

Environmentally assisted fatigue in LWR – Development of a calculation tool

The effect of LWR coolant environments on the fatigue life of the reactor materials

Master's thesis in Applied Mechanics

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SCIENCE

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Master of Science Thesis in the Master's Program Applied Mechanics

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Cover: The cover picture represents the primary and the secondary system in a pressurized water reactor.

ABSTRACT

The ASME Boiler and Pressure Vessel Code [1] provide rules for the design and construction of class 1 nuclear power plant components. Current design of fatigue curves is based primarily on strain-controlled fatigue tests of small polished samples at room temperature in air but recent data from the USA and Japan demonstrate that the light water reactor (LWR) coolant has significant effects on the fatigue resistance of materials. The Nuclear Regulatory Commission (NRC) has written a report, NUREG/CR-6909 [2] that reviews and quantifies these effects on fatigue life of typical reactor materials and proposes a method for establishing reference curves and environmental fatigue correction factors for evaluating fatigue life.

The purpose of this project is to use the proposed influence in fatigue life, due to the environmental parameters, such as temperature, rate of strain, dissolved-oxygen and amount of sulphide in the LWR coolant environment, according to the NRC report [2] to calculate the environmental fatigue correction factor. If the fatigue correction factor becomes larger than 1, the fatigue usage factor will decrease. The fatigue usage factor is the number of cycles of a certain load divided by the total number of allowable cycles of the same load. In other words, if the fatigue correction factor is larger than 1, for example 2, the number of allowable cycles will decrease to half of the previous number of allowable cycles, i.e., the fatigue life of the specimen will decrease.

The research is performed using the software ANSYS Mechanical Workbench [3] and PIPESTRESS [4]. In ANSYS an axisymmetric 2-dimensional model of a pipe is constructed. The load set is one single thermal transient which will, together with convection at surfaces cause a temperature distribution over time over the whole domain. This temperature distribution in turn will cause strains in the material. The strain rates and the temperature changes are then used to calculate the fatigue correction factor.

The report contains the results of the fatigue corrections factors calculated for a single common thermal transient with different ramp times of the temperature load. The load is applied on a straight pipe with different thicknesses of a common reactor material. The results from ANSYS and PIPESTRESS are then compared.

When studying a temperature decrease of the coolant from 290°C to 20°C it can clearly be seen that the fatigue correction factor increases when the thickness of the pipe increases in both PIPESTRESS and ANSYS but the values are always slightly lower in ANSYS. The result differs between the software when studying the ramping time dependence. In PIPESTRESS the worst scenario appears for the thicker pipes when the temperature is down to 20°C in 10 seconds while 1 second is worse for the two thinner ones. In ANSYS on the other hand the worst case is 1 second ramping time no matter thickness except for 60 mm when 10 seconds is worse.

Key words: ANSYS Mechanical Workbench, ASME, NUREG, PIPESTRESS, LWR, fatigue correction factor, fatigue usage factor

SAMMANFATTNING

The ASME Boiler and Pressure Vessel Code [1] har satt upp regler för konstruktion av klass 1 komponenter i kärnkraftverk där den nuvarande utformningen av utmattningskurvor främst baseras på tøjningskontrollerade utmattningsprov av små polerade prover i rumstemperatur i luft. Men nyligen framtagna utmattningsdata från USA och Japan visar att lättvattenreaktor (LWR) miljöer kan ha stora effekter på utmattningshållfastheten av material. The Nuclear Regulatory Commission (NRC) har skrivit en rapport, NUREG/CR-6909 [2] som sammanfattar och kvantifierar effekten av LWR miljön på utmattningstiden för typiska reaktormaterial. I rapporten föreslås en metod för att fastställa referenskurvor och korrigeringsfaktorer på grund av påverkande miljöfaktorer som används för att evaluera livstiden för reaktorkomponenter som utsätts för LWR miljön.

Syftet med detta projekt är att använda det föreslagna inflytandet på livslängden på grund av olika miljöparametrar så som temperatur, tøjningshastighet, mängden löst syre och mängden sulfid i LWR miljön, enligt NRC [2], för att beräkna utmattningskorrektionsfaktorer. Om utmattningskorrektionsfaktorn blir större än 1 kommer utmattningsanvändningsfaktorn minska. Denna utmattningsanvändningsfaktor utgörs av antalet cykler av en viss belastning dividerat med det totala antalet tillåtna cykler av samma belastning. Med andra ord, om utmattningskorrektionsfaktorn är större än 1, t.ex. 2, kommer antalet tillåtna cykler minska till hälften av den ursprungliga mängden tillåtna cykler, d.v.s. livslängden av provet kommer att minska.

Undersökningen görs med hjälp av programvaran ANSYS Mechanical Workbench [3] och PIPESTRESS [4]. I ANSYS illustreras röret av en axisymmetrisk modell i 2D. Röret belastas av en termisk transient och som, tillsammans med konvektion vid ytorna orsakar en temperaturfördelning över tiden över hela domänen. Denna temperaturfördelning kommer i sin tur att orsaka tøjningar i materialet. De tøjningshastigheter och förändringar av temperaturen används sedan för att beräkna utmattningskorrektionsfaktorn.

Rapporten innehåller resultaten över beräknade korrigeringsfaktorer framtagna för en vanligt förekommande termisk transient med olika ramptider för temperaturlasten på ett rårör av ett vanligt reaktormaterial med olika tjocklekar. Resultaten från ANSYS och PIPESTRESS jämförs därefter.

När man studerar en temperaturminskning av kylvätskan från 290 °C till 20 °C kan man tydligt se att utmattningskorrigeringsfaktorn ökar när tjockleken på röret ökar i både PIPESTRESS och ANSYS men värdena är alltid något lägre i ANSYS. Resultatet skiljer sig mellan mjukvaran när man studerar ramptidsberoendet. I PIPESTRESS uppstår det värsta scenariot för de tjockare rören när temperaturen är nere på 20 °C på 10 sekunder medan 1 sekund är sämre för de två tunnare. I ANSYS å andra sidan är det värsta fallet 1 sekunds ramptid oavsett tjocklek förutom 60 mm då 10 sekunder är sämre.

Nyckelord: ANSYS Mechanical Workbench, ASME, NUREG, PIPESTRESS, LWR, utmattningskorrektionsfaktor, utnyttjandefaktor

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NOTATIONS

F_{en}	Fatigue correction factor
ε_z	Strain in axial direction
ε_{max}	Maximum axial strain
ε_{min}	Minimum axial strain
$\Delta\varepsilon$	Axial strain difference
$\dot{\varepsilon}$	Strain rate in axial direction
T	Temperature
E	Modulus of elasticity
σ	Stress
ΔT	Radial temperature difference
U	Fatigue usage factor
n	Anticipated cycles during life time
N	Allowable cycles
a	Inner radius of pipe
b	Outer radius of pipe
α	Coefficient of thermal expansion
ε_{th}	Threshold value
DO	Dissolved oxygen level
S	Sulphide content in the steel
S^*	Transformed sulphide content for carbon and low-alloy steels
T^*	Transformed temperature for carbon and low-alloy steels
T'	Transformed temperature for austenitic stainless steels
T'	Transformed temperature for Ni-Cr-Fe alloys
O^*	Transformed dissolved oxygen-level for carbon and low-alloy steels
O'	Transformed dissolved oxygen-level for austenitic stainless steels
O'	Transformed dissolved oxygen-level for Ni-Cr-Fe alloys
$\dot{\varepsilon}^*$	Transformed strain rate for carbon and low-alloy steels
$\dot{\varepsilon}'$	Transformed strain rate for austenitic stainless steels
$\dot{\varepsilon}'$	Transformed strain rate for Ni-Cr-Fe alloys
ASME	American Society of Mechanical Engineers
NRC	Nuclear Regularity Commission
LWR	Light Water Reactor
PWR	Pressurized Water Reactor
BWR	Boiling Water Reactor
SCWR	Supercritical Water Reactor

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1 Introduction

In a nuclear power plant, the heat source is a nuclear reactor, i.e., production of electricity via fission. This procedure frees energy in the fuel that heats up a refrigerant, often water, so that steam is generated which drives a steam turbine connected to a generator which produces electricity. The light water reactor is a type of thermal reactor that uses normal water, as opposed to heavy water, as its coolant and neutron moderator and a solid compound of fissile element as its fuel. Thermal reactors are the most common type of nuclear reactor and light water reactors (LWR) are the most common type of thermal reactor. There are three varieties of LWR's; the pressurized water reactor (PWR), the boiling water reactor (BWR) and the supercritical water reactor (SCWR).

The American Society of Mechanical Engineers (ASME) recognizes fatigue as a possible mode of failure in pressure vessel steel and piping materials. Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading, i.e., a repeated loading and unloading. If the loads are above a certain threshold, microscopic cracks will begin to form at the stress concentrators, such as the surface or grain interfaces et cetera, eventually a crack will reach critical size and the structure will suddenly fracture. To prevent fatigue, you make sure that the structural is built to withstand the number of cycles of each load that occurs during a certain lifetime. Cyclic loadings on a reactor pressure boundary component occur because of changes in mechanical and thermal loadings as the system goes from one load set to another.

1.1 Background

Section III, subsection NB of The ASME Boiler and Pressure Vessel Code [1] provides rules for the design and construction of nuclear power plant components and figures that specify the Code design fatigue curves for applicable structural materials. The current design fatigue curves are based primarily on strain-controlled fatigue tests of small polished specimens at room temperature in air.

Section III, subsection NB-3121 of the ASME Code states that the effects of the LWR coolant environment on fatigue resistance of materials were not intended to be addressed in these design curves. Therefore, the effects of the environment on fatigue resistance of materials used in operating PWR and BWR plants, whose primary-coolant pressure boundary components were designed in accordance with the Code, are uncertain. However, recent fatigue-strain-vs.-life (ϵ -N) data obtained in the U.S. and Japan demonstrate that LWR environments can have potentially significant effects on the fatigue resistance of materials. Specimen lives obtained from tests in simulated LWR environments can be much shorter than those obtained from corresponding tests in air. This is due to the reactor chemistry and other parameters such as the temperature of the coolant, the sulphide contents of the steel and the strain rate amplitude caused by mechanical and thermal loadings.

The Nuclear Regulatory Commission (NRC) has written a final report, NUREG/CR-6909 [2]. This report summarizes the work performed at Argonne National Laboratory on the fatigue of piping and pressure vessel steels such as carbon and low-alloy steels, wrought and cast austenitic stainless steel and nickel-chromium-iron alloys, in LWR environments. The existing fatigue strain-cycle data have been evaluated to identify the various material, environmental and loading parameters that influence fatigue crack initiation, and to establish the effects of key parameters on the fatigue life of these steels. Fatigue life models are presented for estimating fatigue life as a function of material, loading and environmental conditions. The fatigue lives are decreased in LWR environments and the

reduction depends on some key parameters. If these critical loadings and environmental conditions exist during reactor operation the environmental effects will be significant and need to be included in the ASME Section III fatigue evaluations. An approach that can be used to incorporate the effects of LWR coolant environments into the ASME Code fatigue evaluations based on the environmental fatigue correction factor is discussed in the report from NRC. The fatigue design curves are modified by the correction factor to account for environmental effects.

1.2 Problem description

The thesis studies how the LWR coolant environments due to a thermal transient affect the fatigue resistance of class 1 components. The proposed influence in fatigue life due to the environmental parameters according to NRC [2] is used to calculate the environmental fatigue correction factor in three different approaches with help of both ANSYS mechanical workbench [3] and PIPESTRESS [4].

1.3 Purpose / aim

The effects of the LWR coolant environment on fatigue resistance of materials are not addressed in the fatigue design curves in the ASME Code. However, the fatigue correction factors are used to evaluate fatigue life in some countries, but not in Sweden [5]. Therefore, the purpose of this project is to do an investigation of how the changes of the fatigue design curves affect the fatigue usage factors.

Research questions

- What thermal transients affect the pipe in worst manner for fatigue?
- How does the thickness of the pipe affect the fatigue correction factor?
- How severely will it be for more critical parts of the pipe, such as a loading stud, that is exposed to the LWR environment?
- Should compressive stresses also be taken into consideration or is it only the tensile stresses that affect the fatigue correction factor?
- What differences are there between the software PIPESTRESS and ANSYS Mechanical Workbench?
- Is it possible to confirm the new fatigue curves in LWR environment according to NRC?
- How can one implement the calculation procedure for a whole loading set instead of only one single transient?

1.4 Scope / limitations

A simple geometry of a straight pipe with different thicknesses is constructed and one type of thermal transient, believed to be particularly sensitive to environmental influence, with different ramping times is considered. Only thermal stresses are taken into account, not mechanical stresses caused by internal pressure and from supports. The fatigue correction factor is calculated for one single thermal load and not for the whole loading set. The material chosen is one common reactor material. The material is also considered to behave elastically but temperature dependent and two calculations are made with elastic-plastic properties in ANSYS Workbench.

1.5 Method

From my supervisor at ÅF [5] I received a code in PIPESTRESS that was modified to fit my cases. As the resultant *.prf file only contains the temperature distribution for the pipe and not the strains that are needed for calculation of the fatigue correction factor, the strains had to be calculated by hand.

The model in ANSYS Mechanical Workbench was built up from scratch. When the engineering data, the geometry and the model were done, the results were inserted into an excel-file that calculated the fatigue correction factor using three different approaches.

1.6 Key results

When studying a temperature decrease of the coolant from 290°C to 20°C it can clearly be seen that the fatigue correction factor increases when the thickness of the pipe increases in both PIPESTRESS and ANSYS but the values are always slightly lower in ANSYS. The result differs between the software when studying the ramping time dependence. In PIPESTRESS the worst scenario appears for the thicker pipes when the temperature is down to 20°C in 10 seconds while 1 second is worse for the two thinner ones. In ANSYS on the other hand the worst case is 1 second ramping time no matter the thickness except for 60 mm when 10 seconds is worse.

2 Theory

This section describes how the fatigue correction factor is calculated after finding the temperature distribution and the strains with help of ANSYS Mechanical Workbench [3] and PIPESTRESS [4]. Important underlying facts about how PIPESTRESS is working and a brief description about ANSYS Workbench and what is done in ANSYS Mechanical Workbench are also presented in this section.

2.1 The fatigue correction factor, F_{en}

The ASME code fatigue evaluation procedures are described in NB-3200 and NB-3600 [1]. Changes in thermal loadings causes stresses in the material. For each stress cycle or load set pair an individual fatigue usage factor is determined by the ratio of the number of cycles anticipated during the lifetime of the component to the allowable cycles, see equation 2.1.

$$U = \frac{n}{N} = \frac{\text{anticipating cycles during life time}}{\text{allowable cycles}} \quad (2.1)$$

The fatigue design curves in ASME define the allowable number of cycles as a function of applied stress amplitude. In figure 1, the fatigue curves for aluminium and steel are shown, with stress amplitude in vertical direction and the number of cycles in horizontal direction. Both axes are in logarithmic scale.

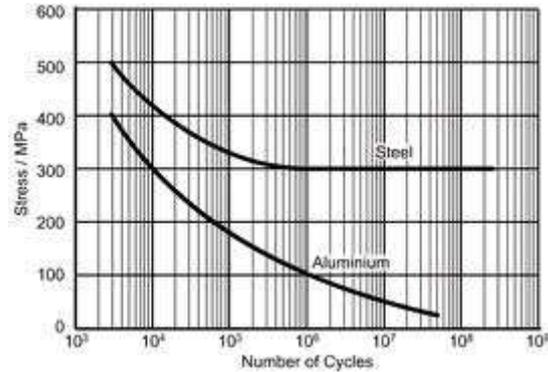


Figure 1: Fatigue curves for aluminium and steel. Stress amplitude and number of cycles in Logarithmic scale

The cumulative usage factor is the sum of the individual usage factors and ASME requires that at each location the cumulative usage factor calculated on the basis of Miner's rule must not exceed 1 [2]. Palmgren Miner's accumulation rule is defined as

$$\sum U_i = \sum \frac{n_i}{N_i} \quad (2.2)$$

When the LWR environmental parameters are taken into account it has been shown that the fatigue usage factor increases by multiplying this with the fatigue correction factor. In the report from NRC, there is presented a method on how to calculate the cumulative fatigue correction factor and it is defined as below [2]

$$F_{en} = \sum_{k=1}^n F_{en,k}(\dot{\epsilon}_k, T_k) \frac{\Delta \epsilon_k}{\epsilon_{max} - (\epsilon_{min} - \epsilon_{th})} \quad (2.3)$$

However, in this thesis, three different approaches based on this equation will be used, see section 2.1.1.

It can be seen in equation 2.3 that the environmental parameters that affect the fatigue resistance are the strain rate and the temperature. Also, the amount of dissolved oxygen in the LWR coolant and the sulphide content of the steel are present in this equation. In LWR coolant environments the procedure for calculating F_{en} includes a threshold strain range, ε_{th} , below which environment has no effect on fatigue life, i.e., $F_{en} = 1$ [2].

The environmental fatigue correction factor for carbon steels is given by

$$F_{en} = e^{(0.623 - 0.101 * S * T * \varepsilon^*)} \quad (2.4)$$

And for low-alloy steels as

$$F_{en} = e^{(0.702 - 0.101 * S * T * \varepsilon^*)} \quad (2.5)$$

For austenitic stainless steels the correction factor is

$$F_{en} = e^{(0.734 - T' \varepsilon' O')} \quad (2.6)$$

For Ni-Cr-Fe alloys it is calculated as

$$F_{en} = e^{(O' T' \varepsilon')} \quad (2.7)$$

For calculation of the parameters in these equations, see Appendix A.

As mentioned before, these equations only consider the sulphide content in the steel, the dissolved oxygen in the coolant, the temperature and the strain rate, to be the main affective parameters in the LWR environments. Other parameters that may be affecting the fatigue resistance are the water conductivity, the flow rate, the surface finish, the heat-to-heat variability, the tensile hold period or different material heat treatments. Normally, plants are unlikely to accumulate many fatigue cycles under off-normal conditions. Thus, the effects of water conductivity on fatigue life have not been considered in the determination of F_{en} . The beneficial effect of the flow rate is presently not included in the fatigue evaluation because of the uncertainties in the flow conditions at or near locations of crack initiation. The effect of surface finish is included in the sub factor for “surface finish and environment” and the effect of the heat-to-heat variability is included in the “data scatter and material variability” that is applied to the mean data curve to develop the Code fatigue design curve in air. Existing data do not demonstrate that hold periods at peak tensile strain affect the fatigue life of carbon, low-alloy steels or austenitic stainless steels and material heat treatments are not considered in F_{en} for austenitic stainless steels [2].

2.1.1 Three different approaches

Three different approaches based on equation 2.3 are used to calculate the fatigue correction factors. For all strains below the threshold value the individual fatigue correction factor becomes 1. Otherwise, it is calculated as in equations 2.4-2.7 depending on the material of the pipe.

The first approach is to calculate the total fatigue correction factor as

$$F_{en} = \sum_{k=1}^n F_{en,k} * \frac{\Delta \varepsilon_k}{\varepsilon_{max}} \quad (2.6)$$

Another approach is to calculate F_{en} as the maximum value of ΔF_{en} , i.e.

$$F_{en} = \max(F_{en,k}) \quad (2.7)$$

The third approach is similar to the first one except that ΔF_{en} that becomes 1 is not taken into account in the total correction factor which is calculated as

$$F_{en} = \sum_k^n F_{en,k} * \frac{\Delta \varepsilon_k}{\varepsilon_{max} - \varepsilon_{th}}, \forall F_{en}(\dot{\varepsilon}_k, T_k) > 1 \quad (2.8)$$

In all these approaches only the strains up to the maximum strain value are included when calculating the individual fatigue correction factors.

The parameters, needed to be able to calculate these factors are the strains and the temperature distribution over time. These are obtained by using two different software, PIPESTRESS and ANSYS Mechanical Workbench.

2.2 PIPESTRESS and Editpipe

PIPESTRESS performs all calculations required by the nuclear piping code in a single program execution. The codes that it uses are ASME Classes 1, 2 and 3 and ANSI/ASME B31.1 and ANSI/ASME B31. Two of the main features are the heat transfer and thermal gradient stress and the fatigue analysis usage factor. Editpipe is a high performance pre-and postprocessor for PIPESTRESS.

PIPESTRESS uses the Euler Bernoulli beam theory in its calculations, but when it comes to the thermal transient analysis, the program calculates the temperature distribution analytically according to [6]. To get the total resulting stresses PIPESTRESS add together the moments and forces from other types of loads (such as the dead weight) from the beam element analysis with the thermal loads from the analytical analysis. It is possible to get both the temperature distribution and corresponding fatigue correction factors in one execution but these are based on one single maximum strain rate. Therefore, in this project, only the given temperatures are used to calculate the strains by hand as the thermal expansion coefficient times the radial temperature difference.

2.2.1 Beam elements

The structural members of beam structures are mathematical “beams”, whose properties are defined in the following section.

According to the Euler Bernoulli theory plane cross sections remain plane and a beam is represented geometrically in 3-dimensional space as a finite segment of a smooth curve which does not intersect

itself. The two end-points of the curve segment are the “nodes” or “points” of the beam. The beam has six degrees of freedom at each point, three for rotation and three for translation. External forces act only at nodes. In particular, the mass of a beam is constructed at its nodes. The beam structure behaves linearly. With each point of the beam is associated a 6×6 flexibility density matrix [6].

2.2.2 Thermal gradients

A thermal gradient is a non-constant temperature distribution across the pipe wall. Thermal gradients give rise to thermal gradient stresses. For a given temperature distribution the thermal gradient stresses are calculated according to the expressions in the ASME Code. Therefore, the problem of calculating thermal gradient stresses reduces to the problem of finding the temperature distribution in the pipe wall [6].

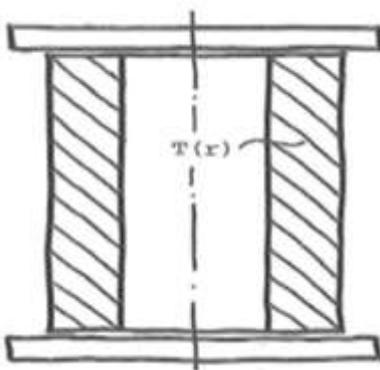
In this case PIPESTRESS solves an axially symmetric heat transfer equation with associated boundary conditions and a heat transfer coefficient between fluid and pipe and between pipe and external environment. The input data that needed to calculate the temperature distribution are the thermal diffusivity of the pipe, the in-and outside heat transfer coefficients and the conductivity.

The rate at which heat crosses the pipe boundaries depends upon the inside- and outside heat transfer coefficients. The outside heat transfer coefficient is chosen to be zero as it is isolated and there is no heat transfer on the outside of the pipe. The inner heat transfer coefficient is calculated based on Reynold’s number, the Prandtl number, the velocity of the fluid, the fluid conductivity, the fluid flow, the dynamic viscosity, inside diameter of the pipe and the specific volume.

2.2.3 Calculation of strains

As the results from PIPESTRESS only give the temperature distribution over time and PIPESTRESS uses a maximum value of the modified strain rate to calculate the fatigue correction factors, the axial strains over time need to be calculated by hand to achieve a fatigue correction factor according to the previously mentioned report established by NRC. Two methods of calculating the axial strain were considered.

2.2.3.1 Method 1



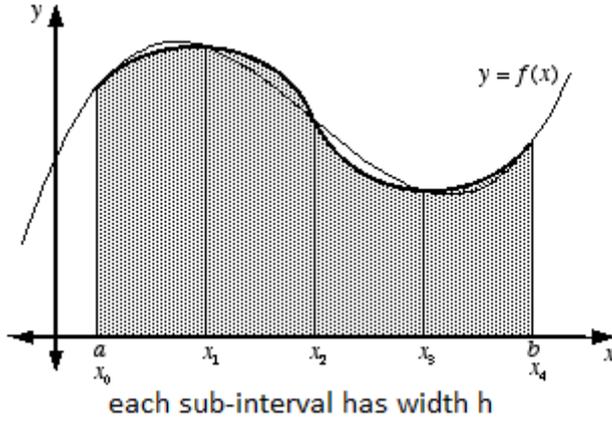
The following equation is used to determine the strains in the axial direction

$$\varepsilon_z = \frac{2\alpha}{b^2 - a^2} \int_a^b T r' dr'$$

Where a = inner radius, b = outer radius [7] and $T(r)$ = temperature through the pipe wall

Figure 2: Tube between rigid, smooth ends without axial compressive forces [7]

The temperature is given at five different points through the wall of the pipe, at the inner radius, at the outer radius and at three points in-between and the temperature gradient is assumed piecewise linear in-between these given temperatures.



Further on, the integral is approximated by Simpson's rule according to [8]

$$\int_{x_0}^{x_{2n}} f(x) dx \approx \frac{h}{3} \left(f(x_0) + 4(f(x_1) + f(x_3) + \dots + f(x_{2n-1})) + 2(f(x_2) + f(x_4) + \dots + f(x_{2n-2})) + f(x_{2n}) \right)$$

$$\text{With } h = \frac{x_{2n} - x_0}{2n}.$$

Figure 3: The shaded area bounded by the parabolas (the thicker curves) is approximately equal to the area bounded by $y=f(x)$

Simpson's rule gives

$$\begin{aligned} \int_a^b T(r)rdr &= \frac{b-a}{3x4} \left[T(a) * a + 4T\left(a + \frac{b-a}{4}\right) * \left(a + \frac{b-a}{4}\right) + 2T\left(a + 2\frac{b-a}{4}\right) \right. \\ &\quad * \left(a + 2\frac{b-a}{4}\right) + 4T\left(a + 3\frac{b-a}{4}\right) * \left(a + 3\frac{b-a}{4}\right) + T\left(a + 4\frac{b-a}{4}\right) \\ &\quad \left. * \left(a + 4\frac{b-a}{4}\right) \right] \\ &= \frac{b-a}{3x4} \left[T_1 * a + 4T_2 * \left(a + \frac{b-a}{4}\right) + 2T_3 * \left(a + 2\frac{b-a}{4}\right) + 4T_4 \right. \\ &\quad \left. * \left(a + 3\frac{b-a}{4}\right) + T_5 * \left(a + 4\frac{b-a}{4}\right) \right] \end{aligned}$$

2.2.3.2 Method 2

As the temperatures are given at five different points through the wall of the pipe, at the inner radius, at the outer and at three points in-between, one more coarse method than the previously mentioned method is to calculate the axial strains at each time point as

$$\varepsilon_z = \alpha * (T_2 - T_1)$$

Where T_1 and T_2 are the temperatures given at the two inner most points of the pipe wall.

Method 1 was used to calculate the strains.

2.3 ANSYS Mechanical Workbench

The ANSYS Workbench environment is an intuitive up-front finite element analysis tool that is used in conjunction with CAD systems and/or DesignModeler. ANSYS Workbench is a software environment for performing structural, thermal, and electromagnetic analyses [3].

In this project the ANSYS Mechanical license is used to calculate the temperature distribution and the strains caused by a thermal transient on a pipe. In the Toolbox a “Transient Thermal” is chosen and then connected with a “Static structural”. The first one is generating the temperatures over time and the second the strains over time. After defining the engineering data for the chosen material, the geometry is constructed with DesignModeler. Then the model is set up in Mechanical where the mesh is created and the loads and other conditions for the pipe wall are defined.

The same mesh is used for both the Static structural analysis and the Transient thermal analysis and the method is quadrilaterals, i.e., four node elements with four Gauss points.

3 Methodology

A simple straight pipe with length of 2 m, outer diameter 817.2 mm and different thicknesses of 60, 30, 15 and 7.5 mm is studied, see figure 4. The material chosen is a common reactor material, SA-376 TP304 (18Cr - 8Ni), which is an austenitic stainless steel. The transient that the tube is exposed to is when the temperature of the LWR coolant goes down from 290°C to 20°C in 1, 10 and 100 seconds. As PIPESTRESS only assumes the material to behave elastically, no yield strengths are needed in the ANSYS model, for best comparison with the results based on the PIPESTRESS analyses.

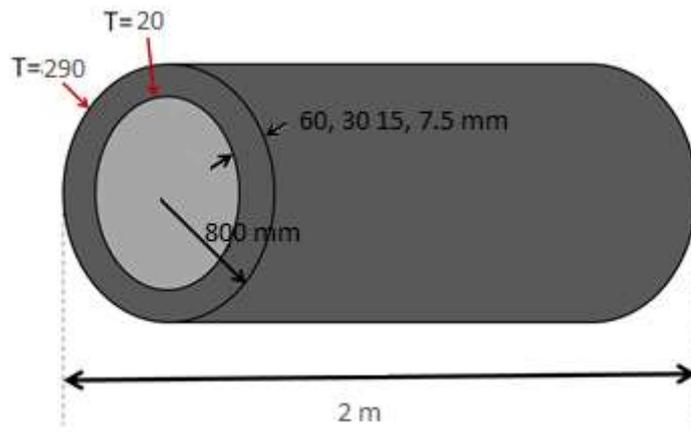


Figure 4: The pipe that is studied

3.1 PIPESTRESS

In Appendix B, the primal code of the PIPESTRESS analysis, is presented. This code was received from my supervisor [5]. Then I modified it to fit the cases that I was studying.

The software is using different cards with characterizing parameters to set up the model. LCAS and TCAS creates the load cases and the transients. It is possible to define different load cases and transients and receive the results for each of these individually or combined.

The LSET card defines the fatigue analysis. It can here be seen that the two last LSET-cards define two different transients that the pipe is exposed to. One is when the coolant goes down in temperature and is marked with a minus-sign. The other one is the opposite case and marked with a plus-sign.

With MATH the material of the pipe is chosen. The material properties are predefined in PIPESTRESS and do not need to be defined. The cross-sectional properties are set up with CROS with OD=outer diameter and WT=thickness. The program provides the mass (MA) based upon the material and the pipe dimensions.

TRAN describes the transients. IT=initial temperature, FT=final temperature and TT=ramp time, i.e., the time it takes for the coolant to reach final temperature. After all this is done the pipe is built up with straight parts et cetera and anchors, which are described as points of maximum resistance to movement, are set as boundary conditions. The displacement in axial direction is put to zero at one edge of the pipe and free in radial direction.

3.2 ANSYS Mechanical Workbench

In ANSYS Workbench a “Transient thermal” and a “Static structural” are chosen from the toolbox and connected together as shown in figure 3

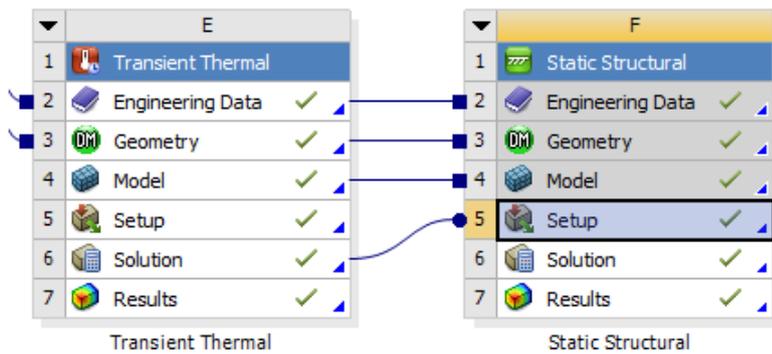


Figure 5: The set up in ANSYS Workbench

In E: Transient thermal, the temperature change is calculated and used to determine the strains in F: Static structural. At first a Transient structural was chosen but as no dynamic effects are affecting the behaviour, it is more convenient to run the static structural analysis over multiple time steps instead.

3.2.1 Engineering data

The first step is then to describe the material behaviour in Engineering Data. In the Toolbox density, isotropic secant coefficient of thermal expansion, isotropic elasticity, isotropic thermal conductivity and specific heat is chosen to define the material of the pipe. All the material properties are taken from the material database [9]. As the coefficient of thermal expansion, Young’s modulus, thermal conductivity and the specific heat are temperature dependent, all these parameters must be defined in tables, see table 1.

Table 1: Material data for SA-376 TP304 (18Cr - 8Ni)

Temperature, [°C]	Young’s modulus, E [Pa]	Coefficient of thermal expansion, α [K^{-1}]	Conductivity, $\left[\frac{W}{m K}\right]$	Specific heat, $\left[\frac{J}{g K}\right]$
21	$195e^9$	$1.53e^{-5}$	14.9	0.483
93	$190e^9$	$1.60e^{-5}$	16.1	0.506
149	$186e^9$	$1.66e^{-5}$	16.9	0.520
204	$183e^9$	$1.71e^{-5}$	18.0	0.535
260	$178e^9$	$1.76e^{-5}$	18.9	0.544
314	$174e^9$	$1.76e^{-5}$	19.5	0.551

The reference temperature is 20°C, Poisson's ratio, $\nu = 0.3$ and the density is $7.9 \frac{g}{cm^3}$.

3.2.2 DesignModeler – Creating the pipe wall

Before creating the geometry of the pipe, the analysis type is changed to 2D in advanced geometry options. Then a rectangle is created in the XY-plane by doing the following steps:

- Click on the XYPlane and then “Look at face/pane/sketch”
- Go to Sketching-Draw-Rectangle and draw a rectangle with one edge on the x-axis.
- Define the dimensions in Sketching-Dimensions
- Click on Concept and then “Surfaces on sketches”. Highlight the four edges of the rectangle and click apply on “Base objects” in the details view.
- Generate

The geometry is shown in figure 6.

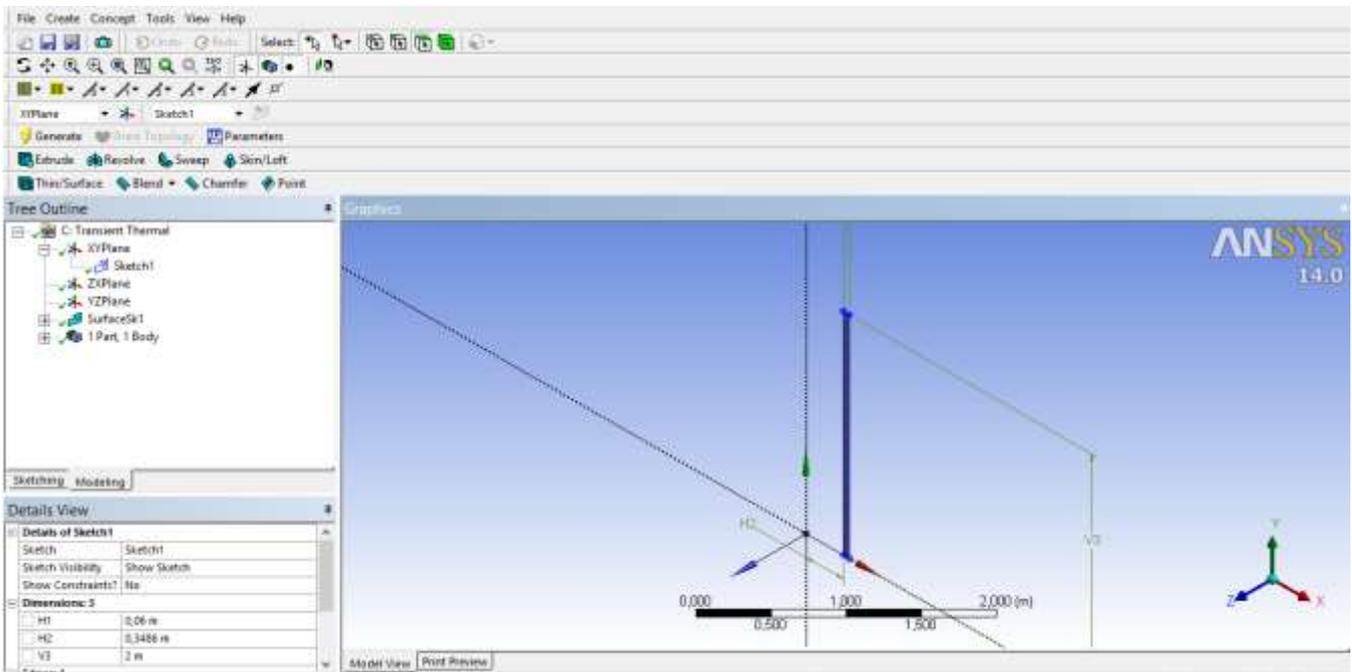


Figure 6: The 2D axisymmetric rectangle drawn in DesignModeler forming the pipe wall

3.2.3 Mechanical – Setting up the model

Before anything else is done Geometry is highlighted in the outline tree. Then the 2D behaviour is changed to axisymmetric in Details of “geometry”.

To create the FE mesh, “Edge Sizing” is inserted. The four edges are highlighted and applied in the Details window. The element size is set to $3 \times 10^{-3} [m]$ and the behavior to hard which means that the program can't interpret the mesh size freely. The reason why this element size is chosen is because it was the same size that had been used in a previous analysis based on a T-shaped pipe made in ANSYS Classic [5] and that tests with varying element sizes were made in the thermal transient analysis until

convergence was reached. A refinement of one corner on the inside tube wall is added and a Mapped Face Meshing is inserted. Then the mesh is generated.

In the Transient thermal analysis initial temperature is set to 290°C. In all the analyses convection is taken into account by inserting a heat transfer coefficient chosen as constant $45000 \left[\frac{W}{m^2 \cdot ^\circ C} \right]$ i.e., independent of the temperature. In an ordinary case, the heat transfer coefficient is dependent of the temperature. However, in this study the temperature in the pipe will decrease almost exactly in the same rate as the fluid. Hence, a constant, very high heat transfer coefficient is a good approximation. This is due to that no internal pressure in the pipe, caused by the flow are taken into account.

In the analysis settings, the number of time steps must be defined and this is done by inserting a list, made in Excel, containing the time steps wanted for the analysis. It is important to make the time steps small before and around the quench time due to the fact that it is around the quench time the maximum strain is achieved and only the tensile strains are taken into account when calculating the fatigue correction factor. The format of the resulting output is chosen as a temperature distribution in one corner of the inner pipe wall.

In the Static structural tree, the environmental temperature is chosen to 290°C. At the end of the pipe where the strains are wanted the axial displacements are put to zero and the pipe is free to move in radial direction. The time steps in this analysis are the same as the ones used in the previous analysis. In order to calculate the resulting strains in every desired time step, the times as where the temperatures shall be collected from the previous result need to be specified in the option "Imported load".

The format of the resulting output is chosen as the change in strain in one corner of the inner pipe wall.

3.2.4 An Elastic-plastic model

In ANSYS, it is possible to do an elastic-plastic analysis by defining the yield strengths and the tangent modulus in the tabular for bilinear isotropic hardening. However, the tangent modulus is unknown and needs to be determined. This is done by following ASME VIII procedure [1] for calculating the exact stress-strain curve and then using the result to approximate the tangent modulus as the inclination of the plastic region.

4 Results

In this chapter the result of the methods described in the methodology chapter is presented.

4.1 Temperature-and strain curves

In figures 7-12 the axial strains in one corner of the inner pipe wall in a cross section and the temperatures at the inner radius achieved from ANSYS and PIPESTRESS for all the different cases are presented.

The results from ANSYS are presented by the solid lines and the temperatures from PIPESTRESS and the calculated strains based on these are presented by the dashed ones. All the results are from the linear elastic analysis.

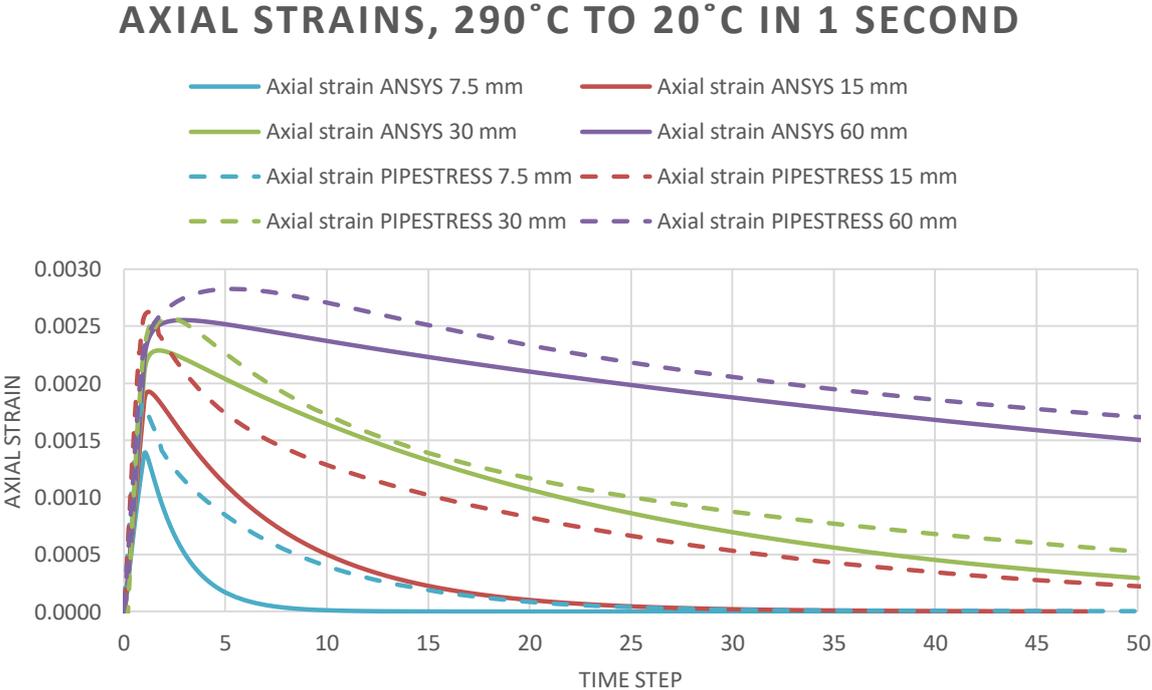


Figure 7: The resulting axial strains in a cross section when the temperature drops from 290°C to 20°C in 1 second in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

TEMPERATURES, 290°C TO 20°C IN 1 SECOND

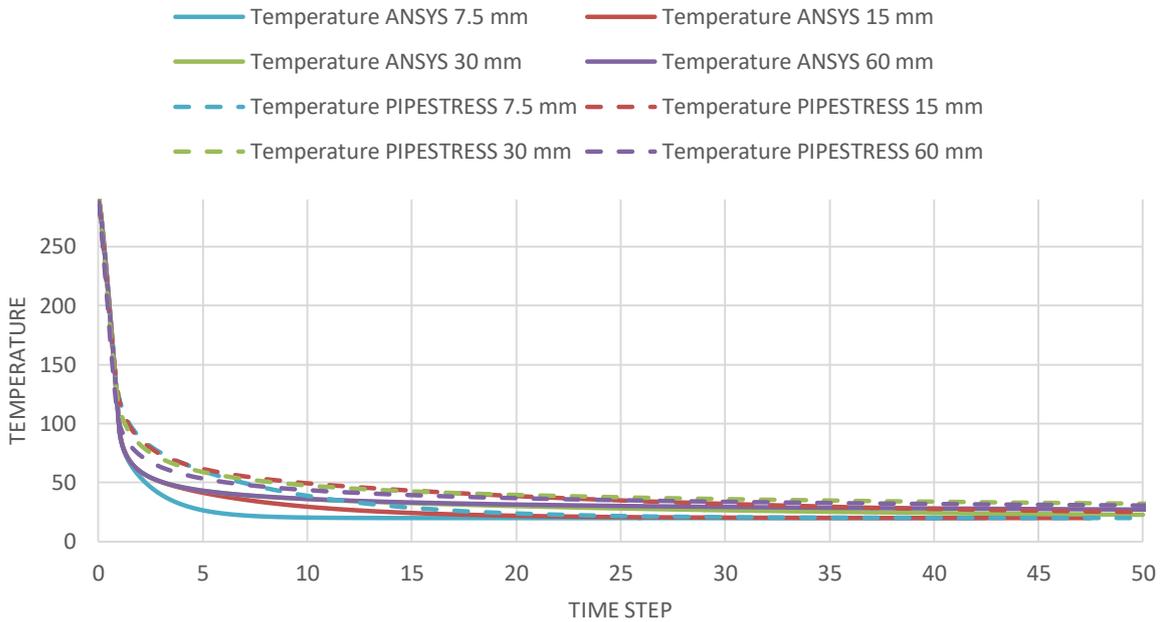


Figure 8: The resulting temperatures at the inner radius when the temperature drops from 290°C to 20°C in 1 second in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

AXIAL STRAINS, 290°C TO 20°C IN 10 SECONDS

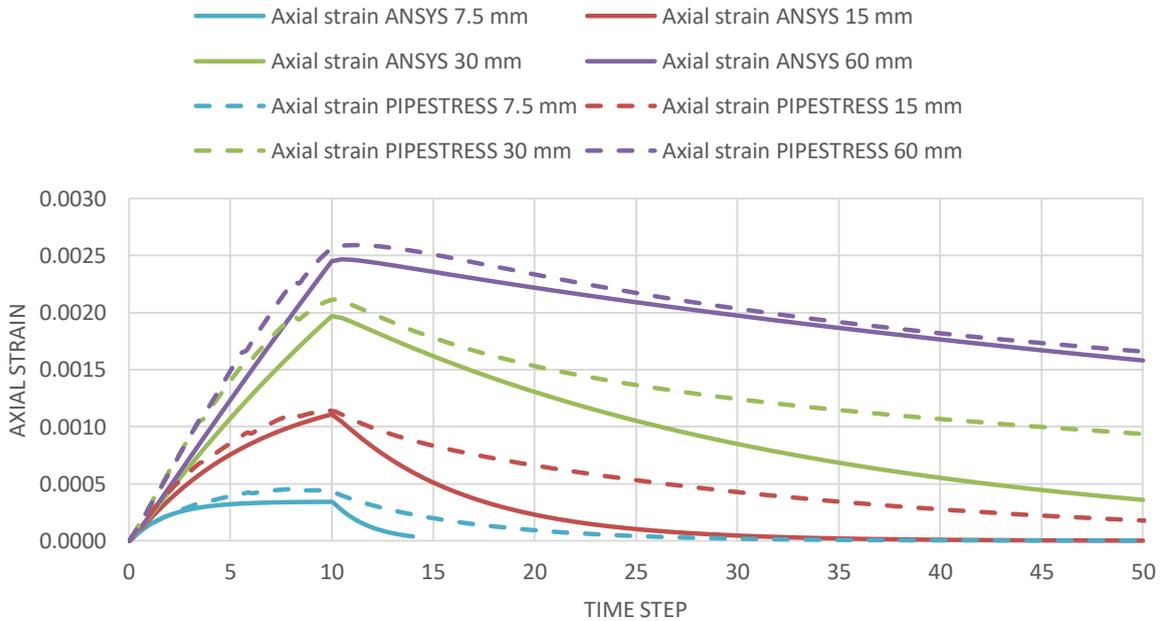


Figure 9: The resulting axial strains in a cross section when the temperature drops from 290°C to 20°C in 10 seconds in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

TEMPERATURES, 290°C TO 20°C IN 10 SECONDS

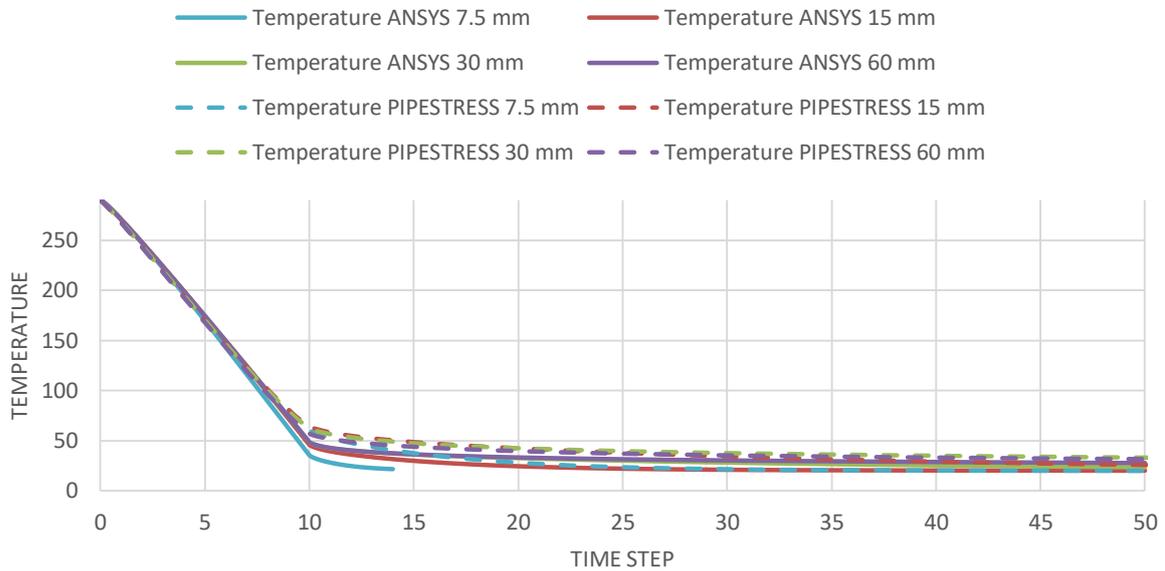


Figure 10: The resulting temperatures at the inner radius when the temperature drops from 290°C to 20°C in 10 seconds in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

AXIAL STRAINS, 290°C TO 20°C IN 100 SECONDS

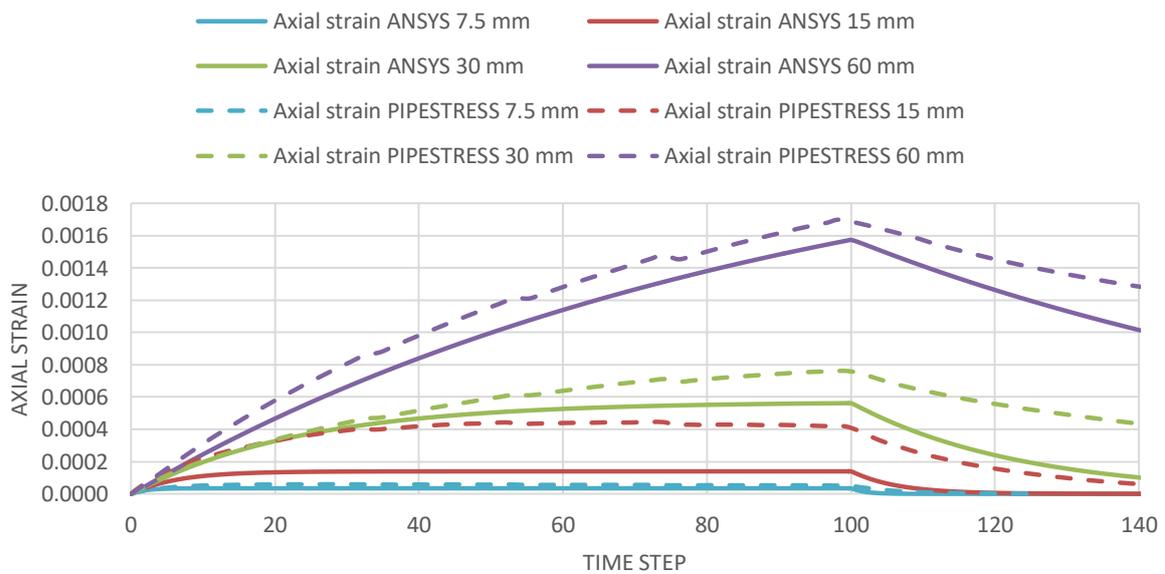


Figure 11: The resulting axial strains in a cross section when the temperature drops from 290°C to 20°C in 100 seconds in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

TEMPERATURES, 290°C TO 20°C IN 100 SECONDS

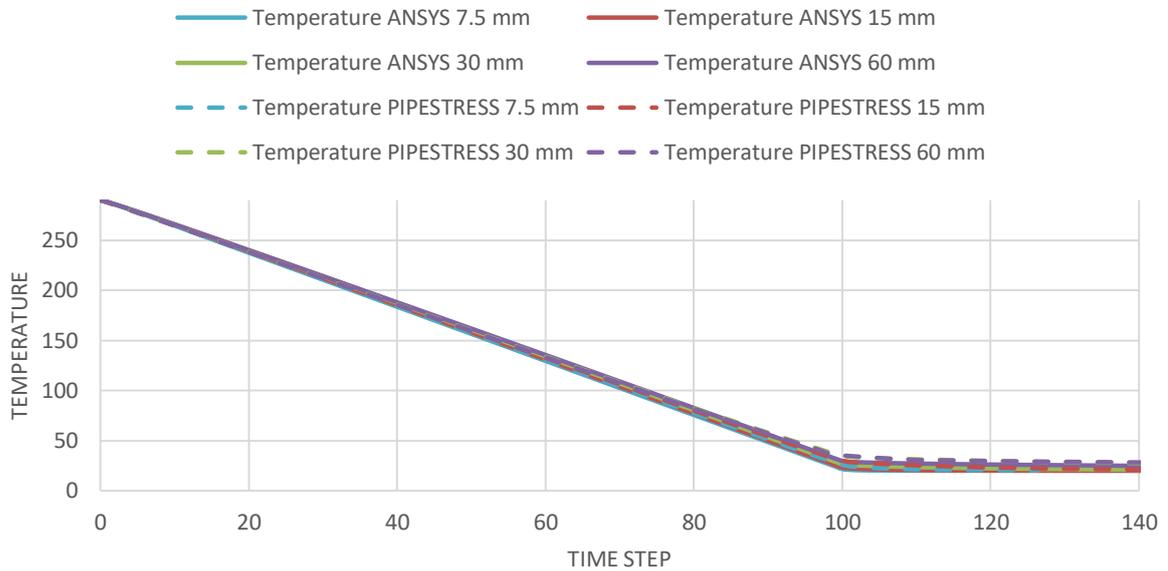


Figure 12: The resulting temperatures at the inner radius when the temperature drops from 290°C to 20°C in 100 seconds in a pipe with four different wall thicknesses of the pipe (ANSYS=solid lines, PIPESTRESS=dashed lines).

4.2 The fatigue correction factor

In Table 2 and 3 the fatigue correction factors are tabulated. Table 2 represent the factors that have been calculated with the results from PIPESTRESS and Table 3, the factors with results from ANSYS Mechanical workbench.

Table 2: The calculated fatigue correction factors in three different approaches using the results from PIPESTRESS

	Thickness	60 mm	30 mm	15 mm	7.5 mm	Approach 1-3
Ramp time						
		1.83	1.76	1.72	1.53	$F_{en,1}$
1 second		2.18	2.19	2.24	2.34	$F_{en,2}$
		2.09	2.08	2.11	2.16	$F_{en,3}$
		1.95	1.85	1.55	1.00	$F_{en,1}$
10 seconds		2.79	2.79	2.95	1.00	$F_{en,2}$
		2.25	2.31	2.29	-	$F_{en,3}$
		1.90	1.25	1.00	1.00	$F_{en,1}$
100 second		3.64	2.22	1.00	1.00	$F_{en,2}$
		2.49	2.10	-	-	$F_{en,3}$

Table 3: The calculated fatigue correction factors in three different approaches using the results from ANSYS

	Thickness	60 mm	30 mm	15 mm	7.5 mm	Approach 1-3
Ramp time						
		1.68	1.63	1.54	1.29	$F_{en,1}$
1 second		2.18	2.18	2.18	2.12	$F_{en,2}$
		2.10	2.10	2.10	2.09	$F_{en,3}$
		1.69	1.56	1.11	1.00	$F_{en,1}$

10 seconds		2.56	2.44	2.08	1.00	$F_{en,2}$
		2.16	2.13	2.08	-	$F_{en,3}$
		1.40	1.00	1.00	1.00	$F_{en,1}$
100 second		2.31	1.00	1.00	1.00	$F_{en,2}$
		2.10	-	-	-	$F_{en,3}$

4.3 Elastic-plastic model

Table 4: Comparison of fatigue correction factors for an elastic and an elastic-plastic model

	Thickness	60 mm Elastic- plastic	60 mm	30 mm Elastic- plastic	30 mm	Approach 1-3
Ramp time						
		1.67	1.68	1.31	1.63	$F_{en,1}$
1 second		2.18	2.18	2.13	2.18	$F_{en,2}$
		2.10	2.10	2.16	2.10	$F_{en,3}$

5 Discussion

The result differs between the software when studying the ramping time dependence. This is due to a quite rough estimation on the axial strains based on a temperature gradient modelled piecewise linear with four segments, given with PIPESTRESS, through the pipe wall at each time point. In PIPESTRESS the worst scenario appears for the thicker pipes when the temperature is down to 20°C in 10 seconds while 1 second is worse for the two thinner ones. In ANSYS on the other hand the worst case is 1 second ramping time no matter thickness except for 60 mm when 10 seconds is worse. Therefore, a short quench time seems to give a higher correction factor.

5.1 Confirmatory data and model validation

One may be concerned as the fatigue factor approaches 15, which is a theoretic maximum [5]. In reality the factors are lower, often around 2. As seen in table 2 and 3, the fatigue correction factor lies between 1-3 which is in the same range as for the already calculated correction factors. Since the obtained results correspond to the already existing correction factors one may say that the results are reliable.

A report at Ringhals presents a graph which includes results from this work, see figure 13 [5]. It illustrates the temperature and the strain which results from an analysis made for a pipe with thickness 63 mm and an outer diameter of 917 mm. The material is steel 2333 (also stainless) with simplified material data when compared to this research. The temperature of the fluid decreases from 290°C to 20°C in 1 second. In the figure below the denoted project are from the current work and the Ringhals report are denoted reference.

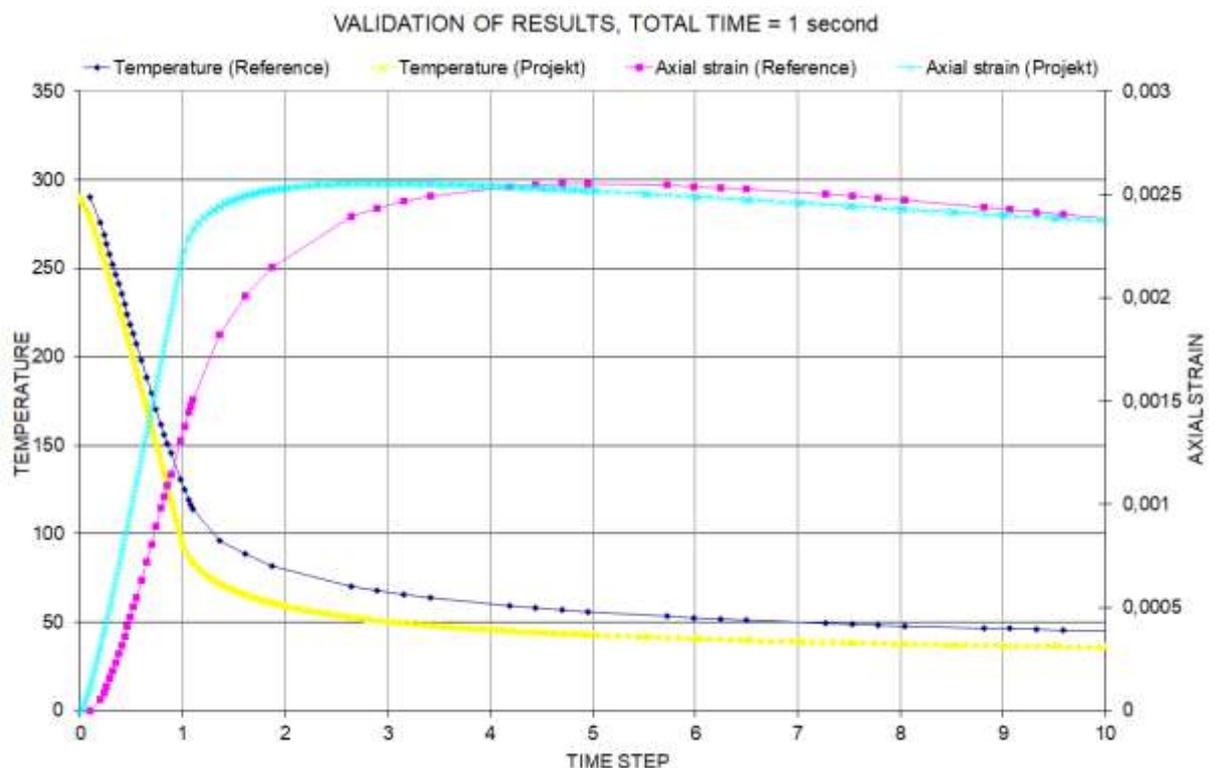


Figure 13: The temperature at the inner radius and the axial strain in a cross section for a pipe with thickness 63 mm, outer diameter 917 mm. Material steel 2333. Transient 290°C to 20°C in 1 second.

An analysis has been made earlier in ANSYS Classic on a pipe with wall thickness 60 mm with ramp times of 60 and 120 seconds. By comparing these two reference curves with the curve obtained in the current report with a quench time of 100 seconds it can be seen that the curve lies in-between the two reference curves which also confirms the present results. See figure 14-15. Blue = temperature reference curve, yellow = temperature current work, pink = strain reference curve, turquoise = strain current work

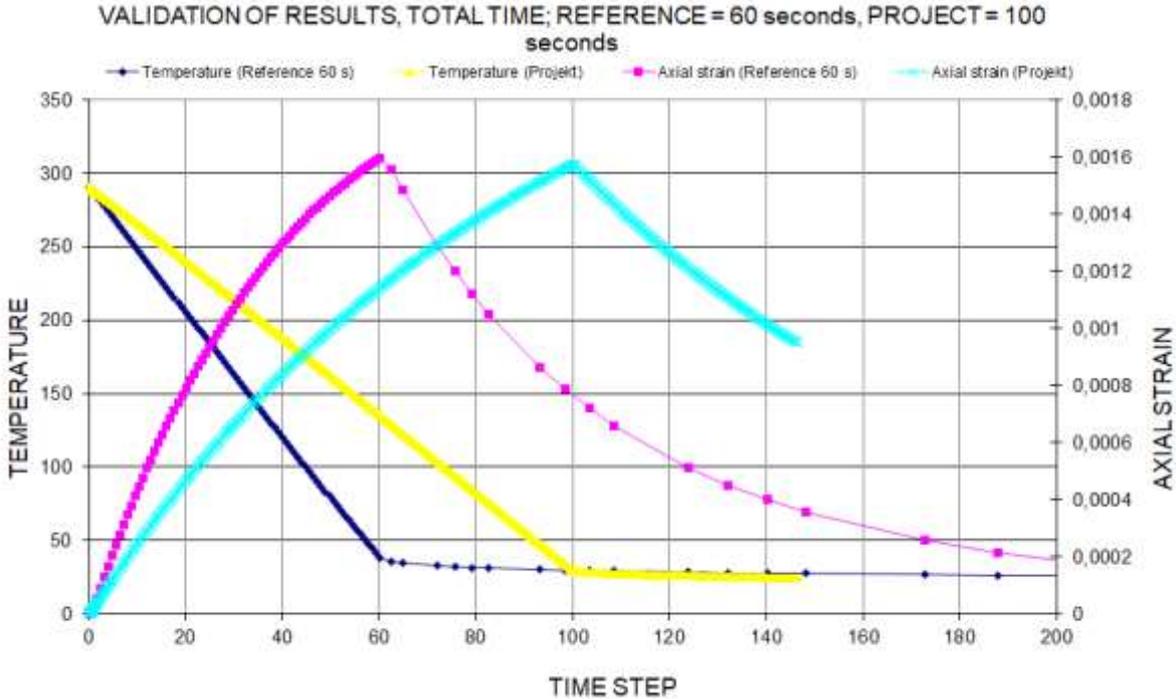


Figure 14: The temperature at the inner radius and the axial strain in a cross section for a pipe with thickness 63 mm, outer diameter 917 mm. Material steel 2333. Transient 290°C to 20°C in 60 sec (reference), 100 seconds (this project).

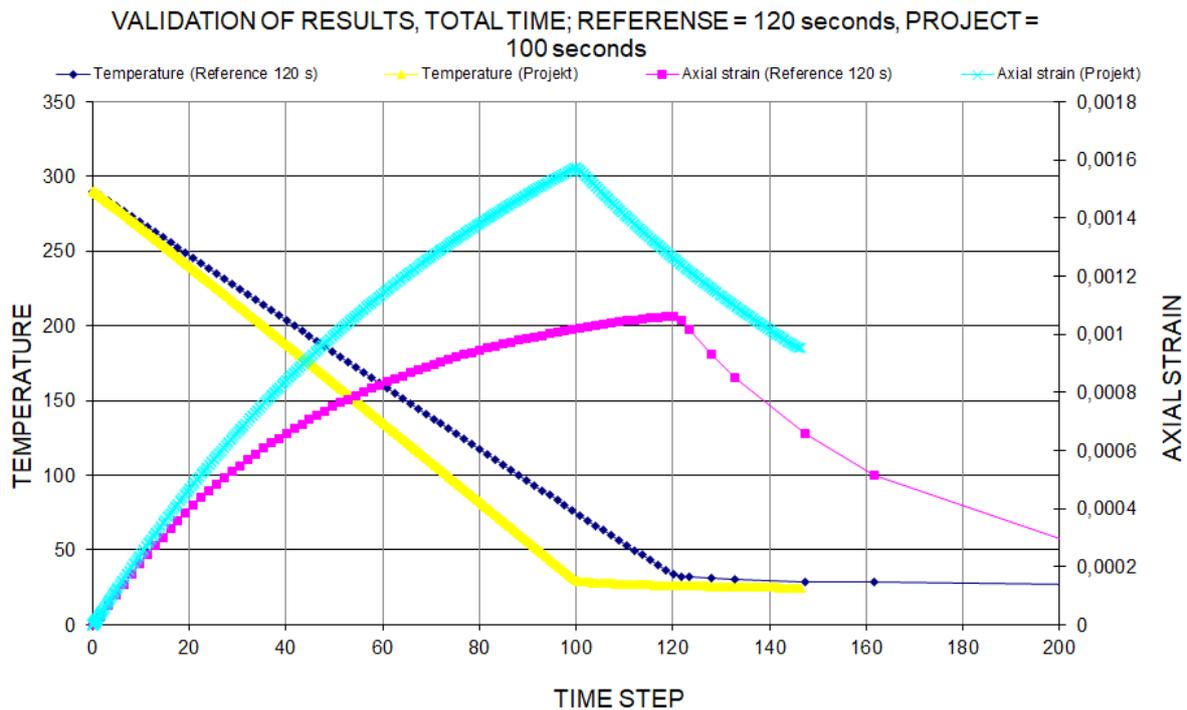


Figure 15: The temperature at the inner radius and the axial strain in a cross section for a pipe with thickness 63 mm, outer diameter 917 mm. Material steel 2333. Transient 290°C to 20°C in 120 sec (reference), 100 seconds (this project).

5.1.1 The elastic-plastic model

From a script made in Excel [5], a stress-strain curve, i.e., the axial stress as a function of axial strain has been modelled when plasticizing of the material is taken into account. The script follows the procedure according to ASME VIII [1]. The real material curve is illustrated at two temperatures, 20°C and 300°C. These are marked with the dark blue and the magenta curve, see figure 16-17, where figure 17 is a close-up of the beginning of the sequence of events in figure 16. An estimation of a bilinear curve is also made. With a tangent modulus of 2 GPa the bilinear curve seems to be a good approximation to the real material data. For a simple geometry, differences should be relatively small. For a complicated geometry, one can expect greater effects.

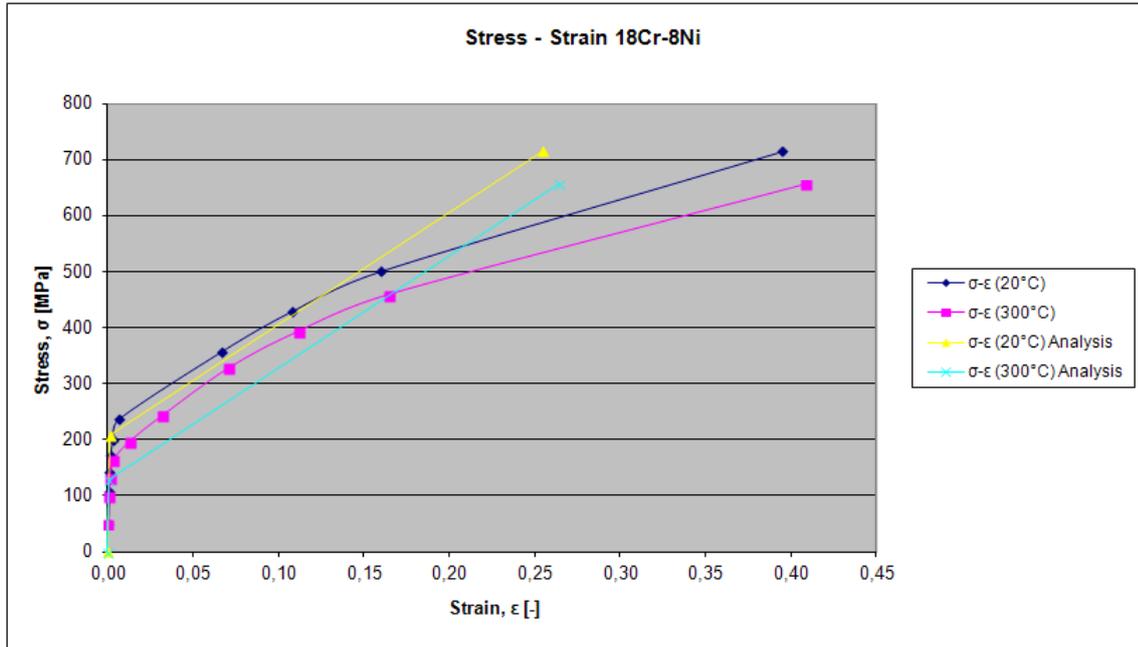


Figure 16: Real stress-strain curves and bilinear curves for 20°C and 300°C

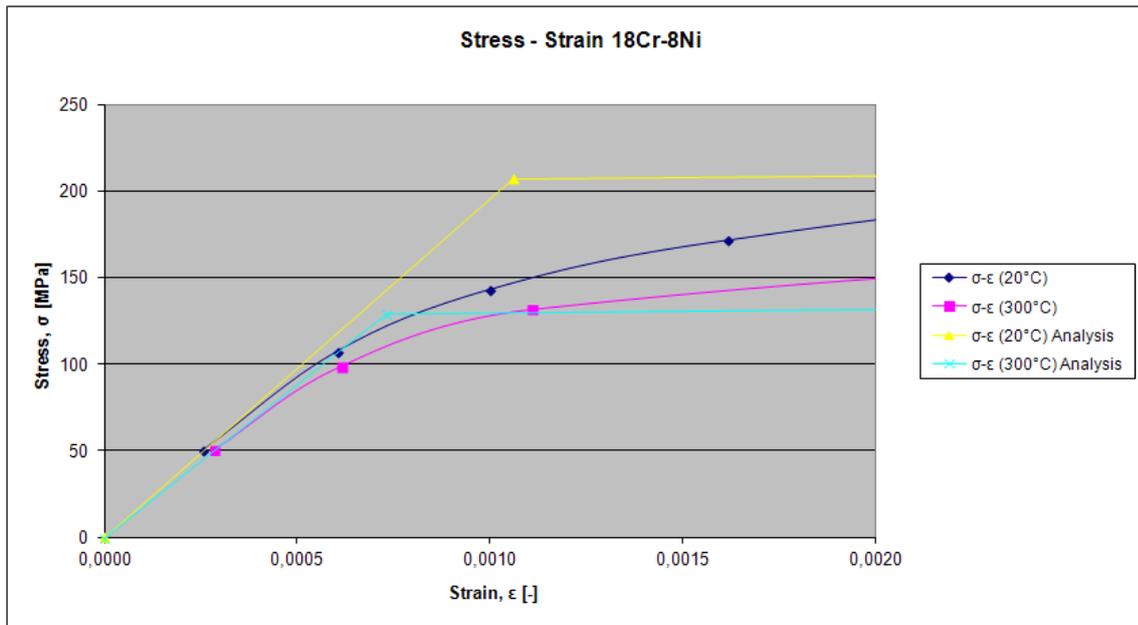


Figure 17: Real stress-strain curves and bilinear curves for 20°C and 300°C

5.2 Combined transients

In this thesis study only one transient is considered. However, in a real system there will be many transients occurring and there exist theories on how to calculate the fatigue correction factor with combined transients, see Appendix C.

6 Conclusion

The hope was to be able to use this model in ANSYS and apply it on other geometries that would probably be more sensitive to such huge transient temperatures. Another goal with this report was also to look at other transients and then combine several of these.

However, this model is thought to be a good basis that can be used for further investigation and that it will be possible to confirm new fatigue curves in light water reactor environments according to NRC due to the fact that the obtained results correspond to the already existing correction factors and previous examinations made at Ringhals.

To summarize what has been concluded in this report it can be seen that the thermal transients that affect the pipe in the worst manner for fatigue appears in PIPESTRESS for the thicker pipes when the temperature is down to 20°C in 10 seconds while 1 second is worse for the two thinner ones. In ANSYS on the other hand the worst case is 1 second ramping time no matter the thickness except for 60 mm when 10 seconds is worse and even though the results differ calculated with the two software the results are believed to be true, the temperatures essentially follow the same curve and the strains have the same shape of the slope. Why the strains aren't exactly the same is probably due to a too coarse approximate temperature gradient throughout the pipe wall and one average strain rate and you will thus be forced to calculate the strains by hand based on too few temperatures at each time point.

As for the effect of the thickness and the shape of the pipe on the fatigue correction factor it can clearly be seen when studying a temperature decrease of the coolant from 290°C to 20°C that the fatigue correction factor increases when the thickness of the pipe increases in both PIPESTRESS and ANSYS but the values are always slightly lower in ANSYS. When studying the elastic-plastic analysis where a stress-strain curve is used to calculate an approximate tangent modulus as the inclination of the plastic region, one can see that the bilinear curve seems to be a good approximation to the real material data. For a simple geometry, differences should be relatively small but for a complicated geometry, one can expect greater effects.

How can one implement the calculation procedure for a whole loading set instead of only one single transient and should compressive stresses also be taken into consideration or is it only the tensile stresses that affect the fatigue correction factor? In the methodology proposed in ASME, different transients are combined. If the environmental effects shall be taken into account during such a combination, it can be complicated to determine how the fatigue correction factor should be applied but there are two approaches on how to do this, one expressed in ASME and one less conservatively according to [10]. Since you should only consider F_{en} at tensile stresses, a suggestion is that when two transients are combined where one of these gives tensile stresses and the other compressive stresses, it is possible to calculate a total F_{en} factor, see Appendix C.

7 References

- [1] *An International Code - 2010 ASME Boiler & Pressure Vessel Code Section VIII Rules for Construction of Pressure Vessels - Division 2: Alternative Rules*. ASME. 2011.
- [2] O. K. Chopra and W. J. Shack. *Effect of LWR Coolant Environments on the Fatigue Life of Reactor Materials*. Prepared for Division of Fuel, Engineering and Radiological Research. 2007
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- [10] EPRI Project Manager J. Carey. *Materials Reliability Program: Guidelines for Addressing Fatigue, Environmental Effects in a License, Renewal Application, (MRP-47 Revision 1)*. ELECTRIC POWER RESEARCH INSTITUTE. 2005
- [11] Lennart Josefson, Chalmers University of Technology, personal communication, 2013
- [12] <https://confluence.cornell.edu/display/SIMULATION/ANSYS+Learning+Modules>

8 Appendices

8.1 Appendix A

The transformed sulphide content, dissolved-oxygen level, temperature and strain rate for carbon, low-alloy steels, austenitic stainless steels and Ni-Cr-Fe alloys respectively.

$$S^* = \begin{cases} 0.015, & DO > 1.0 \text{ ppm} \\ 0.001, & DO \leq 1.0 \text{ ppm and } S \leq 0.001 \text{ wt. \%} \\ S, & DO \leq 1.0 \text{ ppm and } 0.001 < S \leq 0.015 \text{ wt. \%} \\ 0.015, & DO \leq 1.0 \text{ ppm and } S > 0.015 \text{ wt. \%} \end{cases}$$

$$T^* = \begin{cases} 0, & T \leq 150^\circ\text{C} \\ T - 150, & 150 < T \leq 350^\circ\text{C} \end{cases}$$

$$O^* = \begin{cases} 0, & DO \leq 0.04 \text{ ppm} \\ \ln\left(\frac{DO}{0.04}\right), & 0.04 < DO \leq 0.5 \text{ ppm} \\ \ln(12.5), & DO > 0.5 \text{ ppm} \end{cases}$$

$$\dot{\epsilon}^* = \begin{cases} 0, & \dot{\epsilon} > 1\%/s \\ \ln \dot{\epsilon}, & 0.001 \leq \dot{\epsilon} \leq 1\%/s \\ \ln(0.001), & \dot{\epsilon} < 0.001\%/s \end{cases}$$

$$T' = \begin{cases} 0, & T < 150^\circ\text{C} \\ \frac{T - 150}{175}, & 150 \leq T < 325^\circ\text{C} \\ 1, & T \geq 325^\circ\text{C} \end{cases}$$

$$\dot{\epsilon}' = \begin{cases} 0, & \dot{\epsilon} > 0.4\%/s \\ \ln\left(\frac{\dot{\epsilon}}{0.4}\right), & 0.0004 \leq \dot{\epsilon} \leq 0.4\%/s \\ \ln\left(\frac{0.0004}{0.4}\right), & \dot{\epsilon} < 0.0004\%/s \end{cases}$$

$$O' = 0.281 \quad \text{At all DO levels}$$

$$T' = \begin{cases} \frac{T}{325}, & T < 325^\circ\text{C} \\ 1, & T \geq 325^\circ\text{C} \end{cases}$$

$$\dot{\epsilon}' = \begin{cases} 0, & \dot{\epsilon} > 5\%/s \\ \ln\left(\frac{\dot{\epsilon}}{5}\right), & 0.0004 \leq \dot{\epsilon} \leq 5\%/s \\ \ln\left(\frac{0.0004}{5}\right), & \dot{\epsilon} < 0.0004\%/s \end{cases}$$

$$O' = \begin{cases} 0.09, & \text{NWC BWR water} \\ 0.16, & \text{PWR or HWC BWR water} \end{cases}$$

8.2 Appendix B

The PIPESTRESS code used to calculate the temperature distribution over time for the pipe, received from my mentor Robert Magnusson.

```
IDEN JB=1000 CD=1 VA=0 GR=-Z IU=0 OU=0 SQ=10 AB=T
  EN=/Robert Magnusson/ PL=/40313/
TITL CV=17 GL=1 BL=3 PR=0 MD=1 HS=1 FL=1 FM=2 SM=0 SU=1 CU=1 SR=0 *17=ASME III
NB 2001+A02 * Rev 2
  TI=/R4 Connection HL to SG, Klass 1/
FREQ FR=1 MP=1 MX=1 RP=9 RF=201
  TI=/HOT MODAL EXTRACTION/
1000  PP  ITMAX 999

*1000  PP  MODPUR 25
*1000  PP  MGOOD 10
*1000  PP  ITMAX 900

*-----
*RBMG - Test av utmattning
*-----*
*----- LOAD CASES -----*
*-----*
LCAS CA=50 RF=201 TY=3 EQ=1 TI=/PD+DW/ * Dead Weight, filled

* COLD / HOT System
LCAS CA=200 TY=0 EQ=1 TI=/Cold/
LCAS CA=201 TY=0 EQ=8 TI=/Hot/
LCAS CA=202 TY=0 EQ=1 TI=/Transient Cold/
LCAS CA=203 TY=0 EQ=8 TI=/Transient Hot/

*****

*-----*
*-----TRANSIENTS-----*
*-----*

TCAS CA=11 RP=0 TI=/Transient 11/
TCAS CA=12 RP=0 TI=/Transient 12/

*-----*
*----- RESPONSE SPECTRUM LOADS -----*
*-----*

RCAS CA=221          * LOADCASE
  TY=2              * TYPE OF ANALYSIS (2=REQ SAFE SHUTDOWN)
  SU=0              * MODAL SUPERPOSITION OPTION
  LO=1              * LEFT OUT FORCE OPTION
  EQ=1              * OUTPUT
  FX=1              * LOAD FACTOR USED
  FY=1              * LOAD FACTOR USED
```

FZ=1 * LOAD FACTOR USED
EV=1 * EVENT NUMBER
TI=/GV_SSE/

FINAL DYNAMIC LEVEL A&B-CASES FOR FATIGUE ANALYSIS-

CCAS CA=222 RF=200 TY=1 EQ=1 ME=3
 C1=221 F1=1 OP=0
 TI=/RCAS, COLD DISP/
CCAS CA=223 RF=200 TY=1 EQ=1 ME=3
 C1=221 F1=-1 OP=0
 TI=/RCAS, COLD DISP/

-----FINAL DYNAMIC LEVEL A&B-CASES FOR EQ13-----

CCAS RF=200 CA=260 OP=1 FL=1 CY=5 C1=221
 TI=/PURE INERTIA RCAS/

**-----*
**-----FATIGUE ANALYSIS EQ 10, 11, 12, 13, 14-----*
**-----*

LSET RF=200 CY=1 PR=200 MO=200 *FC=1 * 1
 TI=/Shut Down/
LSET RF=201 CY=1 PR=201 MO=201 *FC=1 * 2
 TI=/Start Up/

LSET RF=202 CY=600 PR=202 MO=202 TR=+11 *FC=1 * 3
 TI=/Transient 11/

LSET RF=203 CY=600 PR=203 MO=203 TR=-12 *FC=1 * 4
 TI=/Transient 12/

* DYNAMIC CASES (Not used, no stresses)

LSET RF=200 CY=1 PR=200 MO=222 FL=1 *FC=1 * 32 *Dynamiskt, se CCAS
 TI=/RCAS UP/
LSET RF=200 CY=1 PR=200 MO=223 FL=1 *FC=1 * 33 *Dynamiskt, se CCAS
 TI=/RCAS DOWN/

*FATG AT=1010 AF=1000

--DUMMY-----
*LSET RF=201 CY=0 PR=201 MO=201 FC=0 TI=/DUMMY/

----- SPECTRUM -----

SPEC EV=1 FP=0 ME=3 TI=/Spectra/
LV=1 DI=X
0.500/0.000 50.000/0.00
DI=Y
0.500/0.000 50.000/0.00
DI=Z
0.500/0.000 50.000/0.00

----- MATERIAL DEFINITION -----

* MATERIAL: SA351 CF8A
* KÄLLA: ASME II Part D, 2010 (metric)
MATH CD=351 TA=20 EX=1 TY=4 TX=375 AL=/SA351 CF8A/
MATD TE=20 EH=195 EX=15.3 SH=152 SM=161 SY=216
MATD TE=50 EH=192.8 EX=15.6 SH=148 SM=161 SY=216
MATD TE=100 EH=189 EX=16.2 SH=142 SM=161 SY=198
MATD TE=125 EH=187.5 EX=16.4 SH=138 SM=159 SY=188
MATD TE=150 EH=186 EX=16.6 SH=134 SM=156 SY=180
MATD TE=200 EH=183 EX=17.0 SH=130 SM=150 SY=167
MATD TE=250 EH=179 EX=17.4 SH=128 SM=142 SY=157
MATD TE=300 EH=176 EX=17.7 SH=128 SM=135 SY=150
MATD TE=325 EH=174 EX=17.8 SH=128 SM=132 SY=147
MATD TE=350 EH=172 EX=17.9 SH=128 SM=130 SY=144
MATD TE=375 EH=170.5 EX=18.0 SH=128 SM=127 SY=141

----- CROSS-SECTIONAL PROPERTIES -----

* Side A
CROS CD=918 OD=918.75 WT=63 MA=1328 SO=1 ST=1
CROS CD=961 OD=961.50 WT=87 MA=1876 SO=1 ST=1

----- PIPING -----

----- RUN C1 -----
DESN PR=17.1 TE=343

OPER CA=200 PR=0 TE=20 * Cold
OPER CA=201 PR=15.41 TE=290 * Hot
OPER CA=202 PR=0 TE=20 * Kall omgivning
OPER CA=203 PR=15.41 TE=290 * Varm omgivning

* Transients Section 2 ****

PAIR CA=11 CO=71.3 EX=15.3 DI=4.52
PAIR CA=12 CO=71.3 EX=15.3 DI=4.52

TRAN CA=11 IS=1 FS=1 IT=20 FT=290 TT=1 FL=35000 * Flöde ca 21000m3/h
IP=15.41 FP=15.41 TP=0 AL=/Transient 11/
TRAN CA=12 IS=1 FS=1 IT=290 FT=20 TT=1 FL=35000 * Flöde ca 21000m3/h
IP=15.41 FP=15.41 TP=0 AL=/Transient 12/

MATL CD=351

CROS CD=918
ANCH PT=1000 LV=1
TANG PT=1010 DX=0.5
TANG PT=1020 DX=0.5
*INDI AT=1020 B1=0.5 B2=1.0 C1=1.4 C2=1.2 C3=1.0 CP=0.6 K1=1.5 K2=1.8 K3=1.5
*CROS CD=961
*TANG PT=1030 DX=0.5
*TANG PT=1040 DX=0.5
*MOMT PT=1040 CA=201 MX=1930

ENDP

8.3 Appendix C

In the methodology proposed in ASME, different transients are combined. In ASME the combining is performed as follows:

- Calculate the principal stresses S_1, S_2, S_3 at the combined load cycle (two transients).
- Calculate the maximum stress range throughout the cycle by $S = \max(S_1 - S_2; S_2 - S_3; S_3 - S_1)$.
- Calculate the alternating stress, $S_{alt} = 0.5 * S$. Use this to get the number of allowable cycles from a fatigue curve. In other words, from the graph you get the number of allowable cycles, N , for the current combination and then compare this with the actual number, n .
- The fatigue usage factor is then determined by $U = \frac{n}{N}$.

If the environmental effects shall be taken into account during such a combination, it can be complicated to determine how the fatigue correction factor should be applied.

- In approach 2, section 2.1.1, equation 2.7, an easy way of calculating an appropriate fatigue correction factor for a single transient is presented. When combining two transients one can use the maximum correction factor for the current transients and expressed as $U * F_{en}$ to calculate the fatigue correction factor in a conservative manner.
- Less conservatively the fatigue correction factor can be calculated according to MRP-47 [10], where two transients are put together in a rough manner.
- Since you should only consider F_{en} at tensile stresses, a suggestion is that when two transients are combined where one of these gives tensile stresses and the other compressive stresses, one can calculate a total F_{en} factor. Let us say that the first transient gives a tensile strain and a tensile stress of 10 and a F_{en} factor of 2. The second transient gives a compressive strain and a compressive stress of 5 and a F_{en} factor of 1 (because there is pressure). The combined F_{en} ratio could then be calculated as: $\frac{(10 \times 2 + 5 \times 1)}{15} = 1.67$.