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LMP-based Creep Rupture Estimate of an Alloy 617 Serpentine Superheater Tube Using Ansys

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Abstract: Superheater tubes in ultra-supercritical power plants operate at high temperatures and pressures for prolonged periods, making creep rupture an important life-limiting mechanism. This study investigates the thermal-structural behaviour and estimates the creep-rupture life of an Alloy 617 serpentine superheater tube using finite element analysis and the Larson–Miller parameter method. The tube (OD 50.8 mm, ID 37.1 mm) carries steam at 700 °C and 300 bar and is heated externally by flue gas at 1500 K. The maximum wall temperature predicted by ANSYS is 733.84 °C. The computed temperatures agree with published straight-tube results within 0.5%. The inner-wall maximum principal stress of 96.08 MPa agrees with the Lamé thick-wall hoop stress of 98.6 MPa within 2.6%. Changing the mesh size from 4.5 mm to 5 mm alters the maximum stress by only 0.05%, although this does not establish mesh convergence. The bend peak von Mises stress reaches 143.47 MPa, approximately 1.3 times the straight-section inner-wall value of 108.65 MPa. The estimated rupture life is approximately 24,000 h without the standard-error correction and 6,600 h with it at the analytical hoop stress, decreasing to about 2,300 h at the bend peak stress. These estimates are below the 100,000 h design-life target discussed in the reference study. The findings highlight the influence of bend-induced stress concentration and stress selection on predicted rupture life. The results provide a preliminary estimate of creep-rupture life, and further validation is required.

Keywords: Alloy 617; superheater tube; Larson-Miller parameter; creep rupture; finite element analysis

I. INTRODUCTION

Ultra-supercritical (USC) and advanced ultra-supercritical (A-USC) power plants aim to improve thermal efficiency by operating at higher steam temperatures and pressures. These conditions increase the demands on high-temperature components, including final-stage superheater coils, which are exposed to elevated temperatures and sustained mechanical stresses [1]. Nickel-based Alloy 617 is a candidate material for such applications because of its high-temperature strength and oxidation resistance [2]. During operation, a superheater tube is subjected to internal steam pressure and external heating from hot flue gas. Prolonged exposure to these conditions can cause creep deformation and eventual creep rupture. Tube-wall thickness is therefore an important design parameter because it influences pressure-induced stress as well as material use [3].

Heat transfer and power plant engineering principles are essential for evaluating the thermal conditions experienced by superheater tubes operating under high-temperature and high-pressure conditions [4, 5]. In supercritical steam systems, the thermophysical properties of water and steam influence heat-transfer behaviour and should be considered when establishing the thermal boundary conditions of the tube [6]. In addition, analytical stress calculations based on the mechanics of thick-walled cylinders provide a theoretical reference for validating finite element predictions of pressure-induced stress in boiler tubes [7]. These principles support the thermal and structural analysis of an Alloy 617 serpentine superheater coil and help assess its response under representative operating conditions.

Alloy 617 contains nickel as its principal constituent, together with chromium, cobalt and molybdenum, which contribute to its high-temperature properties. Its nominal chemical composition and material specifications are documented by the manufacturer [8]. The thermophysical properties of Alloy 617 have been investigated over a broad temperature range [9], while experimental creep and creep-rupture data have been compiled to support high-temperature design assessments [10], [11–14]. These data provide a basis for estimating rupture life using time–temperature parameters, including the Larson–Miller parameter (LMP) [15].

Pavan and Srinivasan [3] investigated the creep-rupture life of an Alloy 617 superheater tube under conditions representative of a final superheater stage, with steam at approximately 700 °C and 300 bar. Their approach included calculating pressure-induced hoop stress, modelling the tube temperature field using ANSYS, and applying an LMP correlation to estimate rupture life. However, a straight-tube model does not represent the complete geometry of a serpentine coil and therefore cannot fully capture the stress distribution associated with its bends and geometric constraints.

Previous studies have examined the creep behaviour and long-term rupture-life extrapolation of Alloy 617 [11–14], the relationship between microstructural evolution and creep performance [16, 17], and the estimation of remaining creep life in superheater tubes [18, 19]. High-temperature oxidation behaviour has also been investigated under different environmental conditions [20–23]. In addition, research on boiler-tube thermal characteristics and radiation heat

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exchange provides relevant background for modelling the thermal boundary conditions of superheater components [24, 25]. Together, these studies establish the importance of evaluating material behaviour, thermal exposure and mechanical loading when assessing high-temperature tube performance.

A serpentine superheater coil differs from a straight tube because its bends introduce curvature and may produce local stress concentrations. Temperature and stress can also vary along the coil, depending on the thermal boundary conditions, geometry and support constraints. Consequently, assessing only a straight section may not identify the location of the maximum stress or provide a complete picture of the coil's mechanical response. A three-dimensional thermal-structural analysis is therefore useful for identifying critical regions and comparing the stress response of straight sections and bends.

Another important consideration is the stress measure used in the LMP-based rupture-life calculation. The underlying creep-rupture correlation is derived from experimental data obtained under specified test conditions, commonly involving uniaxial stress [10, 15]. In contrast, a finite element model provides spatially varying stress components and derived quantities, such as principal and von Mises stresses. These measures are not interchangeable, and selecting a different stress measure can substantially change the calculated rupture life. Comparing analytical hoop stress with numerical stress measures can therefore help demonstrate the sensitivity of the predicted life to the chosen stress input. The resulting estimates should be interpreted in light of the assumptions and limitations of the correlation.

This study develops a three-dimensional thermal-structural finite element model of an Alloy 617 serpentine superheater tube and uses an LMP correlation to estimate creep-rupture life. The objectives are to:

- Determine the steady-state temperature distribution under representative steam-side and flue-gas-side thermal conditions.
- Evaluate the structural response under internal pressure and thermal loading, and compare the straight-section stress with the analytical Lamé thick-wall solution.
- Identify the stress concentration associated with the coil bends.
- Estimate creep-rupture life using the LMP correlation and assess the sensitivity of the estimates to the stress measure used.

The structural analysis is elastic and does not explicitly simulate time-dependent creep deformation or creep-damage evolution. Accordingly, the reported life values are LMP-based predicted rupture lives under the adopted correlation and assumptions; they should not be interpreted as guaranteed service lives.

The remainder of this paper is organised as follows. Section 2 describes the coil geometry, material properties, thermal and structural models, analytical stress calculations and LMP equations. Section 3 presents the temperature and stress distributions, comparison with the Lamé solution, mesh-sensitivity results and predicted rupture-life estimates. Section 4 discusses the limitations and presents the conclusions.

II. METHODOLOGY

A. Geometry

The model consists of a serpentine tube with three straight runs, each 300 mm long, connected by 180° bends with a specified radius of 75 mm and a 40 mm straight section near each bend, as illustrated in Fig. 1. The tube has an outer diameter of 50.8 mm and an inner diameter of 37.1 mm, giving a wall thickness of 6.85 mm. These dimensions are consistent with the outer and inner diameters reported in [3]. Although [3] reports a wall thickness of 6.35 mm in one place, this value is inconsistent with the stated diameters; therefore, the wall thickness used in the present model is calculated as 6.85 mm from the specified outer and inner diameters.

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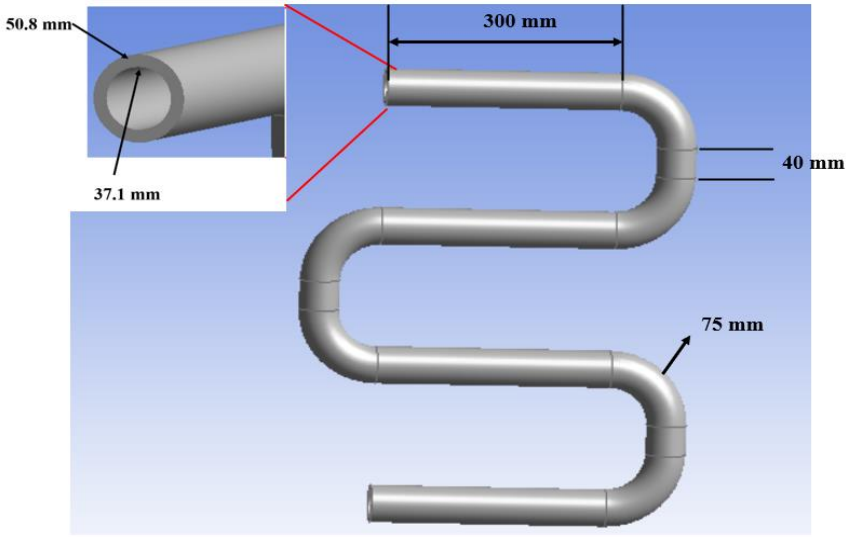


Fig. 1 Geometry of the serpentine tube.

B. Material properties

The material properties of Alloy 617 used in the numerical model are listed in Table 1. The properties were assigned using representative values at 700 °C and were treated as constant over the simulated temperature range. The coefficient of thermal expansion was included to account for thermal strain in the structural analysis.

TABLE 1. ALLOY 617 PROPERTIES USED IN THE MODEL

Property (at 700 °C)	**Value**
Thermal conductivity	26.28 W/m-K
Specific heat	643 J/kg-K
Density	8305 kg/m³
Young's modulus	166 GPa
Poisson's ratio	0.3
Coefficient of thermal expansion (CTE)	11.48×10^{-5} K^{-1}

C. Thermal analysis

A steady-state thermal analysis was performed using ANSYS Mechanical. Steam at 700 °C was assumed to flow inside the tube, with an internal heat transfer coefficient of 4472 W/m²·K. The outer surface was exposed to flue gas at 1500 K, with an external combined convection and radiation heat transfer coefficient of 107.2 W/m²·K [3].

D. Structural analysis

The nodal temperatures obtained from the steady-state thermal analysis were transferred to a static structural analysis. An internal pressure of 300 bar (30 MPa) was applied to the inner tube surface. Atmospheric pressure was assumed on the outer surface, corresponding to zero gauge pressure. The upper-left tube end was restrained using a fixed support (Fig. 4). The structural analysis was performed under a linear-elastic material assumption; therefore, creep strain and stress relaxation were not considered. The mesh was generated using an element size of 4.5 mm for the finer mesh and 5 mm for the coarser mesh. The element formulation and the corresponding node and element counts are reported for each mesh in Table 2. Two mesh sizes, 4.5 mm and 5 mm, were solved for both the thermal and structural cases.

TABLE 2. NODE AND ELEMENT COUNTS FOR DIFFERENT MESHES

Analysis	**Element size (mm)**	**Number of nodes**	**Number of elements**
Thermal	4.5	115058	21603
Thermal	5.0	109428	21000
Structural	4.5	41274	7008
Structural	5.0	38454	6528

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E. Analytical hoop stress

Two analytical references were used to estimate the pressure-induced hoop stress in a straight section of the tube. The first is the thin-wall mean-diameter formula, given below [3]:

$$\sigma_h = \frac{Pd_m}{2t} \quad (1)$$

where (P) is the internal pressure, (d_m) is the mean diameter, and (t) is the wall thickness. For (P=30) MPa, (d_m=43.95) mm, and (t=6.85) mm, the calculated hoop stress is 96.24 MPa.

The second reference is Lamé's thick-wall solution [7], which gives the circumferential stress at the inner wall as

$$\sigma_{\theta,i} = P\frac{r_o^2+r_i^2}{r_o^2-r_i^2} \quad (2)$$

here (ri=18.55) mm and (ro=25.4) mm are the inner and outer radii, respectively. For an internal pressure of 30 MPa and zero external gauge pressure, the inner-wall hoop stress is 98.58 MPa, while the radial stress at the inner wall is −30 MPa. Since the wall thickness is approximately 0.37 times the inner radius, the thick-wall solution provides the more appropriate analytical reference for comparison with the finite-element results. These analytical solutions apply to a straight tube section and do not account for local stress concentrations at the bends or supports.

F. Larson-Miller rupture life

The creep rupture life was estimated using the Larson–Miller parameter (LMP) approach [15] and the relations [3]. The temperature-dependent relation is

$$LMP = T(C + log_{10}t + 1.65\,SEE) \quad (3)$$

The rupture-life constants used in the present study are (a_0=32,976), (a_1=-908), (C=16.73), and (SEE=0.3389). The stress-dependent Larson–Miller relation is expressed as

$$LMP = a_0 + a_1 log_{10}\sigma \quad (4)$$

where σ is the stress in MPa. The rupture time is calculated using

$$log_{10}t = \frac{a_0+a_1\,log_{10}\sigma}{T} - C \quad (5)$$

The lower-life estimate includes the standard-error correction:

$$log_{10}t = \frac{a_0+a_1\,log_{10}\sigma}{T} - C - 1.65\,SEE \quad (6)$$

where T is the absolute temperature in kelvin and t is the rupture time in hours. The uncorrected equation represents the central regression estimate, while the standard-error-adjusted equation provides a lower estimate based on the adopted statistical correction.

The rupture-life curves in Figs. 3 and 4 of [3] are reproduced more closely when the SEE correction is omitted. For example, at a stress of 96.24 MPa, the equation without the SEE term gives log t=5.414 at 960 K and 4.112 at 1020 K. Accordingly, rupture life is evaluated both with and without the SEE correction. The result without SEE represents the central fit, whereas inclusion of the (1.65, SEE) term gives a lower, more conservative life estimate.

III. RESULTS AND DISCUSSION

A. Temperature field and validation

The steady-state thermal analysis predicts a maximum tube temperature of 733.84 °C and a minimum temperature of 714.60 °C, giving a temperature range of 19.24 °C (Fig. 2). Table 3 compares the predicted temperatures with the results reported by [3]. The relative differences in the maximum and minimum temperatures are 0.42% and 0.11%, respectively. This close agreement supports the consistency of the present thermal model with the reference study. The maximum temperature is approximately 3.05 °C higher than the reference value, indicating a small difference between the two model predictions.

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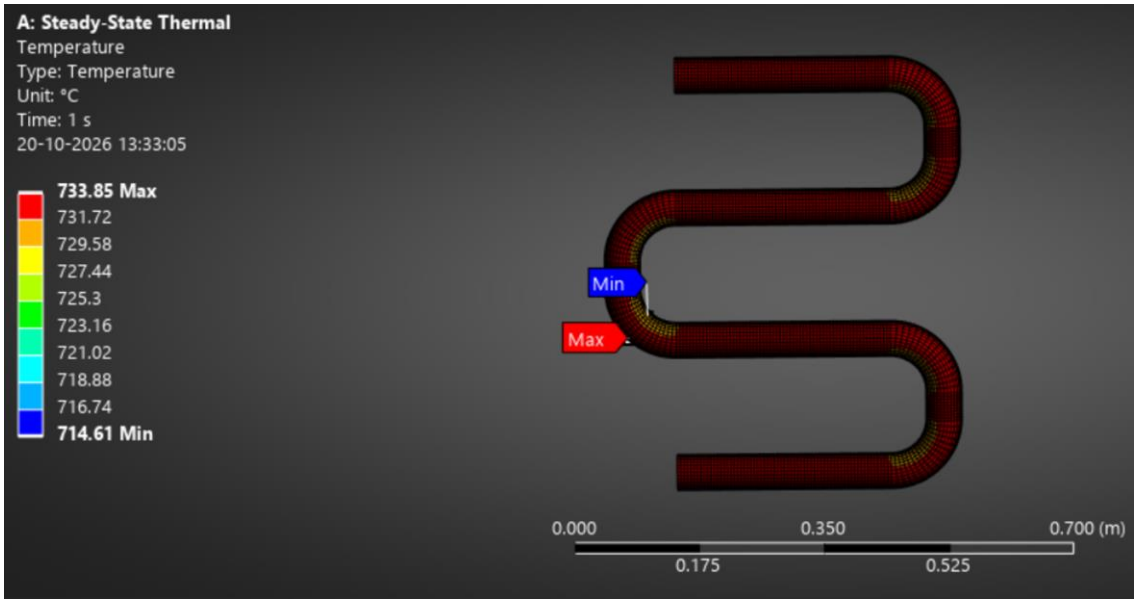


Fig. 2 Temperature distribution (°C), 4.5 mm mesh

TABLE 3. THERMAL VALIDATION AGAINST THE BASE STUDY

Quantity	Present work (°C)	Pavan and Srinivasan [3] (°C)	Difference (%)
Maximum temperature	733.85	730.79	0.42%
Minimum temperature	714.61	715.37	0.11%

B. Mesh sensitivity

Mesh sensitivity was evaluated using element sizes of 4.5 mm and 5 mm for the thermal and structural analyses. As shown in Table 4, the maximum von Mises stress differs by 0.049%, while the maximum total deformation differs by 0.029% between the two meshes. The maximum and minimum temperatures differ by only 0.01 °C. These small differences indicate low sensitivity of the reported results to the two mesh sizes considered. However, this comparison alone does not establish mesh convergence. Further refinement, particularly around the tube bends and other regions of high stress gradients, would provide a stronger assessment of convergence.

TABLE 4. MESH SENSITIVITY OF THE THERMAL-STRUCTURAL SOLUTION

Quantity	4.5 mm mesh	5 mm mesh	Difference (%)
Maximum temperature (°C)	733.85	733.84	0.001%
Minimum temperature (°C)	714.61	714.60	0.001%
Maximum von Mises stress (MPa)	143.40	143.47	0.049%
Maximum total deformation (mm)	0.050928	0.050943	0.029%

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C. Stress field and validation

The von Mises stress is highest at a bend, with a peak of 143.47 MPa, and lowest near the free end, at 21.9 MPa (Fig. 3). The maximum total deformation is 0.0509 mm, so the tube moves very little (Fig. 4), indicating a small overall displacement under the applied loading conditions.

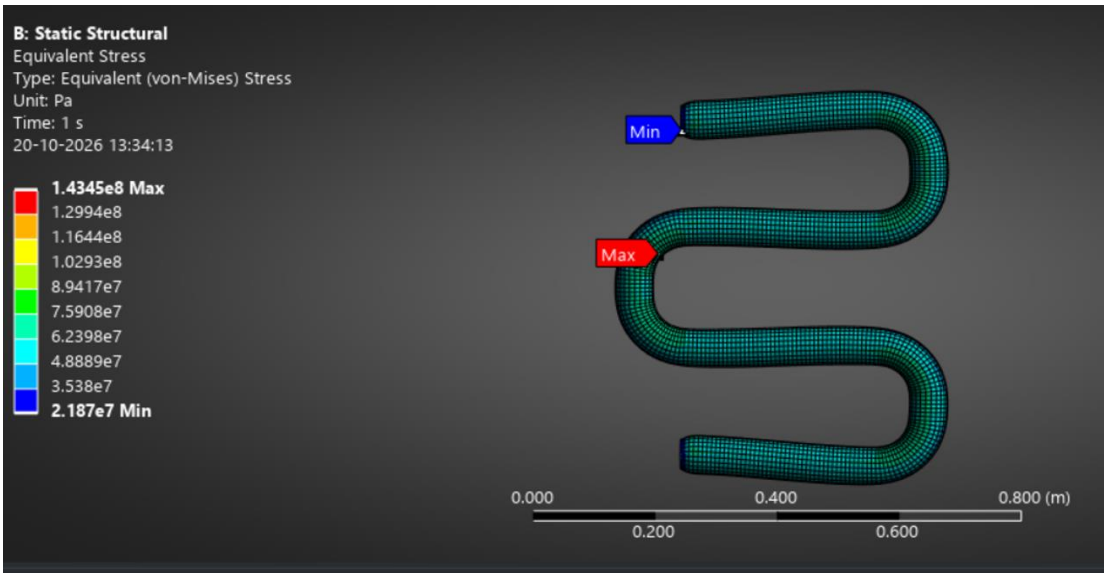


Fig. 3 Equivalent (von Mises) stress (Pa)

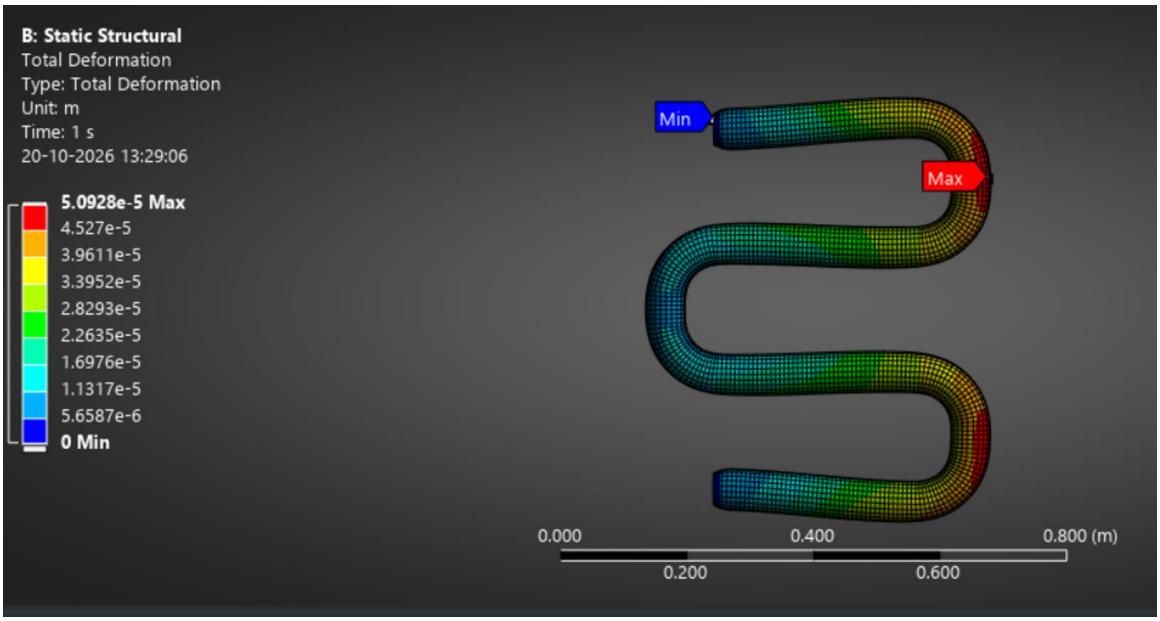


Fig. 4 Total deformation (m)

At the inner wall of a straight section, the maximum principal stress is 96.08 MPa and the von Mises stress is 108.65 MPa. For comparison, the analytical Lamé solution for a straight, thick-walled tube subjected to internal pressure predicts an inner-wall hoop stress of approximately 98.6 MPa. For a closed-end tube under pressure alone, the corresponding inner-wall von Mises stress is approximately 111.4 MPa. The relative differences between the ANSYS results and these analytical reference values are 2.6% and 2.5%, respectively (Table 5). These comparisons provide a useful check against the pressure-only analytical solution; however, the differences may also reflect the thermal loading and the boundary conditions in the finite-element model.

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TABLE 5. STRAIGHT-SECTION STRESS AGAINST THE ANALYTICAL SOLUTION

Quantity	ANSYS (MPa)	Analytical (MPa)	Difference (%)
Inner-wall hoop stress (maximum principal)	96.08	98.6 (Lamé)	2.6%
Inner-wall von Mises stress	108.65	111.4 (Lamé, closed end)	2.5%
Hoop stress, mean-diameter formula	not applicable	96.24	—

The bend peak of 143.47 MPa is about 1.3 times the straight-section inner-wall von Mises stress of 108.65 MPa. The curvature of the bend raises the local stress. The restraint and the thermal gradient may add to it, so the peak should be treated as a local value. Therefore, the bend stress should be interpreted as a local response of the serpentine geometry rather than as the nominal pressure stress in a straight tube.

D. Rupture life

Table 6 gives the rupture life at the maximum temperature of 733.84 °C (1006.99 K) for four stress choices. Since the locations of maximum temperature and maximum von Mises stress do not coincide, combining the maximum temperature with each selected stress provides a conservative temperature–stress assessment for that stress definition.

The rupture-life values are correlation-based estimates rather than validated service-life predictions. The correlation was derived from creep-rupture test data, and its application to finite-element von Mises stresses, particularly at the bend, is approximate. Furthermore, the maximum temperature and maximum bend stress occur at different locations. The corresponding life estimates should therefore be interpreted as stress-sensitivity assessments rather than location-specific rupture predictions.

The primary estimate uses the nominal hoop stress of 96.24 MPa calculated using the thin-wall mean-diameter formula. Based on the central Larson–Miller fit, the estimated rupture life is approximately 24,000 h. Including the (1.65 SEE) correction reduces the estimate to approximately 6,600 h. The predicted rupture lives are below the 100,000 h design-life target discussed in Pavan and Srinivasan [3]. The predicted rupture life is highly sensitive to the selected stress. Using the ANSYS bend-peak von Mises stress of 143.47 MPa reduces the estimated life to approximately 2,3000 h without the SEE correction and 640 h with the correction. These values are sensitivity estimates because the rupture correlation was fitted to uniaxial test data, and equivalent stress at a bend does not necessarily represent the same material response as uniaxial tensile stress. The estimates based on the ANSYS inner-wall von Mises stress are subject to the same limitation.

TABLE 6. LARSON-MILLER RUPTURE LIFE AT 733.84 °C FOR DIFFERENT STRESS DEFINITIONS

Stress basis	Central fit (h)	With SEE (h)
Mean-diameter hoop stress	24,000	6,600
Lamé hoop stress	20,800	5,750
Inner-wall von Mises	11,800	3,250
Bend peak von Mises	2,300	640

The temperature difference across the tube is approximately 19.2 °C. However, the magnitude of the resulting thermal stress cannot be determined from the temperature difference alone because it also depends on the temperature distribution, thermal expansion properties, and structural restraint. A separate pressure-only structural analysis, compared with the combined pressure-and-temperature analysis, is required to assess the contribution of thermal loading to the peak stress. The comparison of the proposed work with previous work is presented in Table 7.

TABLE 7. COMPARISON OF THE PRESENT WORK WITH THE LITERATURE

Ref	Focus	Approach	Relation to the present work
[3]	Alloy 617 final-stage superheater coil at 300 bar and 700 °C steam	ANSYS thermal model of the tube, hoop stress, Larson-Miller life, Robinson's rule	Base study. Reports 730.79 °C maximum and 715.37 °C minimum tube temperature. The present work adds a 3D serpentine geometry, bend stress, a mesh check and the stress sensitivity of life

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Ref	Focus	Approach	Relation to the present work
[10], [11]	Alloy 617 creep and rupture	Creep-rupture data	Source of the correlation constants used here
[12], [14]	Alloy 617 creep life for a very high temperature reactor	Creep behaviour, long-term life extrapolation and reliability of life prediction	Material-level life prediction. The present work applies a correlation to a component model
[18]	Alloy 617 tubes in ultra-supercritical plants	Remnant creep life estimation	Addresses life of tubes already exposed. The present work gives a design-stage predicted life from finite element stresses
[19]	Service-exposed primary superheater tube	Stress analysis and creep behaviour	Combines stress analysis with creep assessment for a superheater tube. The present work treats an Alloy 617 coil at USC conditions
This work	Alloy 617 serpentine superheater tube at 300 bar, 700 °C steam and 1500 K flue gas	3D thermal-structural finite element model, Lamé check, Larson-Miller life	Maximum temperature 733.84 °C. Inner-wall hoop stress 96.08 MPa. Bend peak stress 143.47 MPa. Predicted life about 24,000 h without and 6,600 h with the SEE term at 96.24 MPa

IV. CONCLUSION

A sequentially coupled thermal-structural model of an Alloy 617 serpentine superheater tube was built and checked against the Lamé thick-wall solution and the temperatures reported for a straight tube. The thermal model gives a maximum tube temperature of 733.84 °C and a minimum of 714.60 °C, within about 3 °C and 0.8 °C of the straight-tube values of 730.79 °C and 715.37 °C. The inner-wall hoop stress of 96.08 MPa from ANSYS agrees with the Lamé value of 98.6 MPa within 2.6%, and the maximum stress changes by only 0.05% between the 4.5 mm and 5 mm meshes, which shows low mesh sensitivity but does not establish convergence. These comparisons provide an approximate check of the pressure-induced stress response in the straight section. The bend raises the peak von Mises stress to 143.47 MPa, about 1.3 times the straight-section inner-wall value of 108.65 MPa. At 733.84 °C and a hoop stress of 96.24 MPa, the predicted rupture life under the Larson-Miller correlation is about 24,000 h from the central fit and about 6,600 h with the standard-error term, both below the 100,000 h design target. The life falls to about 2,300 h when the bend peak stress is used. This value is conservative because the peak stress and the maximum temperature do not occur at the same location. The choice of stress measure therefore matters more than the small difference between the analytical and numerical hoop stress. Future work should add creep and damage analysis, a bend life assessment at matched stress and temperature, a finer mesh at the bends, and a comparison with plant or test data.

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