A COMPUTATIONAL FATIGUE APPROACH TO THREADED CONNECTIONS

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Abstract The fatigue design of threaded connections is a complicated task due to the high stress concentration factors, the multi-axial stress state at the thread root and the fact that the tightening torque plays a very important role in the fatigue performance of the connection. In this work, a local fatigue approach, based on the concept of elastic shakedown at the thread or notch root is proposed, and shown to be well suited to the relatively short fatigue lives targeted (i.e. less than 300000 cycles). Two-dimensional, axisymmetric, elasto-plastic finite element models are developed to determine the local cyclic stress state. Contact elements and a piston / rod geometry with an initial inter-penetration are used to model the tightening torque. A non-linear combined kinematic and isotropic hardening law (Lemaitre and Chaboche) is used to describe the cyclic plastic behavior of the material. Constant strain cyclic tensile tests have been done to identify the hardening parameters of the steel investigated.
It is shown that the local cyclic stress, at the most severely loaded thread notch root, can be classed as either: elastic shakedown, plastic shakedown or cyclic racketing, and that this behavior is dependent on both the applied hydraulic pressure and the tightening torque. Full scale experimental tests conducted by SERTA provide good correlation with the results of the proposed methodology, for the targeted number of cycles.
This work was conducted within the framework of commercial collaboration between SERTA and Arts et Métiers ParisTech Angers, hence due to reasons of confidentiality, this article focuses on the proposed methodology and contains only limited results.

1 INTRODUCTION

For over 30 years the French company SERTA, located at La-Roche-Sur-Yon in Vendée (85), have specialized in the manufacture of hydraulic actuators, typically used on earthmoving and construction equipment (see Figure 1(a)). The principal function of a hydraulic cylinder is to transmit axial loads, which are variable or cyclic in nature. Hence, such components are exposed to the phenomenon of fatigue failure. For the fatigue sizing of their products, SERTA have developed various in-house tools which are largely based on empirical relations, handbook recommendations and their extensive experience in the domain.

This work focuses on one of the fatigue critical elements in a hydraulic actuator: the threaded connection between the piston and the piston rod (see Figure 1(b)). As with all threaded connections, the fatigue behavior dependants strongly on the tightening torque applied during assembly. This is a critical notion for SERTA as they have clients who wish to decrease the tightening torque, compared to the standard SERTA practice, so the connection can be easily undone for maintenance. Figure 2(a) shows an axisymmetrical cross-section of a typical piston/rod connection studied here. Fatigue crack initiation is generally localized at the root of the first load-bearing thread of the piston rod. Figure 2(b) shows a typical fatigue failure surface of a piston rod. The number of cycles corresponding to the design life of a typical actuator is generally of the order of 300 000 cycles. The overall objective of this work is to develop a fatigue design strategy that gives SERTA greater confidence and flexibility in their designs.

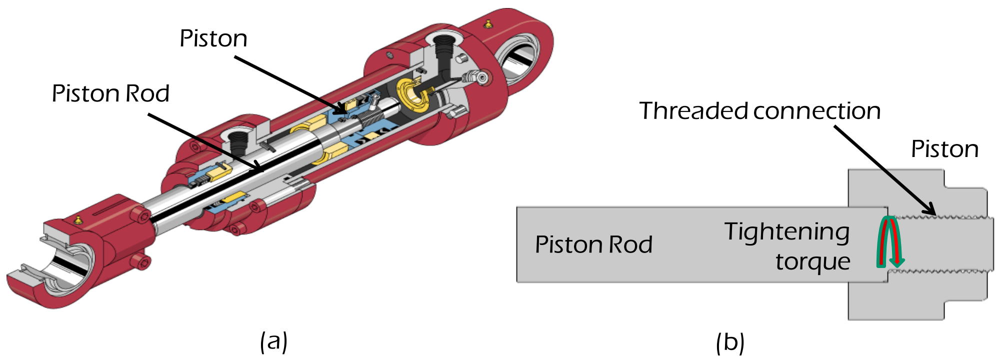

Figure 1. (a) Typical hydraulic actuator (b) Simplified 2D representation of the threaded connection between the Piston and the Piston Rod

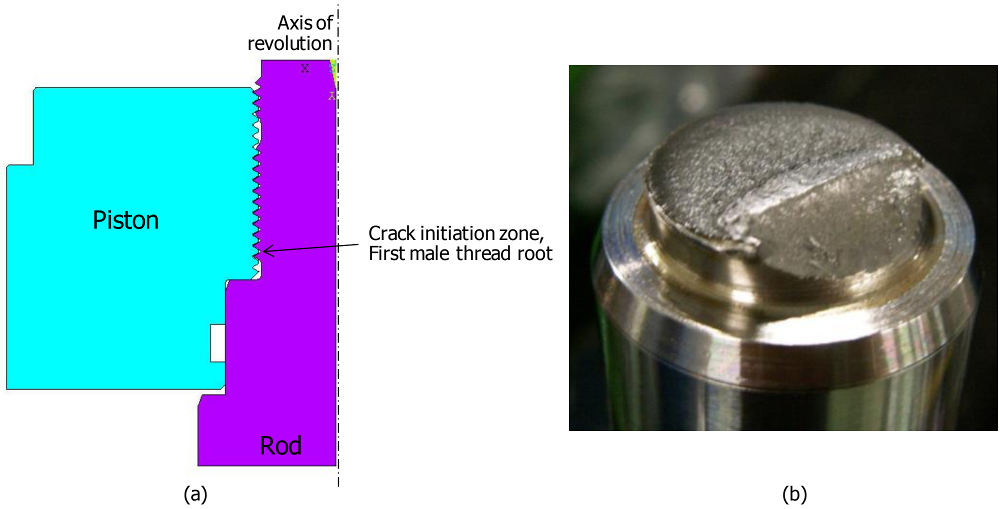

Figure 2. (a) The 2D axisymmetric geometry studied (b) A typical fatigue failure surface, after crack initiation at the root of the first load-bearing thread of the piston rod.

2 THE FATIGUE ASSESSEMENT OF THREADED CONNECTIONS

Bolted joint and threaded connections are one of the most commonly used connection types in industry. As such, their resistance in terms of fatigue is of great importance. There exist a large number of fatigue assessment methods that can and have been proposed for these types of connections. These methods can be classified into the following categories:

- Nominal approaches, based on analytical expressions. These methods are often used in industry and in national or international standards [1][2][3].
- Fracture mechanics approaches, often based on the elastic stress field, or analytical expressions for the stress intensity factor [4] [5].
- Variations of the classical strain-life approach [6].
- Extensions of multi-axial fatigue assessment approaches like the Dang Van criterion [7].

One of the major difficulties involved in the fatigue design of threaded connections is the accurate estimation of the cyclic stress-strain state in the fatigue process zone. This is complicated by the fact that a connection is composed of at least two parts in contact (i.e. nut and bolt), the helical geometry of the threads is three-dimensional by nature and the effect of the tightening torque or the pre-load. It should also be noted that the load is not uniformly distributed in the different threads of the connection. It is well known that the first male thread is the most loaded and sees between 30 and 35% of the total load carried by the connection.

In this work, a simple fatigue assessment method, based on the concept of elastic shakedown at the macroscopic scale, is used in conduction with elasto-plastic finite element calculations to determine the cyclic stress-strain state in the fatigue process zone. This methodology is discussed below.

2.1 Finite element calculations

The cyclic stress state of the threaded connection has been calculated using the commercial finite element software ANSYS.

The geometry: The example presented here is an M30x2 threaded connection between the piston and rod (see Figure 1(a)). The nominal outside diameter of the piston is 120mm and the OD of the rod is 40mm. Two-dimensional axisymmetrical models are used. Note that with this 2D simplification the helical form of the threads is neglected. The piston and rod are modeled as being independent areas and surface-to-surface contact, including friction, is used to model the interface between the piston and the rod. A nominal value of μ=0.13 is used for the coefficient of friction. In order to be conservative, the minimum diameter of the rod is used in conjunction with the maximum diameter of the piston, corresponding to the fabrication tolerance 6H/6g.

The FE mesh: The finite element mesh is automatically generated, using 8-noded QUAD elements (PLANE183). At the root of the first load-bearing thread of the rod, the mesh is refined so that there are approximately 10 elements in the length of the root radius. The average element size in this zone is 0.058mm. The root radius is 0.25mm.

The materials: The piston and the rod are made from two different steels. Their exact designations are confidential.

The material model: An elasto-plastic material model is used. The elasticity is assumed to be linear, isotropic. The plasticity is described by the von Mises yield surface, an associative flow rule and combined non-linear isotropic and kinematic hardening laws based on the theory of Lemaitre and Chaboche [8]. This combined hardening model is implemented in ANSYS and is capable of modeling complicated cyclic plastic behavior of materials, such as cyclic hardening or softening and ratcheting or shakedown. The isotropic hardening can be described by the equation:

$$\sigma_Y = k + Q[1 - exp(-bp)]$$

Where σ_Y is the current yield strength, k is the initial yield strength, Q is the asymptotic value corresponding to the cyclic stabilized regime, b controls the rate of stabilization and p is the accumulated plastic strain.
The kinematic hardening is described by the variation in the backstress X_{ij} and takes the form:

$$\dot{X}_{ij} = \left(C\frac{\sigma_{ij} - X_{ij}}{k} - \gamma X_{ij}\right)\dot{p}$$

Where C and γ are material parameters

Identification of the material model parameters:
For each material there are 4 hardening material parameters (Q, b, C and γ) that must be determined by regression analysis of cyclic experimental test data. In addition, there are two elastic material parameters (Young's modulus, E and Poisson's ratio, ν) and the initial yield strength, k, which can be determined via tensile tests. Constant strain cyclic tensile tests have been done in order to identify the hardening parameters of the steels investigated. Figure 3 shows the comparison between the experimental and numerical results, for the stabilized stress-strain curves, obtained for one of the 2 steels used in this work. Three different values of imposed strain are shown.

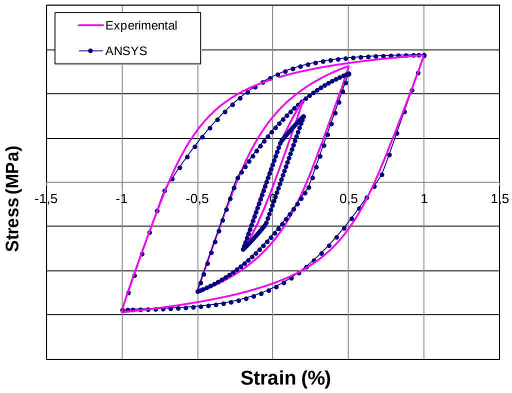

Figure 3. Comparison between experimental and numerical results for the "stabilized" cyclic stress-strain curves

Modeling of the tightening torque:
There exists different ways of modeling the tightening torque (or the preload) in a 2D axisymmetric finite element analysis of a threaded connection. The approach adopted by Tafreshi and Dover [9] involves using "gap elements" to model point-to-point contact at the threads and at the shoulder. These elements have a large stiffness when active (in compression) and a small, but non-zero stiffness when in an inactive state (i.e. tension). An initial opening distance can be specified in order to generate a preload. An alternative method is to include a layer of thermal elements in the female piece of the connector. By introducing an initial temperature which is superior to that of the surrounding material, the thermal expansion of these elements has the effect of creating a preload in the connection and hence enabling the simulation of a tightening torque [10].

In this work the tightening torque is simulated by including an initial inter-penetration at the shoulder between the piston and the rod (see Figure 4). The contact algorithm ensures that this initial overlap is reduced to zero in the first calculation step, hence creating contact pressures at the shoulder and the load carrying threads (see Figure 5). Numerical integration of the contact pressure at the shoulder is then done to determine the value of axial preload, Q. The tightening torque, resulting from a given inter-penetration, is subsequently calculated via a closed form analytical expression proposed by Guillot [6] which is valid for standard metric threads.

$$C = Q\left(\frac{P}{2\pi} + 0.583\ d_2 f + \rho_m f_2\right)$$

Here, C is the tightening torque, Q is the axial preload, P is the thread pitch, d_2 is the nominal thread diameter, f is the coefficient of friction at the threads, f_2 is the coefficient of friction at the shoulder, and ρ_m is the average radius of the contact zone at the shoulder. As discussed by Guillot, this expression has three terms, representing three different components of the tightening torque. The first is the torque resulting in the preload Q which ensures the connection between the assembled parts. The second term is needed to overcome the friction between the threads of the male and female parts. The final term represents the torque necessary to overcome the friction at the shoulder between the male and female parts.

In this work the coefficient of friction between the rod and the piston was assumed to be the same at the threads and at the shoulder (i.e. f is assumed to be equal to f_2). The same numerical value of the coefficient of friction was also used in the contact algorithm. The sensibility of the results on this parameter was investigated. However, the nominal value of $f = 0.13$ is used.

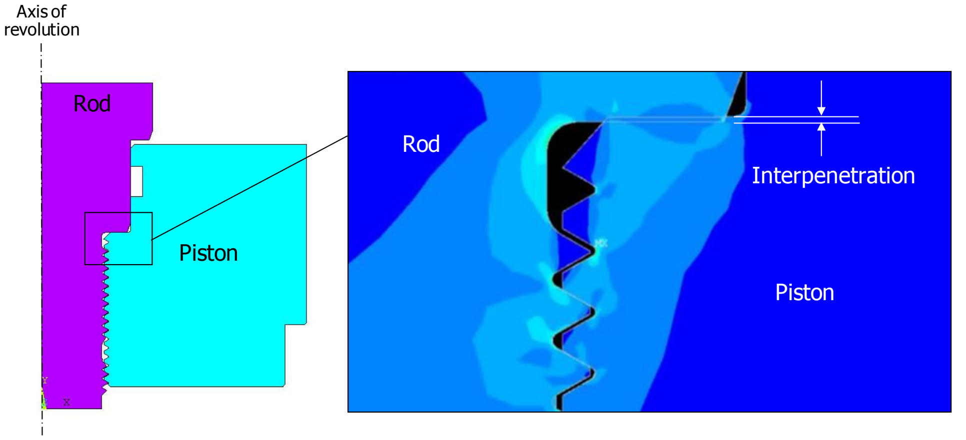

Figure 4. An initial interpenetration between the Piston and the Rod

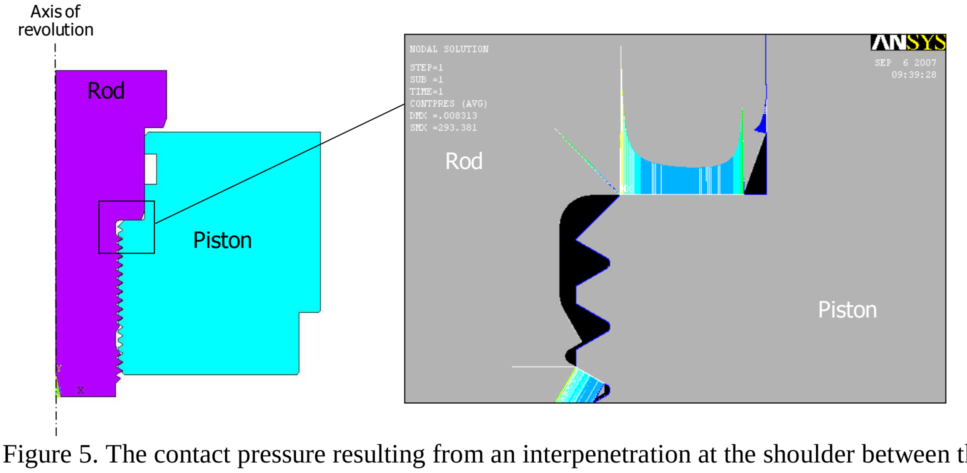

Figure 5. The contact pressure resulting from an interpenetration at the shoulder between the piston and the rod

The boundary conditions and cyclic loads: Figure 6 shows the boundary conditions used in the finite element models. The horizontal line at the top of the rod was completely fixed in space. The length of the rod modeled was found to have little effect on the local stress and strain states at the thread root in the fatigue process zone. As the cylinder bore was not explicitly modeled the vertical line on the exterior of the piston was blocked in the horizontal direction (see Figure 6).

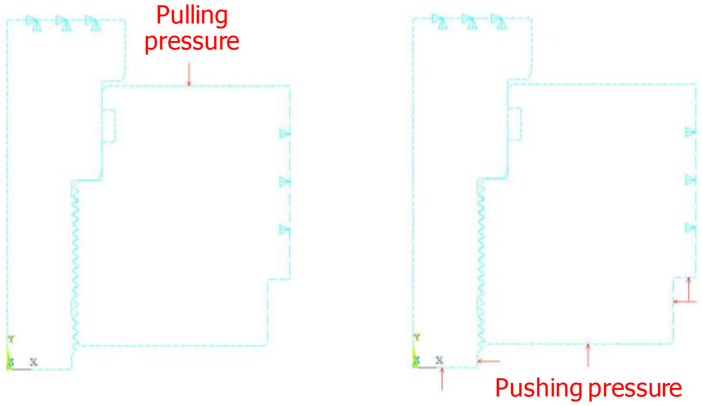

Figure 6. Boundary conditions and loads

A cycle is defined as the application of the maximum pressure in the small chamber (pulling pressure) followed by application of the maximum pressure in the large chamber (pulling pressure). After an initial calculation step used to determine the tightening torque as a function of the interpenetration, the stress and strain are calculated for a total of 10 cycles, using a triangular waveform.

<u>Typical results: </u>Figure 7(b) shows and example of the calculated stress-strain response in the fatigue process zone, for a total of 10 cycles. In this example it can be seen that the material has “almost” achieved a plastic shakedown state. The phenomenon of *mean stress relaxation* can also be observed. That is, the successive stress-strain hysteresis loops have lower mean stress.

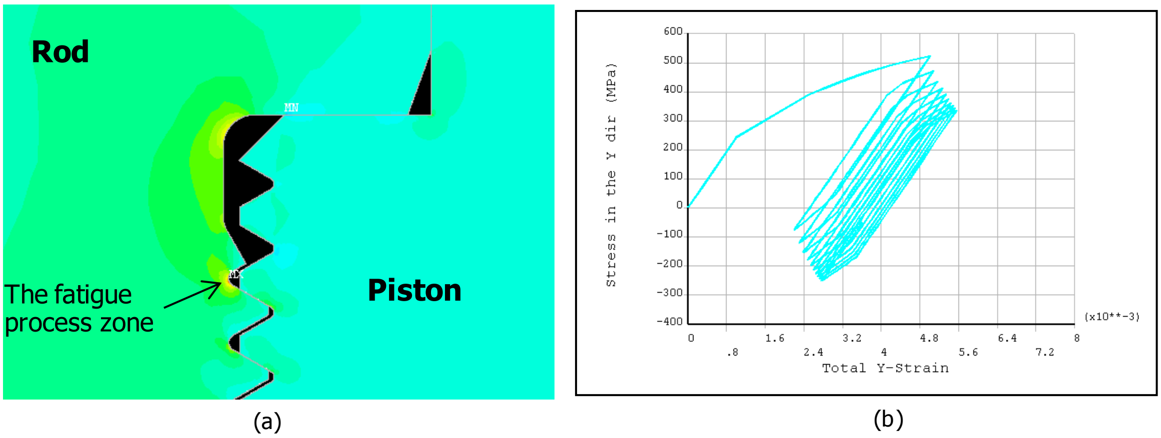

Figure 7. An example of a typical result showing (a) the fatigue process zone (b) the cyclic stress-strain relationship at the thread root

<u>Automation of the analysis:</u> The complete analysis, including the fatigue assessment discussed in the following section, has been automated using scripts developed in APDL (Ansys Parametric Design Language). This automation is particularly important in determining the appropriate value of interpenetration, d, at the rod/piston shoulder, which results in a specific tightening torque. As shown in Figure 8, this is based on an iterative DO-WHILE loop which uses a Newton-Raphson convergence criterion.

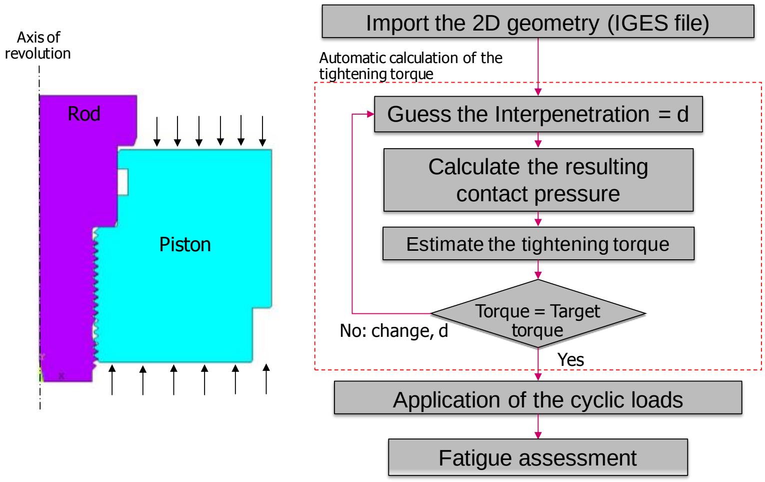

Figure 8. The general algorithm resulting in the fatigue assessment

2.2 Cartographies of the local stress-strain state

Figure 9 shows a schematic example of how the results are presented in this work. That is, cartographies of Tightening Torque versus Cylinder Pressure are established using the finite element modeling techniques described above. Specifically, these cartographies shows the cyclic stress-stain state calculated at the root of the most severely loaded thread of the rod. The different zones of the diagram represent the different possible asymptotic cyclic states:

a) *Elasticity* – The material is never permanently deformed. This is the case for low tightening torques and low cylinder pressure.
b) *Elastic shakedown* – The material initially undergoes plastic deformation, and then re-establishes its elastic behavior.

c) *Plastic shakedown* – A stabilized, stress-strain hysteresis loop is established.
d) *Ratcheting* – The material never achieves a stabilized stress-strain loop.

The Ratcheting and plastic shakedown states are obtained when the cylinder pressure is high and the tightening torque is low and are considered to be extremely undesirable in terms of fatigue.

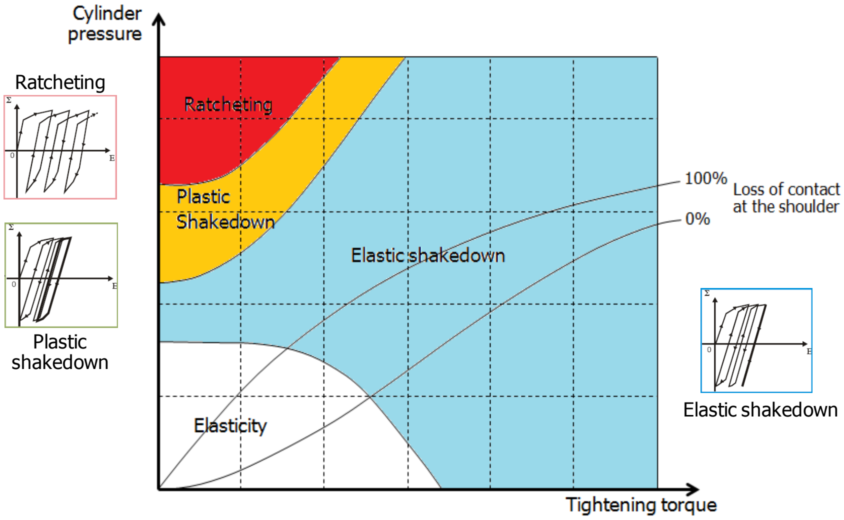

Figure 9. A schematic example of the Tightening Torque verses Cylinder Pressure diagrams used to present the FE results

The other phenomenon that can be described by these cartographies is the "loss of contact" at the piston/rod shoulder. This phenomenon can have serious consequences on the mechanical behavior of a threaded connection, especially if there is 100% loss of contact.

It should be noted that the generation of these diagrams is very computationally intensive. Each diagram is created for a specific geometry (and materials) via finite element calculations for different combinations of pressure and torque. Typically a diagram consists of a minimum of 36 elasto-plastic FE models (i.e. 6 pressures and 6 tightening torques). This is possible due to the fact the calculation process has been automated using the APDL programming language.

2.3 A practical fatigue design approach dedicated to threaded connections

In this work a certain number of fatigue criteria were investigated for the fatigue assessment of this type of connection. In particular, the critical distance methods advocated by Taylor and co-workers [11] and the classical multi-axial fatigue approaches, like the Crossland [13] and Dang Van [12] criteria were tested. However, the performance of these criteria is not discussed here, instead an alternative approach, simply based on the concept of elastic shakedown (at the macroscopic scale) is adopted as it was found to give good results with respect to the experimental data (see section 3 below). As the principal preoccupation of the SERTA Group is to develop a robust design tool, a practical approach has been proposed by Arts et Métiers ParisTech, which is targeted to the fatigue design life of the threaded connections in question (i.e. 300 000 cycles). This criterion can be summarized by the following statement

> *Fatigue failure is likely to occur if the material in the fatigue process zone at the root of the most severely loaded male thread does not achieve an elastic shakedown state.*

In other words, the threshold between the elastic and the plastic shakedown states is determined in the most critical zone. This threshold is dependent on the applied tightening torque and the cylinder pressure.

3 FULL SCALE TESTS

A series of full scale experimental tests have been carried out by SERTA, with the aim of determining the effect of the tightening torque on the fatigue life and to validate (or not) the numerical fatigue assessment procedure discussed above. The tests were done using a QUIRI-HYDROSYSTEMS electro-hydraulic test bench powered by a 270/110-1250 hydraulic power unit (2x55kW, 2x90 L/min at 350 bars max). The test bench and the geometry of the tested actuator are shown in Figure 10. This actuator was chosen to be representative of the SERTA product range and their standard fabrication methods. The geometry was also chosen to ensure fatigue failure within 300 000 cycles. Given that the test frequency is very low (approx. 0.3 Hz), this was an important factor to ensure realistic test times.

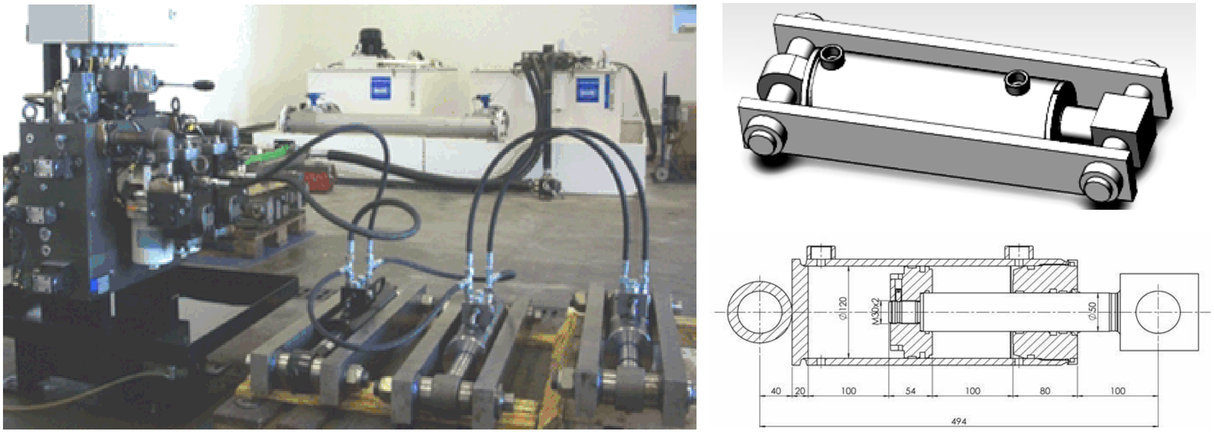

Figure 10. The SERTA test bench and the hydraulic actuator tested in fatigue by SERTA

4 RESULTS

Figure 11 shows the compassion between the numerical and experimental results. Note that the numerical values of pressure and tightening torque have been purposely omitted for reasons of confidentiality. For simplicity, the diagram has been divided into only two zones: (a) the security zone at the bottom which corresponds to an elastic shakedown state in the fatigue process zone and (b) the danger zone, in which plastic shakedown or ratcheting are predicted. The 4-pointed stars on the diagram show the experimental data for different combinations of pressure and tightening torque. The number of cycles to failure for each condition is also shown. Admittedly it would be nice to have more experimental data. However, given the long testing times, it was not possible to continue the experimental campaign. It can however be seen that, for the target fatigue life of 300 000 cycles, the proposed criterion, based simple on the concept of elastic shakedown at the macroscopic scale, results in good correlation with the experimental results.

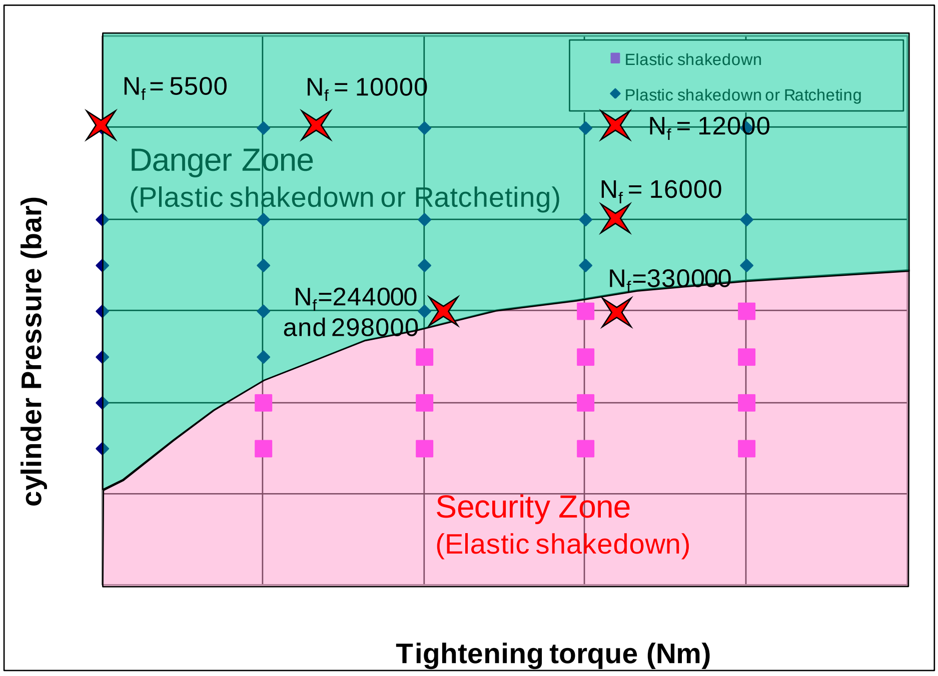

Figure 11. Comparison between the numerical and experimental results

5 DISCUSSION

As discussed in the introduction, the real aim of this work was to create a *practical* fatigue assessment method, adapted to the needs of SERTA for the day-to-day fatigue sizing of their piston/rod threaded connections, taking into account the effect of the tightening torque. It was therefore decided to create a *database* of cartographies, similar to Figure 11, taking into account the following design parameters:

a) Different values of the nominal diameter (d_2 = 24, 30 and 36mm)

b) Different values of the thread pitch (P = 2, 3, 3.5, and 4mm)
c) Different values of the shoulder width at the contact zone between the rod and piston (L_{sh} = 7, 4.5 and 2 mm)
d) Different steels
e) Different values of the coefficient of friction.

Figure 12 shows 6 different geometries investigated to date. The aim is that over time a sufficient number of cartographies will have been created so that the analysis database takes into account the complete product range proposed by SERTA.

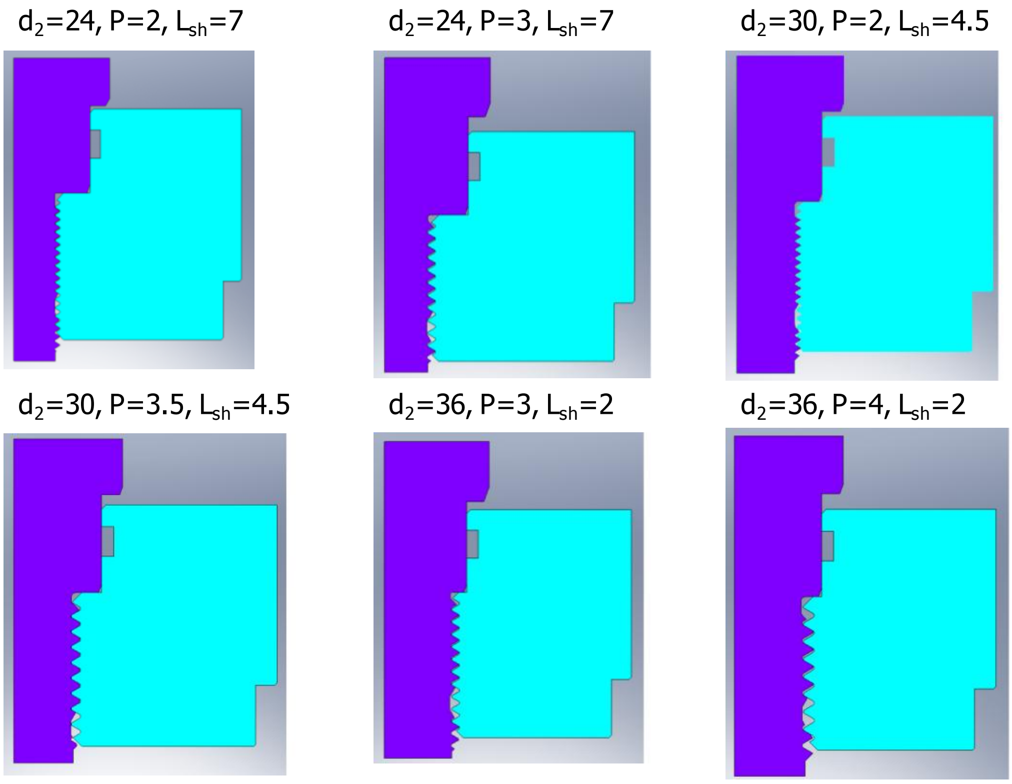

Figure 12. Six different piston/rod geometries investigated

6 CONCLUSION

A computationally intensive approach to the fatigue assessment of threaded connections has been presented, which is specifically intended for the fatigue analysis of the piston/rod connection of a hydraulic actuator.

Two-dimensional, axisymmetric, elasto-plastic finite element models are developed to determine the local cyclic stress state. Contact elements and a piston/rod geometry with an initial inter-penetration are used to model the tightening torque. A non-linear combined kinematic and isotropic hardening law (Lemaitre and Chaboche) is used to describe the cyclic plastic behavior of the material. The analysis procedure has been automated using the Ansys APDL programming language.

Full scale experimental tests have shown that a simple fatigue criterion based on the concept of elastic shakedown of the material in the fatigue process zone is appropriate for the fatigue design life in question of 300 000 cycles.

A way of presenting the results, as a series of "Cylinder Pressure verses Tightening Torque" diagrams, is proposed. These diagrams are specific to a given actuator configuration. The aim is to create a database with a sufficient number of diagrams so that the database takes into account the complete product range proposed by SERTA.

7 ACKNOWLEDGEMENTS

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8 REFERENCES

[1] AFNOR, Éléments de fixation - Assemblages vissés - Partie 1 : règles générales de conception, de calcul et de montage, NF E25-030-1, 2007
[2] German VDI guideline 2230, VDI-Richtlinie 2230, Systematische Berechnung hochbeanspruchter Schraubenverbindungen. 2003
[3] J. Guillot, Assemblages par éléments filetés, Techniques de l'ingénieur, Référence B5560, 1989
[4] F.P. Brennan and W.D. Dover, Stress intensity factors for threaded connections, Engineering Fracture Mechanics, Vol. 50, No. 4, pp.545-567, 1995
[5] L.A. James and W.J. Mills, Review and synthesis of stress intensity factor solutions applicable to cracks in bolts, Engineering Fracture Mechanics, Vol. 30, No. 5, pp. 641-654, 1988
[6] R. Schneider, U. Wuttke and C. Berger, Fatigue analysis of threaded connections using the local strain approach, Procedia Engineering 2 (2010) 2357-2366
[7] M. Ferjani, D. Averbuch, A. Constantinescu, A computational approach for the fatigue desing of threaded connections, International Journal of Fatigue, Vol. 33, pp. 610–623, 2011
[8] J. Lemaitre and J.L. Chaboche, Mécanique des matériaux solides, Paris, Dunod, 1985
[9] A. Tafreshi and W.D. Dover, Stress analysis of drillstring threaded connections using the finite element method, International Journal of Fatigue, Vol. 15, No. 5, pp. 429-438, 1993
[10] S. Baragetti, Effects of taper variation on conical threaded connections load distribution, J. Mech. Design, Trans. ASME, 124:320-329, 2002
[11] D. Taylor, Geometric effects in fatigue - a unifying theoretical model, Int. J Fatigue, 1999, 21:413–20.
[12] K. Dang Van, Sur la résistance à la fatigue des métaux, Ph.D Thesis, Sciences et Techniques de l'Armement, 47, 647-722, 1973
[13] B. Crossland, Effect of large hydrostatic pressures on the torsional fatigue strength of an alloy steel Proceeding of the international conference on fatigue of metals, Institution of Mechanical Engineers, London, 1956