative to the values of the standard API Line Pipe connection, therefore the value of the performance parameter P equals 1 for the standard connection. A value of $P > 1$ implies an improved performance relative to the standard connection, while $P < 1$ implies a performance decrease. In the following paragraphs the performance parameter will be used to quantify the effects of parametrical changes on the behavior of the connection.

B. Results

1) Influence of the coefficient of friction

By changing the coefficient of friction as the only parameter during a series of simulations it was found that the thread opening due to external loading is highly dependent on the value of the COF. Even when the COF is kept between the values mentioned previously, the behavior of the connection changes significantly, as can be seen from Fig. 10.

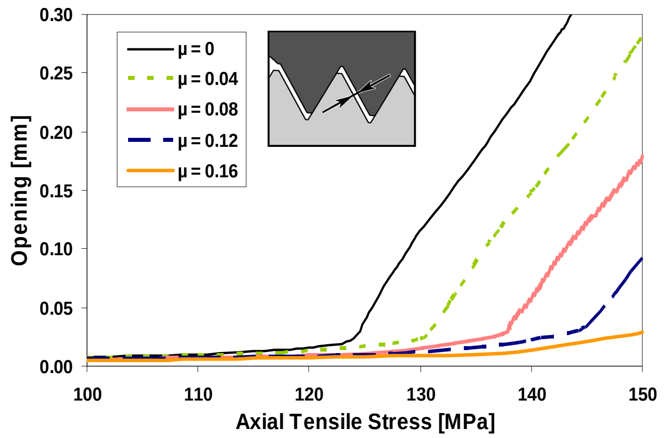

Fig. 10: Influence of the coefficient of friction on the thread opening

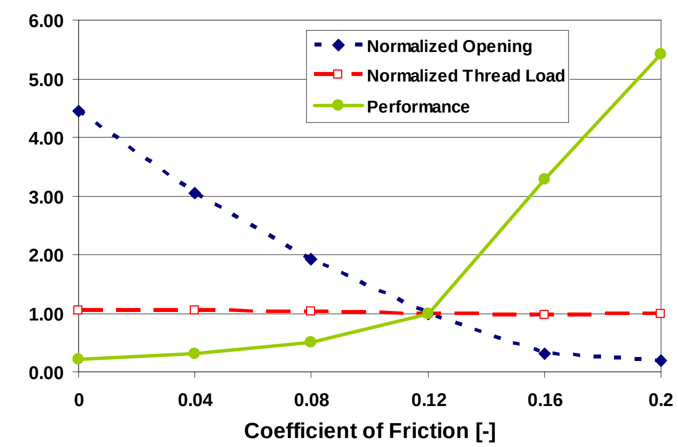

Fig. 11: Influence of the coefficient of friction on the connection parameters

The opening is defined as the perpendicular distance between the thread flanks, and varies for an external tensile stress of 150 MPa between 0.03 mm when μ = 0.16 and 0.41 mm for the frictionless situation. At low values of the COF, the thread opening starts to increase at a lower external load than when a higher COF is used.

From this it can be seen that accurate knowledge of the COF is necessary to model threaded connections. In a previous study by the authors [22], the COF was determined experimentally for an API Line Pipe connection. A value of μ = 0.12 was obtained, which is used as the standard value for the COF throughout the subsequent simulations.

Despite the important influence on the thread opening of the connection, a change in COF has no important effect on the stress and load distribution of the connection, as shown in Fig. 11. For this reason the performance parameter *P* increases with increasing coefficient of friction. As a remark it is noted that increasing the friction between pin and box can be undesirable in some cases because of the higher torque necessary for make-up. Additionally, thread grease or thread compounds are necessary to avoid galling damage that can occur during make-up due to high friction. The performance parameter, however, does not take this phenomenon into account.

2) Box wall thickness variation

When the wall thickness of the box is increased, this component becomes more rigid. This means that the deformation of the box during make-up will be smaller and lower tensile hoop stresses will appear as is illustrated in Fig. 12 (the hoop stresses are taken at outside wall of the box). This means that, for the same number of make-up turns, the pin will have to deform more. For the standard connection, the engaged threads of the pin are yielding at make-up. A further increase in plastic deformation, for the connection with the increased box wall, does not change the pin hoop stress significantly. But in the runout region of the pin, which does not yield at make-up, the magnitude of the compressive hoop stresses are increased due to the higher box stiffness. When the box wall is decreased, box stresses increase and pin stresses decrease in the runout region. The influence of the box wall thickness on the axial stress and von Mises equivalent stress is completely similar.

When the connection is loaded, it can be seen from Fig. 13 that a thinner box results in a lower thread load on the LET of the pin. However, due to the higher deformation of the pin, the opening will be larger. The increased opening dominates the thread load reduction giving a decreased overall performance *P*. Increasing the box wall thickness gives an increased performance (up to 6% for a 6mm wall thickness increase). This is only due to the reduced thread opening, the thread load does not increase with a wall thickness larger than the standard value, although the hoop stress in the runout region of the pin is higher.

Only a box wall thickness increase results in a connection with a better performance. However, this would mean a heavier, more expensive coupling and hence is not desirable.

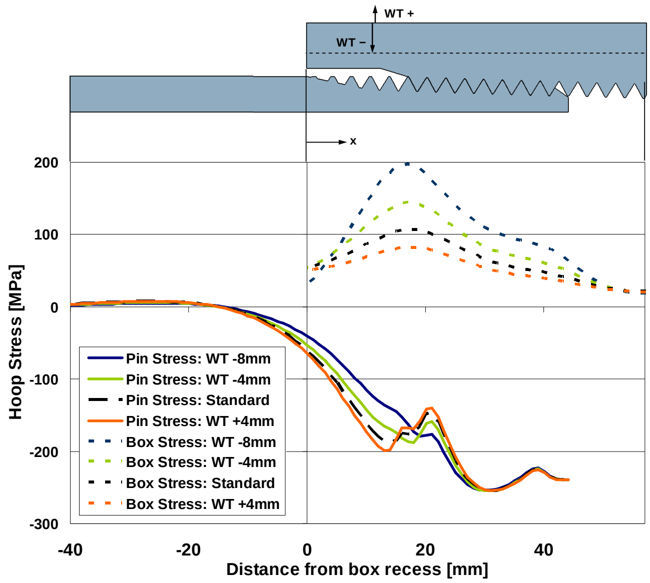

Fig. 12: Hoop stress at the inside wall of the pin and the outside wall of the box at make-up for different values for the box wall thickness.

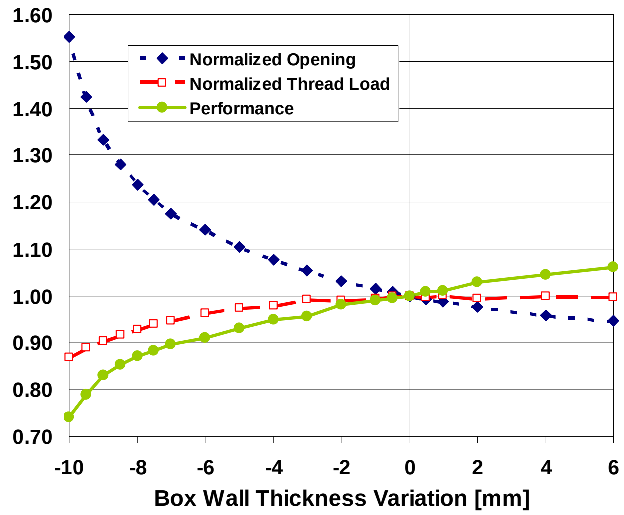

Fig. 13: Influence of the box wall thickness on the connection parameters

3) Pin wall thickness variation

Instead of decreasing the box wall, the pin wall could be increased to get the same change in stiffness between pin and box. As can be seen from Fig. 14, a pin wall thickness increase results in an increased box hoop stress and a decreased compressive hoop stress in the runout region of the pin at make-up.

When an external axial load is applied, it can be seen that both the thread load at the LET of the pin and the thread opening decrease with increasing pin wall thickness. Hence the performance parameter increases with increasing pin wall thickness. Note that an increased wall thickness of the pin, means that the total force on the connection is increased since the external applied axial stresses are kept constant at 100 MPa and 150 MPa.

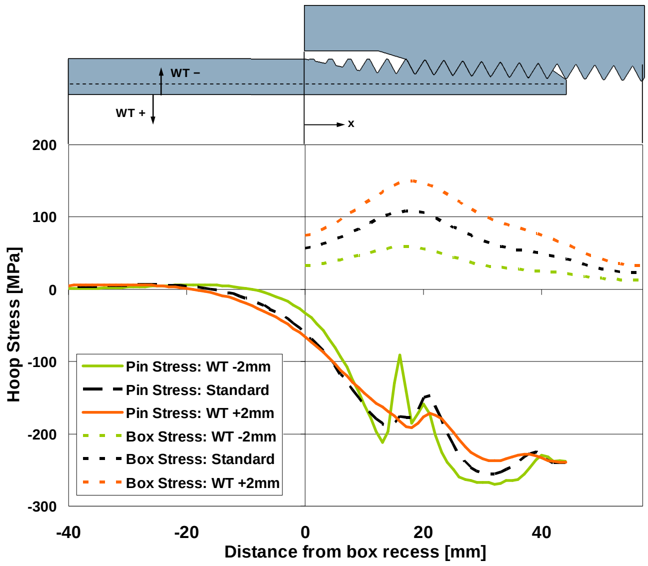

Fig. 14: Hoop stress at the inside wall of the pin and the outside wall of the box at make-up for different values for the pin wall thickness.

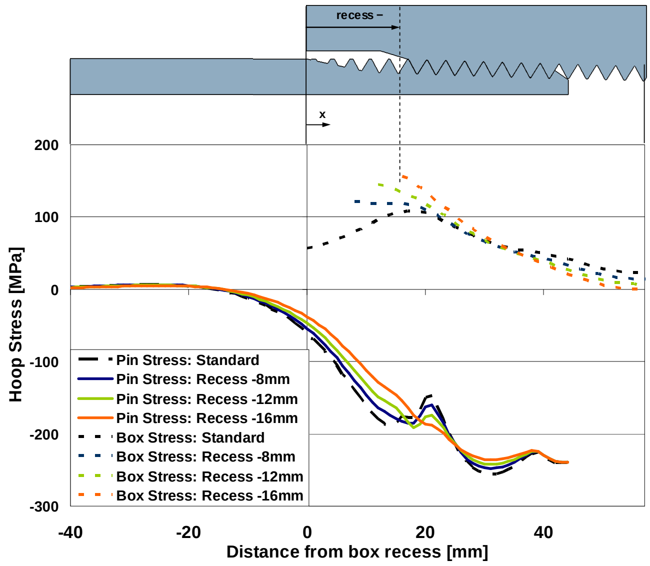

Fig. 16: Hoop stress at the inside wall of the pin and the outside wall of the box at make-up for different values of the box recess length.

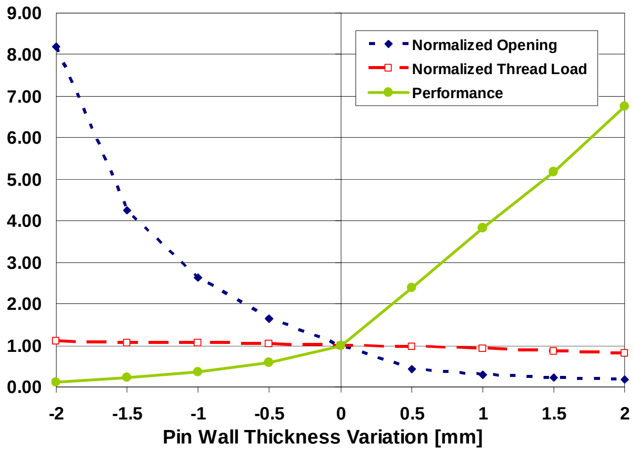

Fig. 15: Influence of the pin wall thickness on the connection parameters

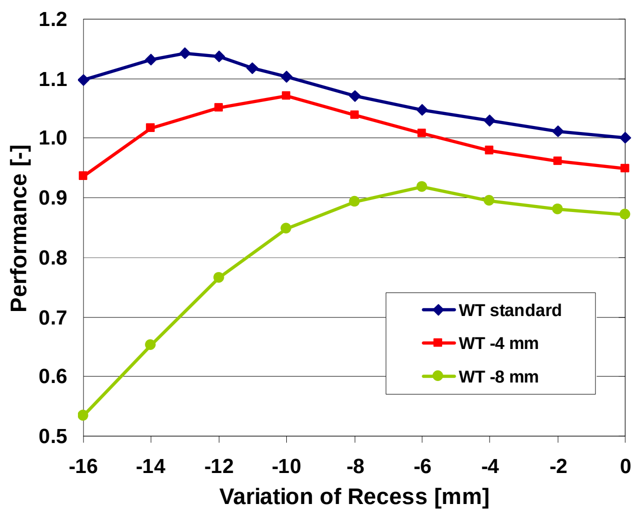

Fig. 17: Influence of the box recess length and wall thickness on the connection performance

4) Recess length variation

As is discussed in paragraph III.B, bending of the recess causes a raised axial stress in the box. For this reason, the recess length was changed during the parametric study. From Fig. 16 it can be seen that reducing the recess length results in a higher maximum value of the hoop stress in the box and reduces the hoop stress in the runout region of the pin. As can be expected from the previous results, this creates a reduction in thread load on the LET of the pin together with an increased opening. In Fig. 17 results for connections with a reduced recess length, together with varying wall thickness are shown. It can be seen that decreasing the recess length increases the connection's performance until a maximum is reached after which the performance will decrease. With a reduced box wall thickness, this maximum appears for a smaller value of the recess reduction.

For any recess reduction, an additional decrease of the box wall thickness does not result in an increased connection performance, which corresponds to the results obtained for the connections where the box wall thickness was altered. The best result is obtained for a box with standard wall thickness and a recess reduction of 13 mm.

V. CONCLUSIONS

A new modeling method, entitled *ThreadGen*©, to perform parametric studies on threaded connections has been presented. The modeling strategy and the used non-linear 2D axisymmetric finite element model were discussed. The modeling method was illustrated by a parametric study of an API Line Pipe threaded connection.

A performance parameter was defined, combining strength and sealability parameters to quantify the influence of the coefficient of friction, pin and box wall thickness and recess length. It was shown that accurate knowledge about the coefficient of friction between the threads is necessary to obtain reliable results.

Improved performance was obtained for a pin with increased wall thickness, reducing the box wall thickness is not desirable. Additionally an optimal value for the box recess length was found, different from the standard size.

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REFERENCES

[1] Griffin, R.C., Kamruzzaman, S., Strickler, R.D., *Casing Drilling Drill Collars Eliminate Downhole Failures*, Offshore Technology Conference, Houston, Texas, USA, May 2004.

[2] Glinka, G., Dover, W.D., Topp, D.A., *Fatigue assessment of tethers*, Institution of Mechanical Engineers Conference, Proceedings nr. C135/86, pp. 187-198, 1986.

[3] Newport, A., *Stress and fatigue analysis of threaded tether connections*, PhD Thesis, University College London, 1989.

[4] Assanelli A.P., Dvorkin, E.N., *Finite-Element Models of OCTG Threaded Connections*, Computers & Structures, 47(4), pp. 725-734, 1993.

[5] Dvorkin, E. N., Assanelli, A. P. and Toscano, R. G., *Performance of the QMITC Element in Two-Dimensional Elasto-Plastic Analysis*, Computers & Structures, 58(6), pp. 1099-1129, 1996.

[6] Assanelli, A.P., Xu, Q., Benedetto, F., Johnson, D.H., Dvorkin, E.N., *Numerical/experimental analysis of an API 8-round connection*, Journal of Energy Resources Technology-Transactions of the ASME. 119(2), pp. 81–88, 1997.

[7] Curley, J.A., Prinja, N.K., *Failure Investigation of an offshore drilling* component, Abaqus Users' Conference, pp. 1-14, Chester, UK, May 1999.

[8] MacDonald, K.A., Deans, W.F., *Stress analysis of drillstring threaded connections using the finite element method*, Engineering Failure Analysis, 2(1), pp. 1-30, 1995.

[9] Dvorkin, E.N. and Toscano, R.G., *Finite element models in the steel industry, Part II: Analyses of tubular products performance*, Computers and Structures, 81(8-11), pp. 575–594, 2003.

[10] Kristensen, A.S., Toor, K., Solem, S.I., *Finite Element analysis of jar connections: modeling considerations*, Journal of Structural Mechanics: Special issue for the 18th Nordic Seminar on Computational Mechanics, pp. 1-4, Helsinki, Finland, 2005.

[11] Guangjie, Y., Zhenqiang, Y., Qinghua, W., Zhentong, T., *Numerical and experimental distribution of temperature and stress fields in API round threaded connection*, Engineering Failure Analysis, 13(8), pp. 1275–1284, 2006.

[12] Bertini, L., Beghini, M., Santus, C. and Baryshnikov A., *Fatigue on drill string conical threaded connections, test results and simulations*, 9th Int. Fatigue Congress, Atlanta, USA, 2006.

[13] Santus, C., Bertini, L., Beghini, M., Merlo, A., Baryshnikov, A., *Torsional strength comparison between two assembling techniques for aluminium drill pipe to steel tool joint connection*, International Journal of Pressure Vessels and Piping, 86(2-3), pp. 177-186, 2009.

[14] Ribeiro Plácido, J.C., Valadão de Miranda, P.E., Netto, T.A., Pasqualino, I.P., Miscow, G.F., de Carvalho Pinheiro, B., *Fatigue analysis of aluminum drill pipes*, Materials Research, 8(4), pp. 409-415, 2005.

[15] Chen, J.J., Shih, Y.S., *A study of the helical effect on the thread connection by three dimensional finite element analysis*," Nuclear Engineering and Design, 191(2), pp. 109–116, 1999.

[16] Zhong, A., *Thread Connection Response to Critical Pressures*, Abaqus Users' Conference, pp. 690–706, Paris, France, May 2007.

[17] ISO 10407-1, *Petroleum and natural gas industries – Drilling and production equipment – Part 1: Drill stem design and operating limits*, 2004.

[18] Ertas, A., Cuvalci, O., Carper, H.J., *Determination of Friction Characteristics of J-55 OCTG Connections Lubricated with Environmental Safe Thread Compound*, Tribology Transactions, 42(4), pp. 881-887, 1999.

[19] API Specification 5B, *Specification for Threading, Gauging and Thread Inspection of Casing, Tubing and Line Pipe Threads (U.S. Customary Units)*, American Petroleum Institute, fourteenth edition, 1996.

[20] API Specification 5L, *Specification for Line Pipe*, American Petroleum Institute, forty-second edition, 2000.

[21] Clinedinst, W.O., *Joint Strength Formulas for API Threaded Line Pipe*, API Circular PS-1533, 1976.

[22] Van Wittenberghe, J., De Baets, P., De Waele, W., Van Autrève, S., *Numerical and Experimental Study of the Fatigue of Threaded Pipe Couplings*, accepted at the Contact and Surface Conference, Algarve, Portugal, to be published in June 2009,.