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Experimental and numerical investigation of expandable tubular structural integrity for well applications

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Analysis and modelling

ABSTRACT

Purpose: The ever-increasing energy demand has forced researchers to search for new and cheaper solutions for oil and gas production. The recent development of solid expandable tubulars (SETs) has resulted in design of slim oil and gas wells. The large plastic deformation experienced by the tubular under down-hole environment may result in premature and unexpected failures. The objective of this research is to investigate the structural integrity of SET for well applications to avoid such failures.
Design/methodology/approach: In order to achieve the objectives, simulation work was carried out using finite element method and experimental tests were conducted on full size tubular for validation of numerical results.
Findings: The required drawing force for expansion under different expansion ratios, surplus deformation, variations in tubular thickness and length were estimated numerically and experimentally. The differences in values using two approaches vary from 5% to 12%. Tubular wall thickness decreases as the mandrel angle, expansion ratio, and friction coefficient increase.
Research limitations/implications: The issue of maximum expansion a tubular can be subjected to needs to be further investigated. Furthermore, the pre and post-expansion material properties need immediate attention of researchers to fulfil the dream of low-cost expandable solution.
Practical implications: In recent years, solid expandable tubular technology has already made significant inroads in replacing conventional telescopic oil wells. It allows design and realization of slim wells, accessing difficult and ultra-deep reservoirs, well remediation, zonal isolation, drilling of directional and horizontal wells, etc.
Originality/value: SET is an emerging technology for oil and gas industry. The current findings are very valuable for researchers and well engineers to design slim wells and enhance the productivity of older wells.
Keywords: Expandable tubular; FEA; Tubular expansion; Well integrity; Plastic deformation

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1. Introduction

The metal forming processes is designed to exploit a remarkable property of metal – plasticity, the ability to flow as solid without substantial changes in post forming properties. The material is simply moved or rearranged to produce the desired shape, hence reducing the waste substantially. Tube drawing, a deformation process, is an important manufacturing process which is used in the production of a wide range of products from house hold pipes to boiler and heat exchanger tubes. The issues such as induced anisotropy, changes in ductility, yield strength, etc. do not deter in utilizing it for production of many components and products. Rather, extensive research has been done to understand the changes emanating from forming processes to improve existing designs in terms of its performance, cost, durability and reliability.

Recently, the process has found an interesting application in the petroleum industry and is termed as solid expandable tubular (SET) technology. Tubular expansion is a cold working process which subjects cylindrical tubulars to permanent radial expansion either by mechanically pulling or hydraulically pushing a conical mandrel through it. Continuing research in SET may be able to provide solutions for many challenging problems, which are still unsolvable such as conservation of well hole size, ultra-deep drilling, extended reach application, hydraulic isolation of selected zones, etc. These issues are not only long-standing but also have far-reaching consequences in petroleum industry. Apart from this, there is a continuous push for performance improvement, cost reduction and innovating new techniques/tools/procedures to push toward realizing the dream of having mono-bore diameter well. All of these involve one of the industry's most fundamental technologies; well-bore tubular.

Solid expandable tubular technology has successfully addressed some of the above-mentioned challenges. Decades of research work on cold drawing [1-4] and extrusion is a proof of the concept but field readiness requires additional numerical and experimental work. Tubular forming processes such as sinking, flaring, expansion, etc. were throroughly investigated between 1950-80's. The pioneering work of Hill [1] for tube sinking was a major breakthrough in determining analytical or semi-analytical solution. Marciniak et al. [2] derived the governing equations for tubular expansion with very simplifying assumptions such as constant wall thickness and negligible axial strain in expnadble zone. Generalized slab method was used for analysis of extrusion of axi-symmetric tubes from hollow circular billets [3]. The mathematical model developed by Fischer et al. [4] for tube flaring considered the variation in pressure and thicknes within the expansion zone. The developed model was used to determine the drawing force required for conical expansion in thin walled tubes under compression. Yeh [5] conducted theoretical and simulation study to determine the relationship between punch stroke, tube thickness, expansion ratio and flaring limit. It was found out that material strain hardening has no effect on the relationship between thickness variation and exapnsion ratios. The first analytical model for slid tubular expansion using force and energy balance was developed by Ruan et al. [6]. The contact pressure between the mandrel and tubular, and the required liquid pressure to push the mandrel to expand the tubular were determined. In addition to this, post expansion performance properties such as burst and collapse pressures were calculated. The model was further developed for expansion of tubes under tension obatain optimal mandrel shape for expansion [6].

The notion of the use of expandable tubular in down-hole applications is an innovative extension. The ease of expanding slotted tubular led to the potential use of the technology as a means of maintaining borehole stability and as an alternative sand exclusion method. This classification of expandable tubular technology is termed as expandable slotted tubular. Here the expansion depends on the dimension and placement of slots and the size of expansion mandrel. The expansion process is based on bending the metal strips between two overlapping slots requiring small expansion forces, approximately 10 tons, depending on the number of pre-installed shear screws [7]. The expansion forces for solid expandable tubular are enormous; about 10-30 times that required for an average expandable slotted tubular. Carl et al. [8] reported first well remediation using expandable casing. Using SET in the upper sections of the well preserves the hole size from the onset and allows to push further the casing to fracture-gradient limit. Preserving hole size contributes to drilling efficiency and minimizes risk associated with small hole sizes in deeper sections of the well-bore [9].

As is always true with any emerging technology, the challenges assocaited with SET are many and need significant research and experimental work before it can be used to its full potential. These include accurate estimation of expansion forces, thickness and length variations during expansion, increasing the envelope of expansion, tubular's post expansion material charaterization, minimizing system instability, optimal design of cone and to develop new material for down-hole applications. Researchers have tried to use aluminimum as an expandable tubular due to its improved formability as compared to steel. However, the severity of down-hole environment excludes the possibility of using aluminum instead of steel [10-11]. Due to the complexity of the process, the researchers have focused their effort on using non-linear explicit finite element analysis to simulate expansion process [12]. The study showed the effects of expansion ratio, friction coefficient and mandrel angle on tube expansion. It was found that for small conical angles of mandrels and large expansion ratios, the failure cannot be avoided even at low values of friction coefficient. A specific application of casing damage repair was simulated using finite element method by Binggui et al. [13] to provide theoretical support and pilot test-basis for the follow-up research on using SET for repairing aging wells.

The objectives of this research project was two fold; a) to design and develop a full scale test facility to conduct tubular expansion for varying tubular sizes, materials, end-conditions and down-hole environment; and b) perform expansion simulation using non-linear finite element method. Material hardening, contact between tubular and mandrel, and geometrical nonlinearities were considred in the finite element analysis. The force or pressure required for expansion, resultant thickness and length reductions, surplus deformation were estimated, which can be readily used by field engineers to improve the structural integrity of oil wells. Good agreement was obtained between experimental and numerical results. The finite element models were caliberated using experimental data, which can be used for further analysis of post expansion strength charaterization. In section 2 we define the expansion process, while section 3 and 4 describe experimental setup and the finite element model. Results and conclusions are given in sections 4 and 5, respectively.

2. The expansion process

The underlying principle, in solid expandable tubular technology, is to put the tubular into an oil-well, and deform it to a larger diameter by hydraulically pushing or mechanically pulling a mandrel through it as shown in Figure 1.

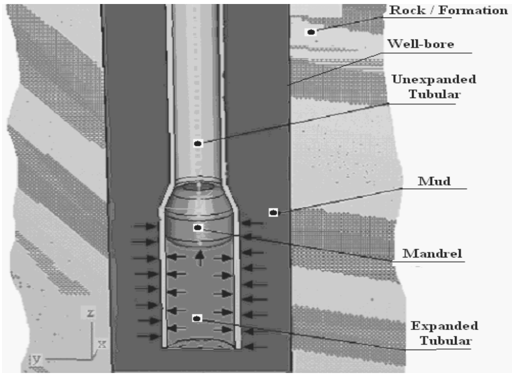


Fig. 1. Tubular expansion in an oil-well [14]

The hydraulic force is applied by pumping through a work-string connected to the mandrel and the mechanical force is applied by either raising or lowering the work-string. The progress of mandrel deforms the tubular to the desired inner and outer diameters, while preserving its integrity. The tubular deforms beyond its elastic limit into the plastic region but remains below its ultimate tensile strength. Expansion of over 25% of the initial tube diameter has been successfully achieved during full-scale lab tests. It is also possible to mechanically pull the mandrel to expand the tubular, but is usually avoided due to operational difficulties. The expansion can be performed both in open and cased holes. In the first case, the tubular is expanded against the formation, while in the second case, expansion is done against a casing previously installed in the well. The expansion ratio is defined using mandrel and pre-expanded tubular diameter:

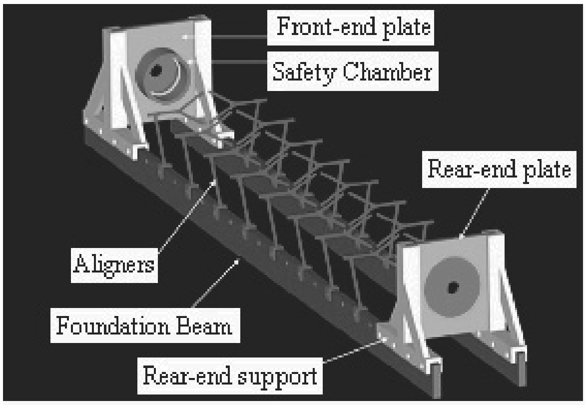
$$\text{Expansion ratio} = \frac{OD_{Mandrel} - ID_{Pre-expanded}}{ID_{Pre-expanded}} \; x \; 100$$

Here OD and ID stnad for outer and inner diameters of the tubular, respectively. For a down-hole expansion process, the tubular must be able to expand to the desired diameter without fracturing, bursting or damaging the tubular. It will result in thickness and length variations. It must be able to maintain hydraulic capabilities to provide sufficient resistance against burst and collapse during service. The expanded tubular should have constant diameter and wall thickness over the whole length of expanded section and should maintain the integrity of expanded tubular connections. Another challenging task is to expand long sections without causing failure to the expanded zones. Since the tubular, as well as the mandrel, experience high interfacial stresses, as the expansion proceeds, the selection of mandrel geometry and material is also crucial. The shape of the mandrel also plays a critical role for successful completion of expansion process. Furthermore, knowledge of post expansion mechanical properties/characterization of tubulars is important for proper utilization during service.

3. Experimental setup for expansion

A full scale experimental facility, as shown in Figure 2, for expanding tubulars ranging from 4” to 10.625” diameter with expansion ratios varying from 8% to 30% has been designed and commissioned in the Engineering Research Laboartory at Sultan Qaboos University. Expansion length ranges from 2 to 10 meters. In case of hydraulic expansion, the maximum test pressure of 2000 bar with a flow rate of 11 liters/min can be attained. For mechanical pull, the expansion force can be varied from 10 to 140 metric tons. Instrumentation and control system is also designed to monitor, control and store data for different system variables. These variables include strain, displacement, expansion force, tubular thickness and length variations, operating fluid temperature, flow rate, and speed and location of mandrel during expansion. The setup is flixible to accomodate various down-hole environment and tubular end-conditions to mimic an oil-well. Pre and post expansion properties of tubular are obtained using ASTM standard test methods. These include; hardness, Young’s modulus, yield and ultimate tensile strengths, ductility and strain at fracture.

a)



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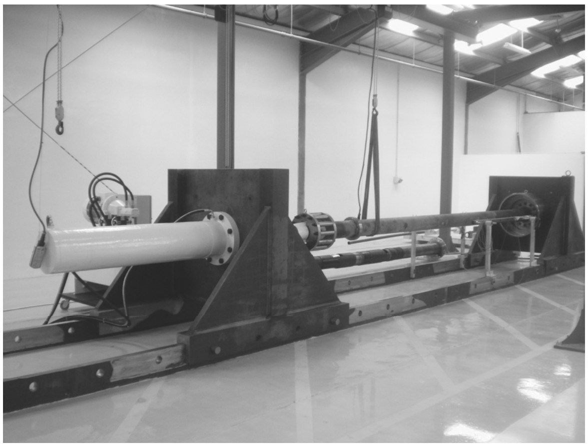


Fig. 2. Solid expandable tubular test setup: a) CAD image of the test setup, b) test setup with expandable tubular ready for test

4. Finite Element Modeling

Finite element analysis of tubular expansion was performed using ABAQUS software. In reality, the tubular expansion is a dynamic process, but due to very low velocity of expansion mandrel, the process was modeled as a quasi-static process. Simulation was done only for vertical oil wells. In case of vertical well, the model consist of expandable tubular and mandrel. The finite element model was created by building up the geometry of the solid tubular and mandrel, inputting their corresponding material properties and imposing appropriate boundary conditions. Due to symmetry only one half of the tubular was considered as shown in Figure 3.

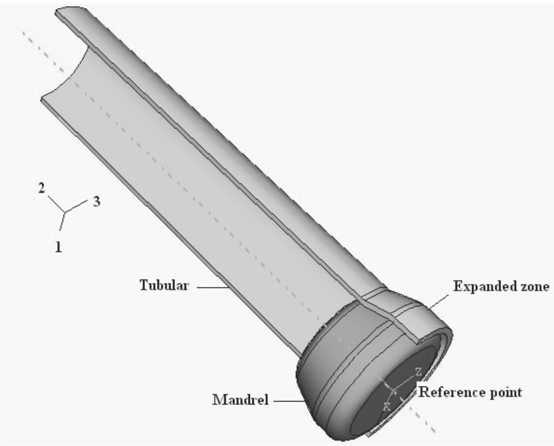


Fig. 3. Tubular-mandrel model used for simulation

All nodes lying on the plane of symmetry were constrained in the direction perpendicular to the plane. In addition, the left end of the tubular was held fixed hence the tubular expands under tension. The right end of the tubular was kept free. This particular condition was used to compare the simulation results with experimental data. The tubular was modeled as a deformable body using 8-node linear brick elements with reduced integration. The material properties of the tubular were obtained based on the uniaxial tensile test conducted on specimens following ASTM standard test methods. Three samples were tested in tension on Universal Testing Machine till it fractured. The modulus of elasticity, yield, ultimate and fracture strength, ductility, and strain at fracture were measured. The stress-strain curve for three samples is shown in Figure 4. Average properties based on these samples were used in simulation work.

The mandrel was modeled as an analytical rigid body with a reference point to control its displacement and permit the expansion. The edges of the mandrel, where contact takes place, were smoothed to avoid any possibility of inducing stress concentration. The cone angle was set at 10 degrees based on the knowledge of previous work done as well as to compare simulation results with experimental data. The mandrel moved from left to right at a fixed low speed of 1.5 m/min. This was achieved by defining a reference node at the center of the mandrel which follows a pre-defined path during expansion process. The mandrel was made of hardened tool steel. No material properties of madrel was needed due to the assumption of rigid body. Due to low mandrel speed, the strain rate effect was assumed to be negligible. All six degrees of freedom for the reference node of the mandrel were fixed.

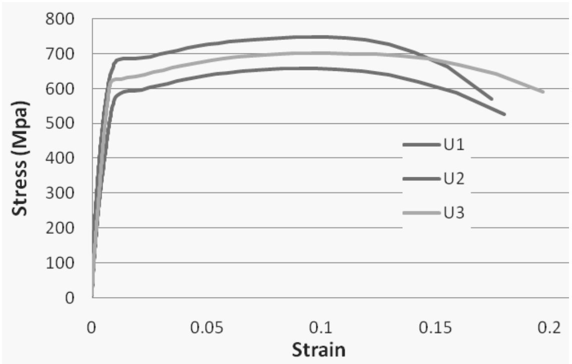


Fig. 4. Stress-strain curve for uniaxial test of tubular specimens

The contact between the mandrel and inner surface of the tubular was modeled using Coulomb friction law to account for the induced friction between the interacting surfaces. The coefficients of friction were set between 0.07 and 0.1, as provided by the manufacturers of the solid expandable tubular. The tubular were coated with special proprietary coating to have this low value of friction coefficient. Simulations were performed without adaptive mesh due to the filleted edges for both the mandrel and tubular.

5. Results and discussions

In accordance with field practice and recent studies, the tubular material of API (American Petroleum Institute) grade L-80 hardened steel was considered. The stress-strain curve for this material is shown in Figure 4. A standard API tubular of 500 mm length, outer diameter of 193.675 mm and wall thickness of 9.525 mm was used for both experimental and simulation study. The expansion ratio was varied from 10% to 25% as given in Table 1. The stress-strain curve for this material is shown in Figure 4. A standard API tubular of 500 mm length, outer diameter of 193.675 mm and wall thickness of 9.525 mm was used for both experimental and simulation study.

Table 1.
Pre and post inner diameter and expansion ratio

Pre-ID	Post-ID	Expansion ratio
6.875	7.750	12.73
6.875	7.875	14.50
6.875	8.000	16.36
6.875	8.250	20.00
6.875	8.375	21.81
6.875	8.500	23.64

Force required for expansion, equivalent plastic strain, effective stress, contact force, thickness and length variations, and surplus deformation were extracted from output database file of simulated cases. The experimental tests yielded expansion force, thickness and length variations, surplus deformation and the movement of the mandrel. However, following results were compared with available experimental data and are reported in this paper.

- Expansion force/pressure required for expansion;
- Variation in tubular length and thickness;
- Surplus deformation.

During experiment two pressure sensors were mounted at the mandrel end of the tubular along its radius. The measured pressures from these two sensors is shown in Figure 5. The maximum pressure of 341.15 bar was measured when the mandrel passed through the welded section of the tubular as indicated in Figure 5. The pressure was below 20 bar in the pre-mandrel section of the tubular. This section was already expanded using heavy duty mechanical press to push in the mandrel. Later this end was closed using blind flange having a 4mm diameter hole to pump the fluid at high pressure for expanding the post mandrel section of the tubular. The average measured expansion pressure required for 20% expansion of the tubular withoud any joint fluctuated between 300 to 305 bar. In case of a weld joint, an increase of 15% was experienced in pressure required for tubular expansion. Similar observations were made for expansions at other expansion ratios considered in this study.

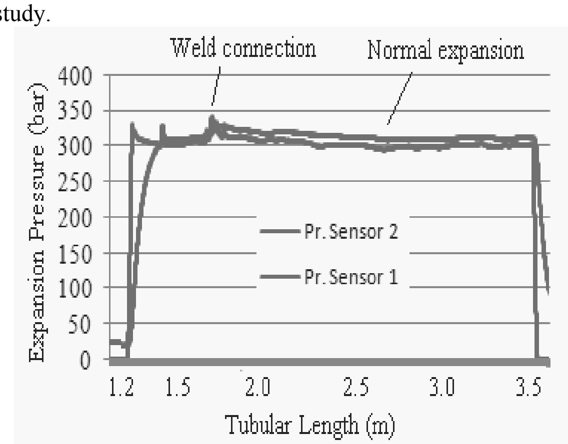


Fig. 5. Variation in expansion pressure as mandrel travels along tubular length

Figure 6 shows the expansion pressure required for 20% expansion ratio obtained from experimental and simulation data. It is clear from the figure that the expansion pressure obtained through simulation is in good agreement with experimental results with an error of less than 4% compared to the experimental data. In all cases, the expansion pressure/force initially reaches a peak value (representing the initial upsetting process) and then drops down to an almost steady-state value. For instance, in the case of 10° mandrel cone angle, 20% expansion ratio, and a friction coefficient of μ=0.1, the initial peak is around 340 bars and then steady-state value of 300 bars is maintained during rest of the expansion process. The small fluctuations in simulated expansion pressure are due to the transient behavior but overall the expansion process is stable.

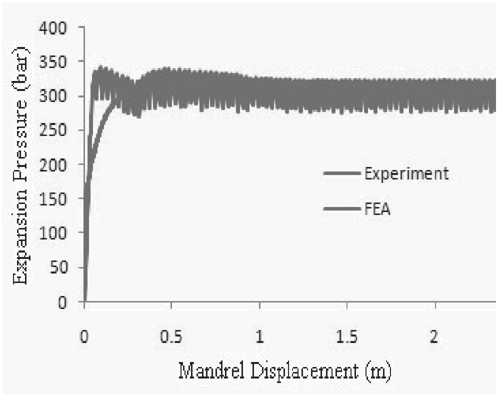
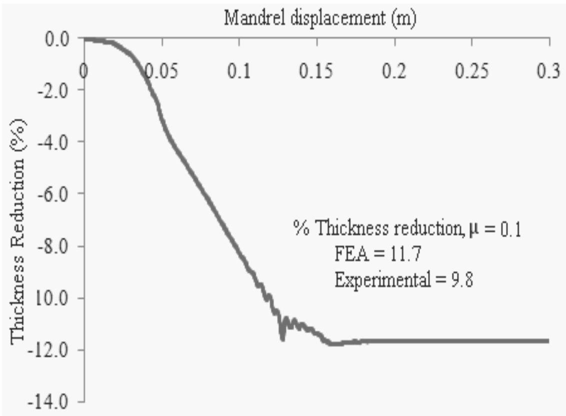


Fig. 6. Variation in expansion pressure along tubular length (20 % expansion ratio)

a)



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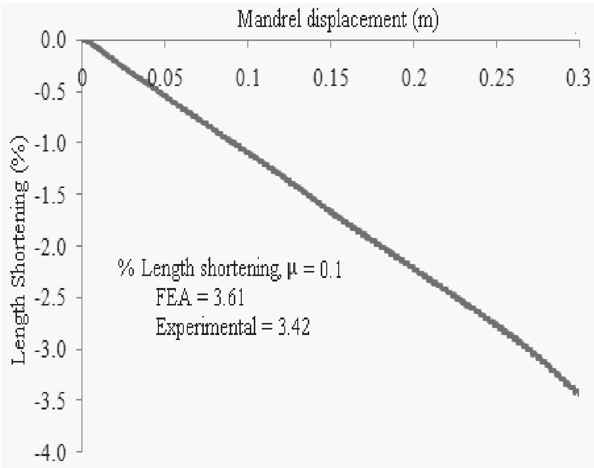

Fig. 7. Thickness (a) and length (b)variations for 20% expansion ratio

Figure 7 shows the reduction in tubular thickness and shortening of its length for 20% expansion ratio with a friction coefficient of 0.1. Tubular thickness, before and after expansion, are measured at four different locations, and an average value of thickness reduction is calculated for comparison with simulation results. Similarly, the length shortening are measured at the begining and end of the tubular. Again, the average value is calculated for comparison purpose. There is an excellent agreement in case of length shortening, while for thickness reduction the difference is more than 10%. These information helps in better understanding of the down-hole expansion process, which ultimately could lead to chalk out proper guidelines to achieve desired expansion without failure.

Figures 8-10 show the variation in force required for expansion, and thickness and length variations as a function of expansion ratio. The expansion force required for tubular expansion increases with an increase in expansion ratio. This is also true for thickness reduction and length shortening. The expansion force varies non-linearly with expansion ratio as opposed to thickness and length variations, which vary linearly. For 25% expansion ratio, the expansion force required (126 tons) is well within the allowable limit (200 tons) of an oil-well rig platform. But the thickness reduction of 10% and beyond results in lower collapse strength rating. This may lead to premature failure of the expandable tubular during its service life. In other words, it is the reduction in thickness, which will limit the maximum allowable expansion in SET and not the expansion force.

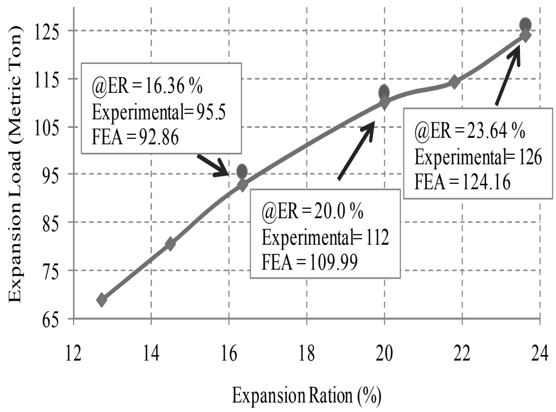

Fig. 8. Variation in expansion load w.r.t expansion ratio

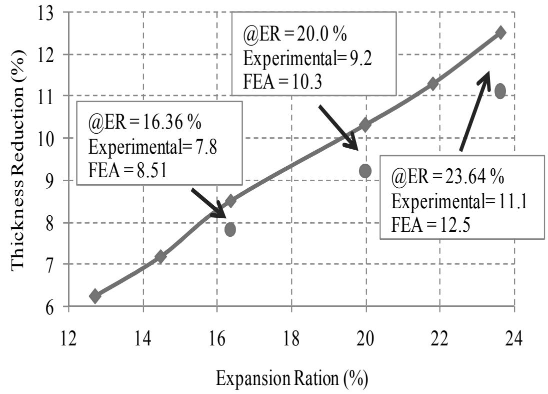

Fig. 9. Variation in thickness reduction w.r.t expansion ratio

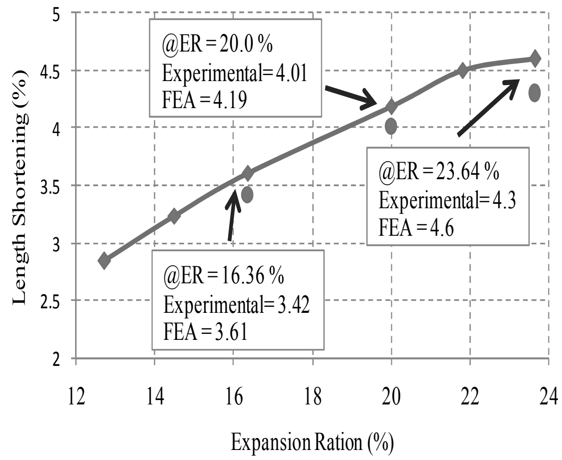

Fig. 10. Variation in length shortening w.r.t expansion ratio

A premimum API threaded connection (VAM type) between two 7.625" outer diameter solid expandable tubular was tested for expansion ratios of 20 and 23.6 %. The measured force required for 20 and 23.6 % expansion ratio and thickness reduction is given in Table 2. The expansion force is significantly higher in magnitude as compared to the normal weld connection (an increase of 100 percent compared to 15 percent increase in welded connection). The thickness reduction for 20 % expansion is similar to that of weld connection. However, the test failed to complete for 23.6% expansion. The tubular ruptured because the expansion pressure increased beyond the burst strength of the tubular.

Table 2.
Experimental result for API connections

Expansion Ratio (%)	Expansion Force (tons)	% Thickness Reduction
20.0	240	10.4
23.6	277*	*

*: Experiment did not succeed and tubular ruptured

It is believed that the expansion process will not yield the exact final diameters as per designed value. There will always be variations in final outer and inner diameters due to the inherent dynamics of the mandrel/tubular system as well as the nonlinear material behavior. The difference in tubular final and desired inner diameters when expressed in percentage of desired inner diameter is termed as surplus deformation. Figure 11 shows the behavior of surplus deformation (%) along the tubular length for various values of friction coefficients. Surplus deformation is smaller in magnitude as compared to length and thickness variations. As the variation band is really small, the effect of friction coefficient and expansion ratios on surplus deformation is trivial.

Surplus deformation should be kept to a minimum in the field because its existence means that the actual expansion is more than the target value. Correct expansion can be achieved by selecting an expansion ratio slightly lesser than the desired value. The selection of appropriate initial values of expansion ratios to achieve desired post-expansion ratios requires another set of simulations. This may be targeted in a future study.

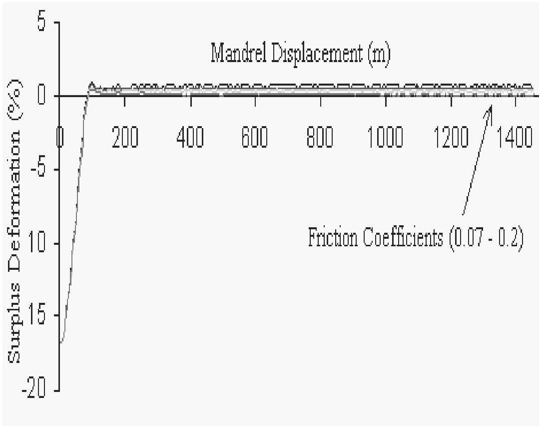

Fig. 11. Variation in surplus deformation along tubular length for friction coefficients between 0.07 and 0.2

Figure 12 shows the variation in expansion force required for expansion with respect to the mandrel cone angle for expansion ratios of 15%, 17%, 20% and 22%. The friction coefficients of 0.1 was used. It is obvious from the figure that the minimum expansion force is required when the mandrel cone angle is around 20 degrees. Similar conclusion was obtained for friction coefficient of 0.2. Nevertheless, the expansion force increases with an increase in coefficient of friction irrespective of variation in mandrel cone angle.

Fig. 12. Variation in expansion force with respect to mandrel cone angle for various expansion ratios

Simulations were also carried out to investigate the effect of mandrel rotation on estimated force required for expansion and thickness variation as shown in Figure 13. Simulation was done for 20° mandrel cone angle, 10% expansion ratio and friction coefficient of 0.1. The mandrel was rotated at 10 rpm.

As can be observed from Figure 13, the required expansion force decreased considerably due to rotating mandrel. This could be due to the screw effect of the rotating mandrel. However, one must keep in mind that proper lubrication and additional power is needed to rotate the mandrel. The lesser magnitude of expansion force for a rotating mandrel can be counterbalanced by the extra power required for rotation. One has to estimate the total power requirement for the system to have a justifiable comparison between rotating and non-rotating mandrel. Contrary to this, 40% thickness reduction was estimated for rotating mandrel, which will definitely lead to failure due to reduction in collapse and burst strengths. A similar observation, but of lesser magnitude, was observed for length variation As pointed out earlier, higher magnitudes of variations in length and especially in thickness may cause structural instability in expanded tubular during its service life.

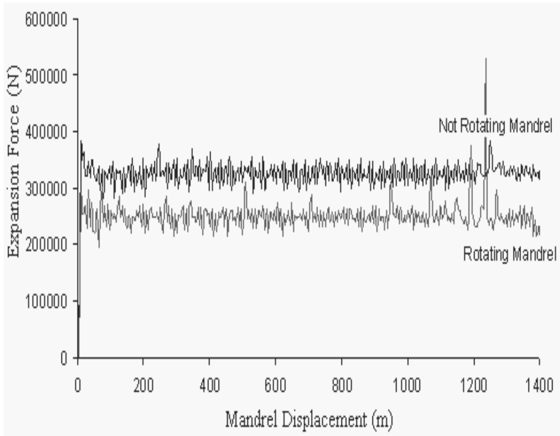

Fig. 13. Variation in expansion force for rotating and non-rotating mandrels

6. Conclusions

A solid expandable tubular test-rig was designed and commissioned to conduct tubular expansion tests similar to the down-hole environment. An API tubular of grade L-80 has been selected for experimental work. A number of tubular expansions were done experimentally as well as simulated using finite element method. Experimental and simulation results were obtained for various conditions to study the effect of tubular expansion ratio, mandrel cone angle, and friction coefficient on the expansion force/pressure, length shortening, thickness reduction, and surplus deformation. Excellent to good agreements were found between experimental and simulation results, particularly for expansion force and length shortening. It was found that the drawing force increases with an increase in mandrel cone angle, expansion ratio, and friction coefficients at tubular/mandrel interface. The expansion force for a weld connection was approximately 15% more than a normal tubular expansion. However, in case of a typical API threaded connection (VAM type), the increase in expansion force was two fold. The wall thickness of expanded tubular decreased linearly as the expansion ratio and friction coefficient were increased. Tubular length shortens under fixed-free end conditions and varies linearly with expansion ratio. The expansion force required for expansion using a rotating mandrel is 15% less than that of a non-rotating mandrel having similar configuration. However, tubular thickness reduction increased by 40% for a rotating mandrel as compared to a non-rotating one.

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