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Application of Optimisation Methods to the Design of High-Performance Composite Pipelines*

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Abstract

This paper describes a contribution to the research carried out by a consortium under the BRITE/EURAM programme of the EC on the application of optimisation techniques to problems of engineering design using composite materials; one area addressed is that of gas transmission pipelines.

The candidate pipeline is filament wound using a glass reinforced epoxy resin and is designed for a gas transmission line with an operational pressure of 70 bar. A factor of safety of 4.5, approximately, is used. An innovative coil-lock joint is shown to successfully perform the connection between pipeline sections. The main problem is that the highest stresses occur at the transition between pipe and joint.

Optimisation software is used as a tool for improving the pipeline design. The optimisation is applied to the pipeline in the area of the joint. The objective function is to minimise the weight of the pipeline. Pipeline to joint transition geometries, as well as overall pipe thickness are optimised. Maximum structural stresses are used as constraints in the optimisation process. Results show a significant reduction in peak stresses at pipeline to joint transitions and a minimisation of pipeline mass.

Keywords: Composite; Pipeline; Joint; Optimisation

INTRODUCTION

Previous research by the authors [1] concentrates on the evaluation of different optimisation techniques with respect to the design of high pressure composite pipelines. The main parameters addressed are those of pipe thickness, winding angle, layer numbers and shape optimisation with respect to joint configurations. The objective of the present work is to apply the selected optimisation algorithms to a specific pipeline and associated joint design.

The candidate pipeline is filament wound using a glass reinforced epoxy (GRE) resin and is designed for a gas transmission line with an operational pressure of 70 bar with a factor of safety of 4.5, approximately. The internal diameter is 105.2 mm and the pipe has a wall thickness of 11.5 mm. The joint is a patented [2] design incorporating an innovative spiral locking mechanism based on a tapered helical thread and an integral ductile helical key.

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Details of the pipeline design and some preliminary failure data from pressure tests are outlined in the following section. Two-dimensional finite element analyses using the SAMCEF [3] code are then described which serve to identify possible areas of high stress concentrations and scope for optimisation of the design. Numerical and test results indicate that shape optimisation of the transition between pipeline and joint is the most promising area for enhancing the design. Studies executed using optimisation software named BOSS [4] are outlined and show that a significant reduction in peak stresses at the transition may be achieved.

PIPELINE DESCRIPTION

The GRE pipeline and jointing system selected for the purposes of illustrating the optimisation procedures are based on an internal diameter of 105.2 mm with actual dimensions as shown in Figure 1. The pipes are manufactured using a continuous roving of E-glass fibres in a matrix of epoxy resin wound on a cylindrical mandrel at an angle of ±54º. The uniform section of the pipe has a nominal thickness of 11.5 mm with a varying thickness in the joint region.

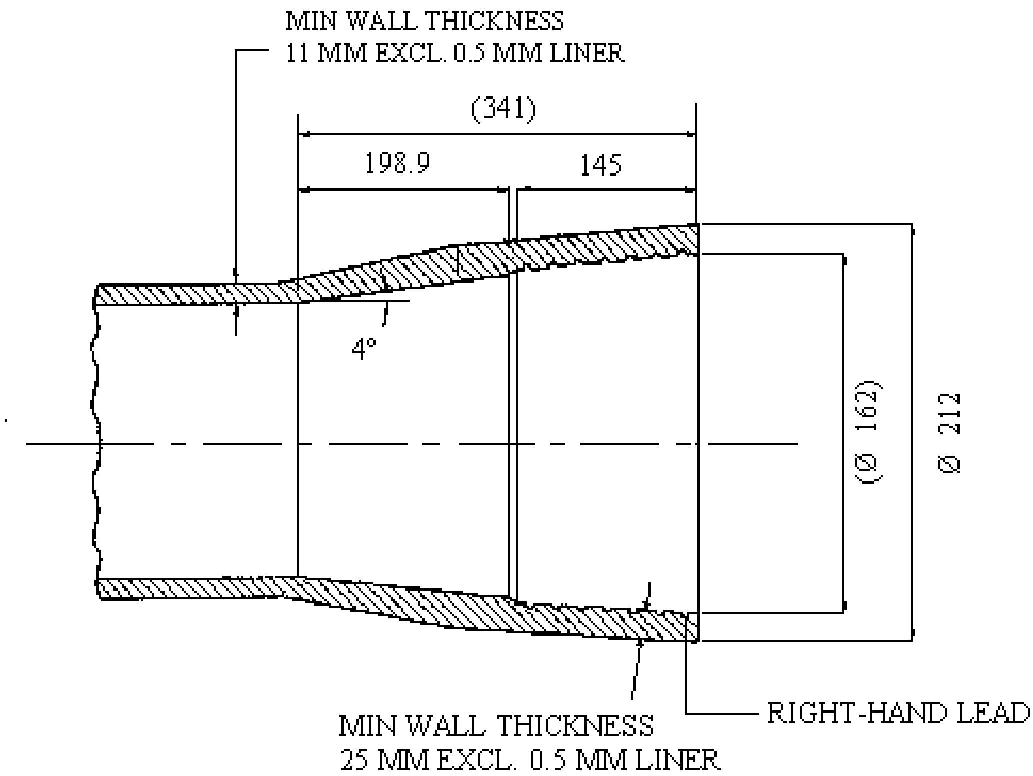
The pipes are joined using a coil-lock threaded connection which combines a tapered lap joint with an integral ductile helical key. The arrangement is illustrated in Figure 2 and is similar to that described in [5]; this joint is covered by a patent [2]. The joint may be sealed utilising dual mechanical seals and, additionally, an adhesive sealant may be injected after assembly between the seals for improved leak-tight integrity. The joint may be tightly assembled with just a few rotations of the pipe due to the taper and large pitch of the threads.

Assuming a constant winding angle of ±54º for the fibres, the material properties are taken as orthotropic and constant along the pipe and are listed with respect to the pipe axes in Table 1; this particular set of values are taken from [6].

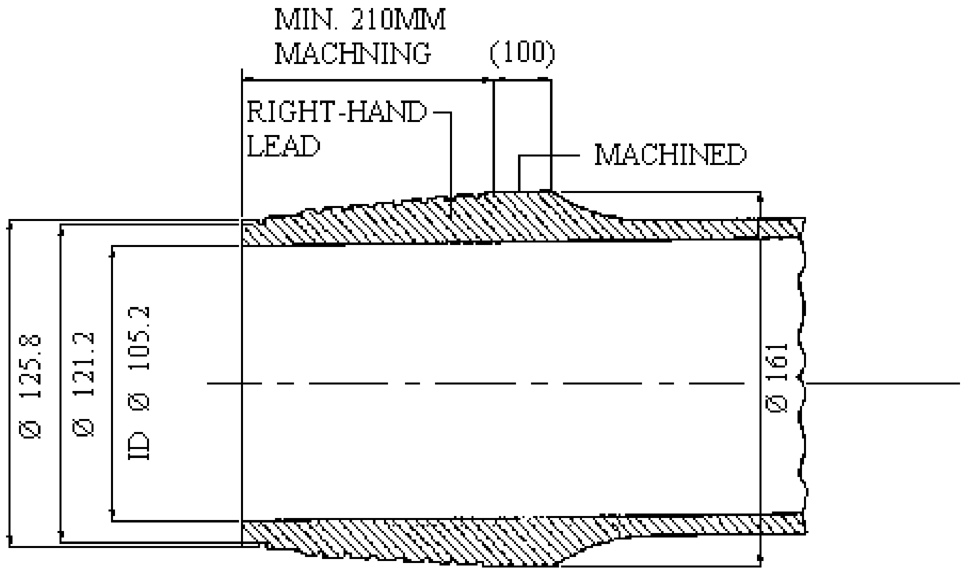
Table 1
Material properties for the pipe and joint

	Radial	Axial	Circumferential
Young’s modulus (MPa)	2000	10000	25300
Ultimate strength (MPa)	65	65	300
Poisson’s ratio	0.35	0.4	0.49
Shear modulus (MPa)	741	3471	8490

The pipeline loads are listed in Table 2 and are consistent with those for a gas pipeline. A factor of safety of 4.5 approximately is imposed on the operating pressure of 70 bar in order to allow for the degradation of mechanical properties over a projected lifetime of 20 years. The axial load of 269 kN occurs due to the endcap effect.



(a)



(b)

Fig. 1. Coil-lock pipe/joint system showing (a) female component and (b) male component

Pressure testing of pipe and joint

The pressure testing of samples of the GRE pipeline and jointing system is described in [7]. The jointed sections were approximately 2 metres in length and were manufactured by Ameron BV. The objectives of the tests were to validate the mechanical performance of the pipe and joint and identify any failure modes. Failure pressures in the range from 280 bar to 320 bar were recorded and these occurred at the transition between the uniform pipe section and the joint taper in the female end. The mode of failure appeared to be due to axial stresses in the pipe wall.

Table 2
Pipeline design loads

	105.2 mm pipe
Internal pressure (bar)	310
Axial force (kN)	269

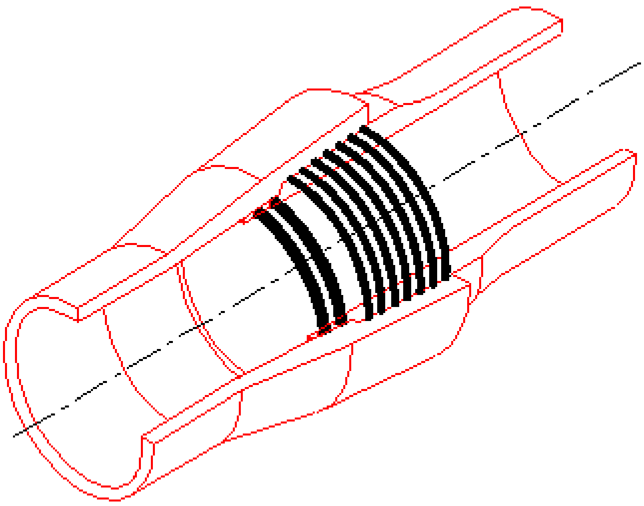


Fig. 2. Schematic overview of a coil-lock joint suitable for high pressures

FINITE ELEMENT MODELLING OF PIPELINE AND JOINT

A finite element analysis is undertaken to study the stress distribution across the joint section. The meshes in the area of the female and male sections are given in Figures 3 and 4 respectively. The male section is shown in the darker colour, while the female section is shown in the lighter colour. Four geometrical assumptions are present in the finite element model and these are listed as follows. Firstly, the joint structure is assumed to be axisymmetric. Secondly, the male and female sections of the pipeline are modelled as a single structure, so that they share nodes at their intersection. This makes the analysis much more efficient than if the two sections were defined independently and contact between them was simulated. Thirdly, the helical locking key mechanism is omitted at the intersection of the male and female joint sections. These simplifications can be justified by noting that the most critical stresses were experimentally determined to occur at the transition between the uniform pipe section and the joint taper in the female section of the joint. In other words failure occurred in the pipe rather than in the joint. Finally, the effect of the O-ring seal is negligible and so is neglected from the analysis. Therefore, a small square shape in the model, is left unmeshed as shown in Figure 3.

The pipeline and joint system is modelled with 4-noded axisymmetric elements. Pipe sections of approximately 300 mm are modelled on either side of the joint. In Figure 3 the mesh is more refined in the area of transition between the pipe section and the female joint section. The mesh is also more refined around the position of the O-ring seal. In Figure 4 it can be seen that a more refined mesh is given in the area of transition between the pipe section and the male joint section.

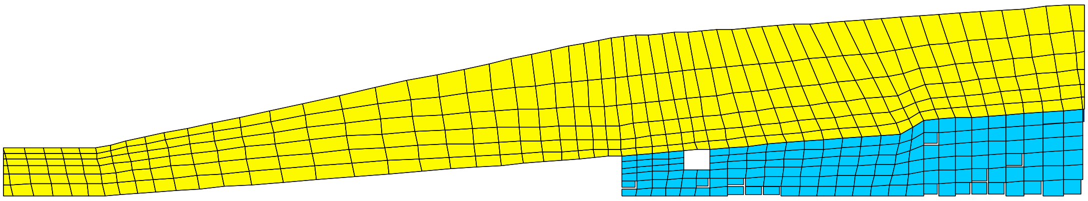


Fig. 3. Finite element mesh of female joint section region of pipeline

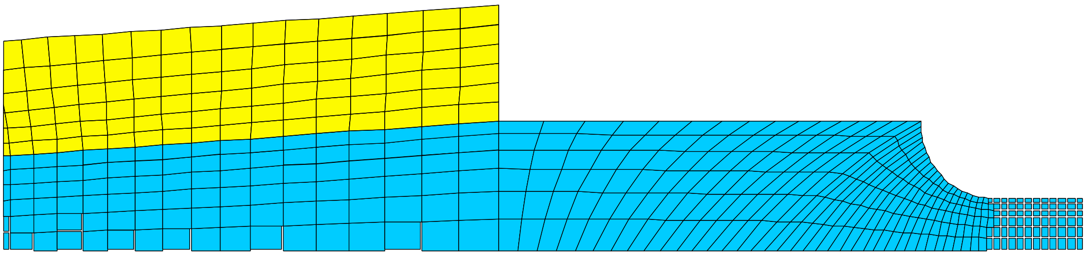


Fig. 4. Finite element mesh of male joint section region of pipeline

The loading listed in Table 2 is applied to the finite element model. The internal pressure is applied to the internal surface of the pipeline, including both the pipe and joint regions. One end of the pipe is constrained from axial motion, while the other is subject to the axial load due to the end-cap effect of the internal pressure. A linear static analysis is then carried out to determine the location and magnitude of the critical stresses.

Results of finite element analysis

Results of the finite element analysis of the initial pipeline design indicate that the most critical stresses are those in the axial direction. The axial stress in the uniform pipe sections is approximately 63 MPa. Two areas of stress concentration are found, which significantly exceed the axial stress limit of 65 MPa. The highest axial stress of 119 MPa occurs at the transition from the pipe section to the male component of the joint. This is highlighted in the contour plot of axial stresses shown in Figure 5, where the area of maximum axial stress is circled. A similarly high stress of 108 MPa occurs at the corresponding transition from the pipe section to the female component, and is shown in Figure 6.

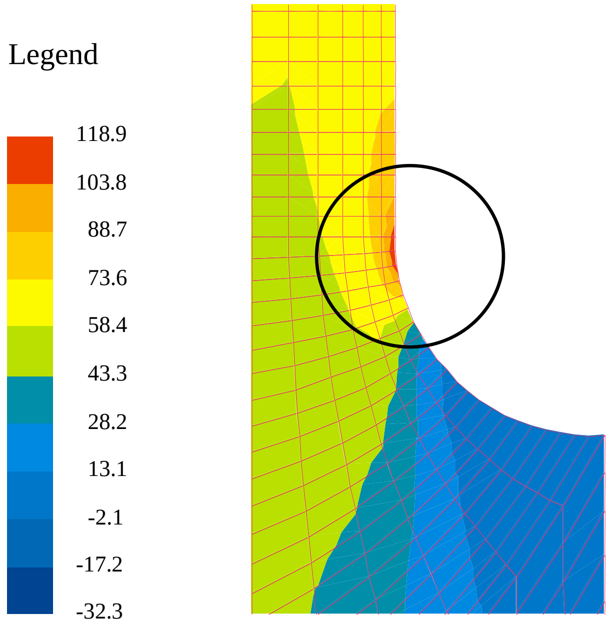


Fig. 5. Axial stress (MPa) contour plot in area of transition from pipe to male section

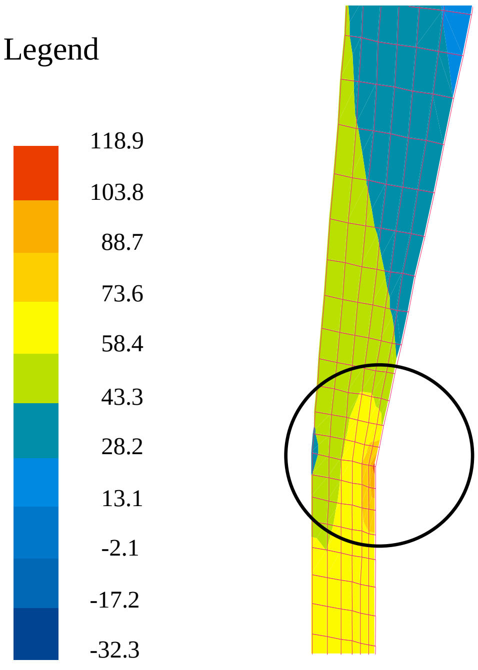


Fig. 6. Axial stress (MPa) contour plot in area of transition from pipe to female section

DESIGN OPTIMISATION PROCEDURE

Both experimental testing and numerical modelling of the pipe joint have identified that the failure region occurs in the transition from the joint to the pipeline. The geometry of the transition plays a key role in the level of stress concentration in this region. Optimisation of the geometry can lead to a reduction in the peak stresses while at the same time keeping the overall increase in weight of the transition to a minimum. Additionally, it is possible to minimise the weight of the pipeline itself by optimising the overall thickness.

The optimisation process starts with the selection of an initial set of values, x_i, for the design variables. The design variables are those quantities, such as thickness or geometric parameters, which are allowed to change during optimisation. The objective function is the quantity, such as the structure weight, which is to be minimised during the optimisation. This quantity is a function of the design variables. The minimisation is subject to design constraints such as stress or deflection limits. The design process is iterative incorporating both a finite element analysis and an optimisation procedure in each cycle and is shown schematically in Figure 7.

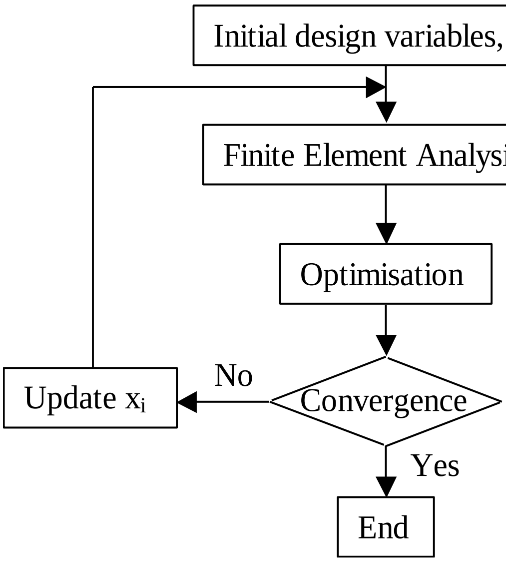


Fig. 7. Design optimisation loop

Having input the initial design variables, a finite element analysis is carried out to provide values for the objective function, the constraints and their derivatives. These values are then used by the optimisation routines to provide updated values of the design variables. These new values are resubmitted to the finite element program and the cycle repeats itself until convergence is achieved.

A wide range of optimisation algorithms is available. Arising from the studies carried out in [1], the Conlin algorithm [8] is deemed to be the most efficient for the shape and thickness optimisation of the pipeline. This algorithm is based on the Approximation Concepts Approach [9], in which the objective function and the constraints are replaced by a sequence of explicit approximated sub-problems. Conlin uses a linear approximation when the corresponding first derivatives are positive and an inverse approximation when the first derivatives are negative. This ensures that the approximation is convex. The convex, separable sub-problems are then solved in the dual space.

NUMERICAL RESULTS

The optimisation process can be used to obtain an improved geometry for the pipeline; one that will significantly reduce the peak stresses described in Section 3.1. The shape of the transition from the pipe section to the male joint section is to be optimised, as an example. The objective function is a minimisation of the total pipeline mass, while the constraint is that the axial stress does not exceed a certain value. In order to achieve this a single design variable Y is used to control the shape of the transition curve from the pipe to the male joint section. This approach is illustrated in Figure 8.

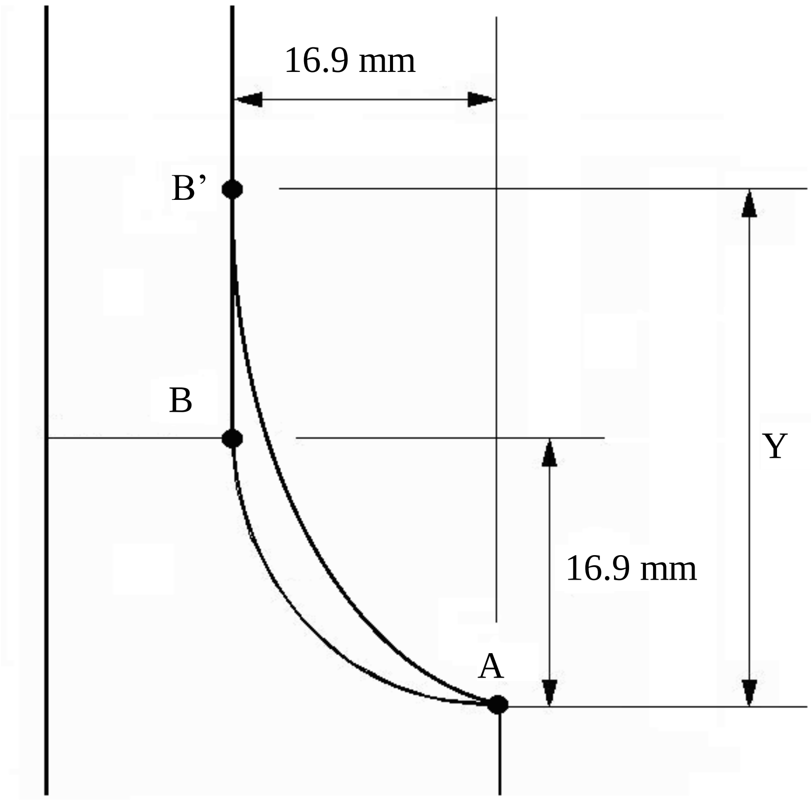


Fig. 8. Design variable Y controlling shape of transition from pipe to male section

Point A represents the start of the transition from the male joint section to the pipe and has a fixed position. Point B represents the end of the transition from the male joint section to the pipe, and lies somewhere along the pipe outer wall, above point A. The transition curve joins points A and B. The initial transition length is 16.9 mm. Point B' indicates a new position for point B, while the design variable Y indicates a new transition length. Increasing the value of the transition length from 16.9 mm to Y will give a more gradual transition, as shown in

Figure 8. Two types of curves are investigated in order to join the two points A and B. The first is a tangent arc which is defined by containing point A and by being tangent to the pipe outer wall at point B. The second curve is an ellipse defined by a semi-minor axis of length Y and a semi-major axis of length 16.9 mm which remains fixed.

Parameter study

Before carrying out the optimisation procedure it is useful to perform a parameter study. This examines the variation of the axial stress and total pipeline mass with the design variable Y, for both the tangent arc and elliptical shaped transition curves. Results of the axial stress parameter study can then be used to decide on the constraint value to set for the optimisation procedure. Figure 9 shows the variations of total mass for both the transition curves. This is the mass of the finite element model. It can be seen that the elliptical curve produces a lesser increase in total mass than the tangent arc curve for a given increase of the design variable Y. Figure 10 shows the variation of the maximum axial stress for both types of transition curves. It can be seen that the axial stress is more sensitive to an increase in Y for the case of the elliptical shaped transition. The maximum axial stress settles to a value of approximately 67 MPa, which is close to the stress in the uniform pipe section, and therefore no longer represents a significant stress concentration.

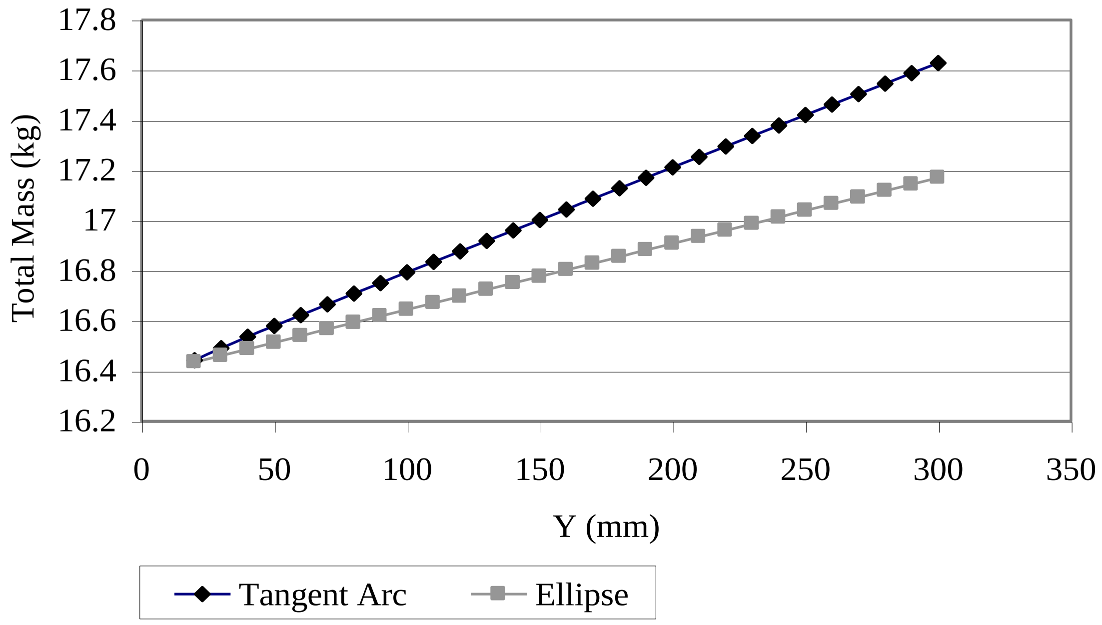


Fig. 9. Variation of total mass for different transition geometries

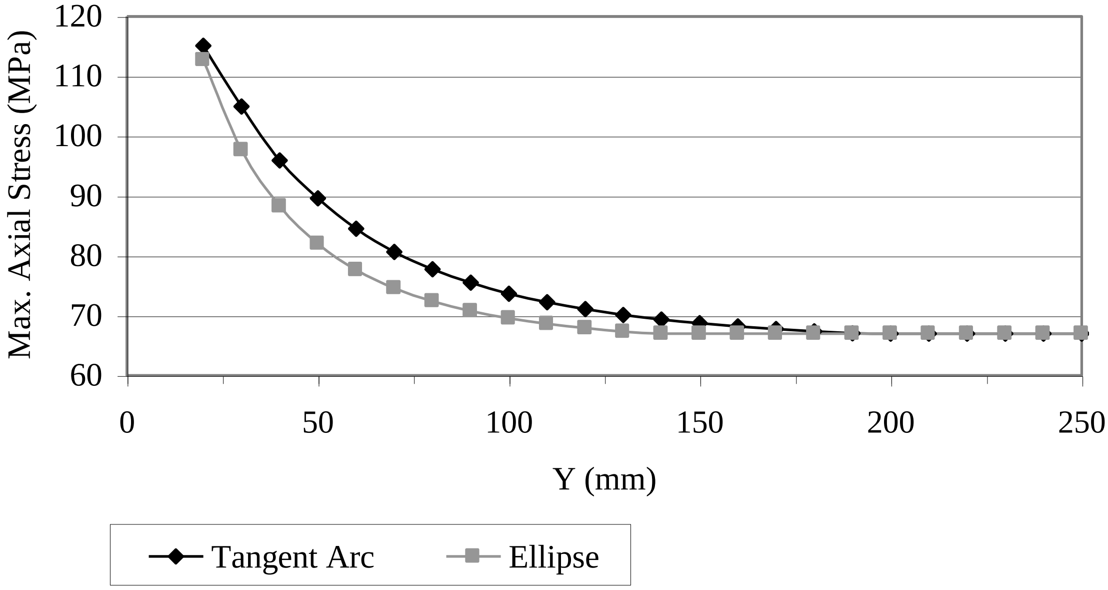


Fig. 10. Variation of maximum axial stress for different transition geometries

Shape optimisation

The optimisation process is carried out in order to determine a value of the transition length Y. The objective function of the optimisation process is to minimise the total mass of the pipe and joint system. It is desired to bring the maximum axial stress to within the stress limit of 65 MPa. Considering both transition geometries, the minimum stress achievable with the design variable is approximately 67 MPa and occurs at the maximum transition length of 300 mm. However, transition lengths of between 120 and 170 mm will give an almost identical stress of 68 MPa, but with reduction in total mass of at least 0.3 kg from the maximum transition length. A constraint value of 68 MPa is deemed most suitable. A marginal reduction in the operating pressure of 70 bar is therefore required in order to keep the axial stress below the limit. The optimisation process is carried out for both the tangent arc and elliptical transition curves. In order for the optimisation process to start the user must select an initial value for the optimisation design variable, which in this case is the transition length Y. This selection is aided by examining the results of the parameter studies for the design variable Y, which are described in the previous section. Figure 10 shows the variation in maximum axial stress with the design variable. Accounting for both transition geometries, it is found that an optimum value of Y, i.e., a value of Y that gives a value of maximum axial stress not exceeding 68 MPa, is in the range 120-170 mm. A good choice of initial value is one that is within this range of values. A desired outcome of the optimisation process is that the total mass of the pipeline be reduced. Looking to Figure 9, which shows the variation in total mass with the design variable, it can be seen the initial design variable value should be greater than the optimum value if mass is to decrease as a result of the optimisation process. Therefore, an initial value for Y of 180 mm is selected. This initial value is used for both types of transition geometries in order to compare their respective optimised solutions.

Figure 11 gives the evolution of the transition length Y for the optimisation process, for both the tangent arc and elliptical shaped transitions. It can be seen that in both cases the transition length starts at a value of 180 mm. The design variable converges to within an acceptable precision in a small number of iterations, in both cases. For the tangent arc transition an optimum value of 163.8 mm is found for the transition length, whereas, for the elliptical transition, a smaller value for the optimum transition length of 118.5 mm is found. Figure 12 gives the corresponding evolution of total pipe and joint mass for the two transition shapes. It is seen that the tangent arc shaped section gives a final mass of 17.1 kg, while the elliptical shaped transition gives a lower final mass of 16.7 kg. This lower mass is evidently due to the lower optimum length of 118.5 mm for the elliptical shaped transition. Finally, Figure 13 shows the evolutions of maximum axial stress for the optimisation process. In both cases it can be seen that the constraint is satisfied, as the stress value reached in both cases at the final iteration is 68 MPa.

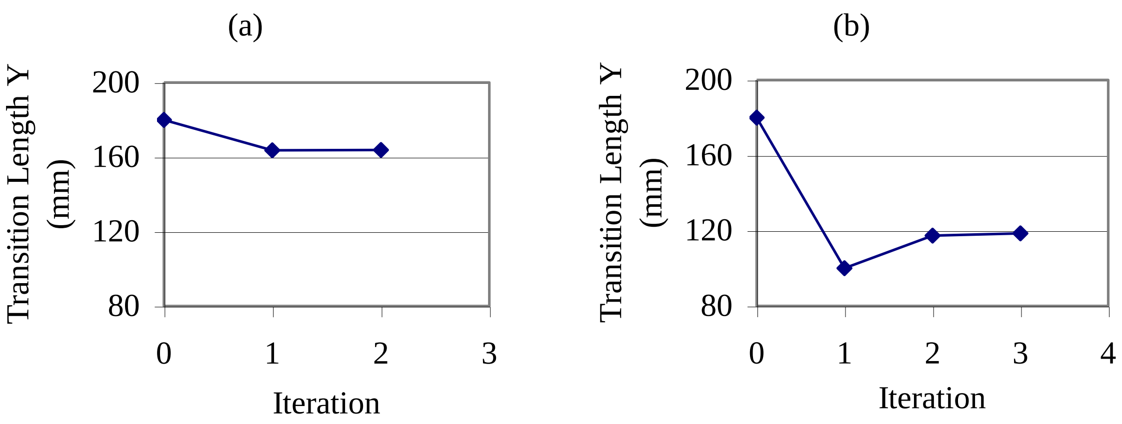


Fig. 11. Evolution of transition length for the shape optimisation process for (a) the tangent arc transition curve and (b) the elliptical transition curve

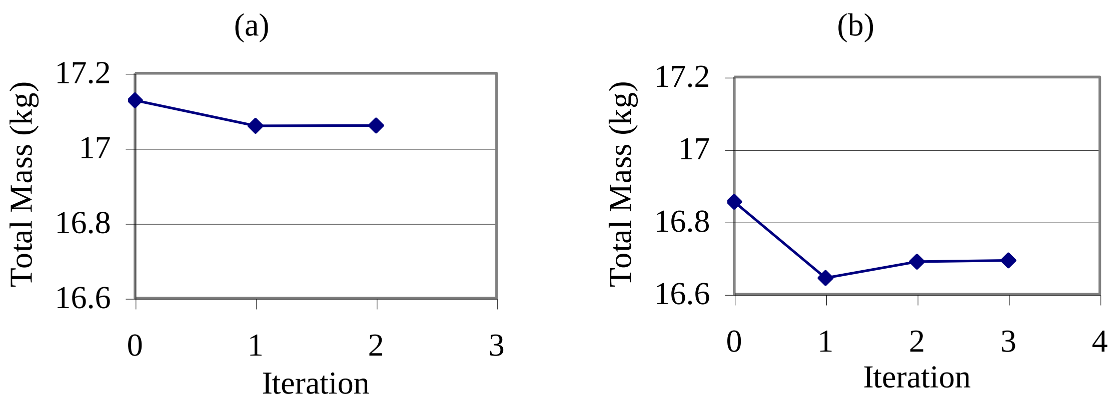


Fig. 12. Evolution of total mass for the shape optimisation process for (a) the tangent arc transition curve and (b) the elliptical transition curve

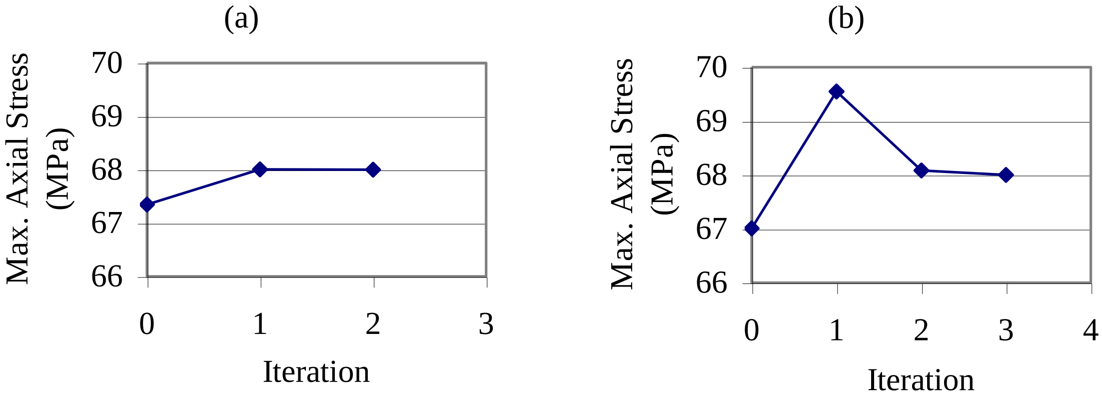


Fig. 13. Evolution of maximum axial stress for the shape optimisation process for (a) the tangent arc transition curve and (b) the elliptical transition curve

Thickness optimisation

Thickness optimisation is applied to a metre-long length of the uniform pipeline section and it is found that as a result there is a negligible reduction in pipeline mass. The reason for this is that the axial stress in the uniform section of the pipe is 63 MPa, which is already very close to the axial stress limit of 65 MPa. Therefore it is concluded the benefit to the pipeline and joint design from the thickness optimisation is very small compared to that given by the shape optimisation.

CONCLUSIONS

Finite element analysis of the pipeline and joint reveal that the highest stresses occur at the transitions from the pipe to the joint. This compares favourably to experimental results from pressure testing of the pipe. Shape optimisation leads to a much greater improvement in the pipeline and joint design than can be achieved with thickness optimisation alone. Optimisation of the transition geometries is shown to significantly reduce the peak stresses. An elliptical shaped transition is shown to be the more successful transition geometry. It gives a smaller transition length, than a tangent arc shaped transition, while still satisfying the design criteria. This results in a lower overall pipeline and joint mass.

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