INFLUENCE OF SUSTAINED AND EXPANSION LOADS ON LIFETIME PREDICTION OF PIPING COMPONENTS OPERATING IN THE CREEP RANGE

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ABSTRACT

One of the most common failure modes for elevated temperature piping working in the creep conditions is initiation, growth of cracks in pipe bends that finally leads to the failure. Failure of bends is much more dangerous than appearance of cracks in the piping joint welds or tees due to size of affected areas.

It is known that for pipe bends, working in creep range, attention must be paid to imperfections of the cross-section, such as out-of-roundness and different wall-thickness, and their interaction with the internal pressure as well as bending and torsion moments.

FE analyses of different bend configuration have been performed to investigate effect of the above factors on the creep damage of this element. It was recognized that out-of-roundness and bending moments are significant contributors to the stress state of pipe bends and hence in the creep cumulative damage. Neglecting of these factors could lead to big uncertainties in the remaining life prediction.

INTRODUCTION

Nowadays Conventional Power Plants designed in Russia as well as in European Countries in 60 – 70 have exhausted their original design operation resource set as 100000 hours. In some cases operating time reaches the value 250000 – 300000 hours.

One of the most common failure modes for elevated temperature piping working in the creep conditions is initiation, growth of cracks in pipe bends that finally leads to the failure.

Failure of bends is much more dangerous than appearance of cracks in the piping joint welds or tees due to size of affected areas.

From the previous studies there was conclusion that "with pipe bends, attention must be paid to imperfections of the cross-section, such as out-of-roundness and different wall-thickness, and their interaction with the internal pressure as well as bending and torsion moments", [1-5].

Actual design piping codes and standards [6, 7] do recognize creep damage as governing failure mode for the high temperature piping, but they are based on the linear-elastic calculations and utilize simplified equations addressing creep damage effects. Codes contain only some limitation for the bend out-of-roundness and do not consider it for Pressure Design.

At the same time the methodologies for assessment of the remaining life for the high temperature components take into account only pressure and neglect mechanical and thermal expansion loads acted during operational life of piping. For the creep assessment they utilize simplified methods with application of isochronous curves or calculation of stress during creep in the steady state.

CONSTITUTIVE MODEL OF VISCOELASTIC MATERIAL

In this study, a mathematical model of the material was defined according to the Kachanov-Rabotnov damage theory [8, 9]. Unlike a conventional Norton power law, used for the materials working in the creep conditions, this model takes into account the tertiary stage of the creep and defines parameter of damage which measures the "time of failure".

The specific characteristics of the material model parameters were taken from the paper [10] for steel 1Cr0.5Mo at 550°C.

Formulas (1) - (4) presented below describe the Kachanov-Rabotnov law:

$$\frac{d\varepsilon_c}{dt} = A\sigma_e^m\left[(1-\rho)+\rho(1-\omega)^{-p}\right] \tag{1}$$

$$\frac{d\omega}{dt} = g\frac{B}{1+r}\frac{\tilde{\sigma}^k}{(1-\omega)^r} \tag{2}$$

$$\omega_{crit} = 1-(1-g)^{\frac{1}{1+r}} \tag{3}$$

$$\tilde{\sigma} = \alpha\frac{|\sigma_1|+\sigma_1}{2}+(1-\alpha)\sigma_e \tag{4}$$

where: ε_c - creep strain; ω - damage parameter. Varies from zero to ω_{crit}, that corresponds to the failure; σ_e - equivalent Misses stress; σ_1 - principal stress; $\tilde{\sigma}$ - stress accounting a multiaxial loading conditions.

Table 1 contains values of the parameters used in the above equations. Units for stresses in this table MPa, time measures in hours.

Table 1. Material Model Parameters for steel 1Cr0.5Mo at 550°C.

A	$1.940\text{x}10^{-15}$	p	4.354	B	$8.325\text{x}10^{-13}$	r	1.423	α	0.43
m	4.354	ρ	0.393	k	3.955	g	0.9755		

Assuming uniaxial stress state with constant stress (σ_0), equations (1) - (4) could be integrated and solved to define relationships for the time-to-rupture t_R, rapture strain ε_R, and time-dependent relationships for the damage parameter ω and creep strain ε_e:

$$t_R = \frac{1}{B\sigma_0^k} \tag{5}$$

$$\varepsilon_R = A\sigma_0^m t_R\left\{(1-\rho)+\rho\frac{\lambda}{g}\left[1-(1-g)^{\frac{1}{\lambda}}\right]\right\}; \quad \lambda = \frac{1+r}{1+r-p} \tag{6}$$

$$\omega = 1-(1-g\frac{t}{t_R})^{\frac{1}{1+r}}; \quad 0 \le \omega \le \omega_{crit} \tag{7}$$

$$\varepsilon_c = A\sigma_0^m t_R\left\{(1-\rho)\frac{t}{t_R}+\rho\frac{\lambda}{g}\left[1-(1-g\frac{t}{t_R})^{\frac{1}{\lambda}}\right]\right\} \tag{8}$$

Figure 1 and Figure 2 show time-dependent plots for the damage parameter and creep strain for the constant stress.

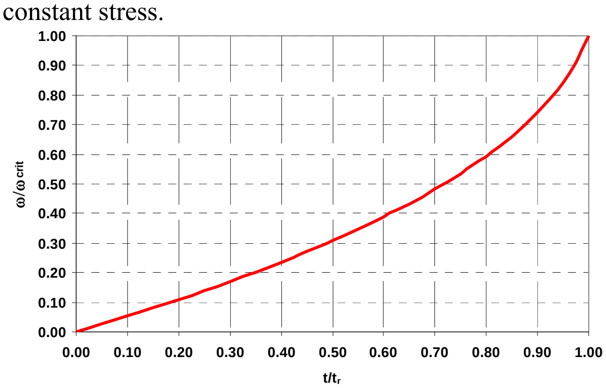


Figure 1 Relative damage versus relative time to failure.

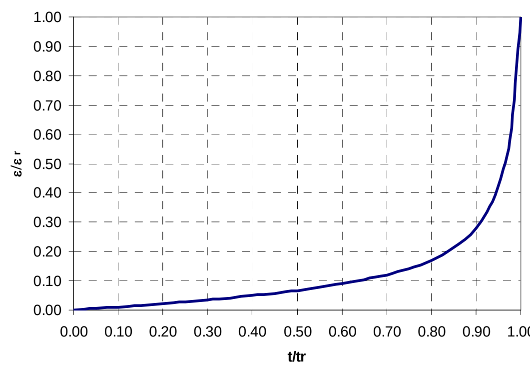


Figure 2 Relative creep strain versus relative time to failure

For temperature T = 550°C Young Modulus for the steel 1Cr0.5Mo was taken as E = $1.6\text{x}10^5$ MPa, Poisson ratio $\nu = 0.3$.

DIMENSIONS OF PIPE BENDS

Three standard components of the hot reheat piping 426x19 were considered in the frame of this study: two bends with different radii (600 mm (R600) and 1700 mm (R1700)) and straight pipe.

Thickness variation for the bend cross-section was defined according to equation (9), [11]. Calculated values for three typical locations are given in Table 2.

Initial ovality of the bend cross-section was defined by (10) and was varied in the range from 0% to 6%. To introduce this ovality in analysis the shape of the bend cross-section was modeled by two semi-ellipses, Figure 3.

$$\frac{s}{s_0} = \frac{2R}{2R \pm (D_o - s_0)} \quad (9)$$

$$a = \frac{D_{\max} - D_{\min}}{D_o} \times 100 \quad (10)$$

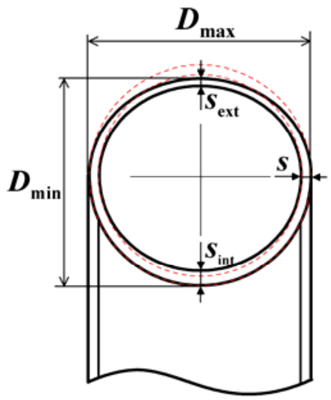


Figure 3 Geometry of the bend cross-section

APPLIED LOADS

An internal pressure for all considered variants of analyses was assumed to be 5 MPa. To define values for the in-plane bending moments the following approach was applied: according to EN 13480 (Design of piping components under internal pressure) the wall thickness for straight pipe and bends should satisfy to Equations (11) and (12). Assuming wall thickness for considered components as defined above, it is possible to recalculate corresponding level of stresses. A maximum stress (58.4 MPa) corresponds to the bend R600 due to thinning on the extrados. This value was taken as a basis for the further analyses. Substituting this value to the equation (5) service life (time to failure) for R600 bend is tR = 124 000 hours. Calculated stresses and the corresponding "normative" period of service for the elements under study are listed in Table 2.

wall thickness for the straight pipe:

$$s = \frac{pD_o}{2[\sigma] + p} \quad (11)$$

wall thickness for Bend:

intrados

$$s_{\text{int}} = s\frac{(R/D_o) - 0.25}{(R/D_o) - 0.5}$$

extrados

$$s_{ext} = s\frac{(R/D_o) + 0.25}{(R/D_o) + 0.5} \quad (12)$$

Table 2. Code Based Stresses and recalculated service life.

Component	p, MПa	Do, mm	s, mm	R, mm	s_{ext}/s	s_{int}/s	s_{ext}	s_{int}	σ, MPa	$[t_R]$, hours
Pipe	5	426	19						53.6	174698
Bend	5	426	19	600	0.80	1.45	15.2	27.6	58.4	124115
Bend	5	426	19	1700	0.89	1.14	17.0	21.6	56.8	138657

Further: according to EN (Flexibility analysis and acceptance criteria) the equivalent stresses under different load sets are defined as:

$$\sigma_1 = \frac{pD_o}{4s} + 0.75i\frac{M_A}{Z} \le [\sigma] \quad (13)$$

$$\sigma_5 = \frac{pD_o}{4s} + 0.75i\frac{M_A}{Z} + 0.75i\frac{M_C}{3Z} \le [\sigma] \quad (14)$$

$$0.75i \ge 1; \; i = \frac{0.9}{h^{2/3}}; \; h = \frac{sR}{r_m^2} \quad (15)$$

Substituting in equation (13) the values for the internal pressure and stresses calculated on the previous stage (Table 2) we can obtain the value of the maximum-allowable moment from the sustained mechanical loads (M_A) and similarly, considering Equation (14) and assuming $M_A = 0$, the maximum-allowable moment from the thermal expansion loads (M_C). Results of these calculations are summarized in Table 3.

Table 3. Calculations for applied bending moments

Component	P, MPa	D_o, mm	t, mm	R, mm	$PD_o/2t$, MPa	Z, mm^3	h	i	0.75i	M_A, N*mm	M_C, N*mm
Pipe	5	426	19		56.1	2.37E+06		1	1	7.19E+07	2.16E+08
Bend	5	426	19	600	56.1	2.37E+06	0.275	2.127	1.595	4.51E+07	1.35E+08
Bend	5	426	19	1700	56.1	2.37E+06	0.780	1.062	1.000	7.19E+07	2.16E+08
Pipe	0	426	19		0.0	2.37E+06		1	1	1.38E+08	4.15E+08
Bend	0	426	19	600	0.0	2.37E+06	0.275	2.127	1.595	8.67E+07	2.60E+08
Bend	0	426	19	1700	0.0	2.37E+06	0.780	1.062	1.000	1.38E+08	4.15E+08

Table 4 presents a matrix of the performed FE calculations.

Table 4. Matrix of performed calculations

Load Combination	Bend Radii, mm	Ovality
P	600, 1700	0%, 3%, 6%
$P\pm M_A$; $P\pm M_A/2$		
$P\pm M_C$; $P\pm M_C/2$		
$\pm M_A$; $\pm M_C$		

Totally 78 FE calculations have been performed for the bends and also several calculations for the straight pipe under pressure and sustained bending moment M_A.

FINITE ELEMENT MODELING

All FE analyses have been performed with ANSYS, [12]. To reflect a real boundary conditions of the bend as a part of piping, FE models of bends included bend itself and 2 parts of straight pipes attached to both ends. One end of this model was fixed while the second kept free. Mechanical in-plane bending moment M_A was applied at free end. Thermal expansion moment M_C was simulated by applying equivalent rotation at free end. Both ends of the pipe were considerer as closed, Figure 4.

Since the bending moments were applied in plane, a quarter of the model was considered due to symmetry. Model was meshed with use of three layers of 3D elements SOLID186 along the wall. Closed ends were modeled by means of MPC184 elements. Internal pressure was applied as distributed load to the inner surface of the FE model, together with an axial tension equivalent to the internal pressure applied at the end of pipe to simulate the closing end.

A typical finite element model for the pipe bend is shown in Figure 5.

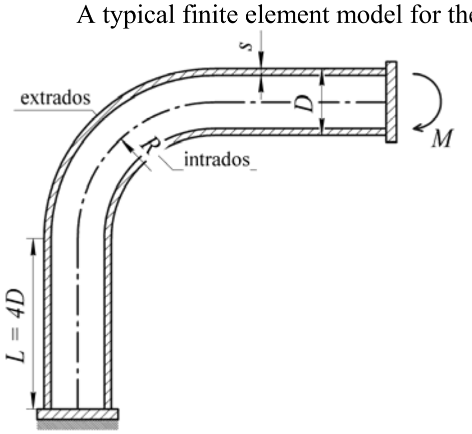


Figure 4. Loading Scheme

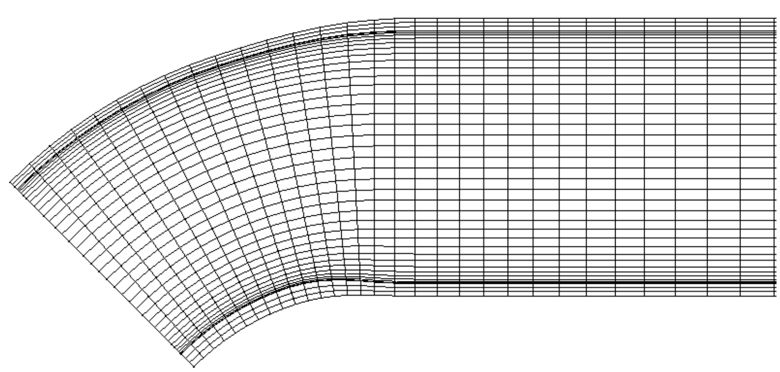


Figure 5. FE model of pipe bend

Two custom subroutines USERCREEP and USEROUT were developed to reflect Kachanov - Rabotnov damage theory in the frame of ANSYS. The calculations were performed in geometrically nonlinear formulation.

After applying of the initial loading the step by step solution was performed up to the time when automatically controlled value of damage parameter at any point of the model reached the critical value ω_{crit}. Time step of solution was controlled by program automatically (parameter CREEPLIM was 0.25).

RESULTS OF ANALYSES

Influence of the Studied Parameters on the Service Life of Piping Components

The influence of investigated parameters on the service life of piping components is demonstrated in the following diagrams and figures. The ratio $t_r/[t_r]$ represents a relative time before failure: t_r is numerically calculated value from the FEA, while $[t_r]$ is value defined as "normative" service life (see Table 2). As was explained above t_r corresponds to the time of "failure", when damage parameter ω reaches in any point the critical value ω_{crit}. It should be noted that the difference in the time between first appearing ω_{crit} in one point and spreading this value along the whole wall thickness is practically negligible.

The following conclusions could be drawn from the Figures 6 and 7:

1. Ovality up to 3% has slight effect on service life of components.
2. Ovality 6% reduces the service life about 20% for bend R600 and about 30% for bend R1700 compared with the case where there is no ovality. It should be noted that this reduction does not depend on the combination of load factors.
3. Thus, the first important conclusion from this work is that during design of the high temperature piping bends it is necessary to take into account ovality on the stage of Pressure Design to properly define wall thickness. At the same time, there is no need to include this effect in other stages of the code based stress analysis.
4. Thermal Expansion bending moment M_C has no practical influence on the service life. At the same time the bending moment from the mechanical loads, M_A leads to the reduction of service life up to 40% (see results for R600 for combination of pressure and opened moment: P-M_A). A significant reduction of the service life occurs only when the moment has a value that corresponds to the maximum that Code allows. An other observation is that according to the equation (13) Code allows sufficiently high value for the bending moment M_A. At the same time, in practice, the magnitude of the bending moment from the weight loads is limited by the recommended span of supports. For example, for the given pipe size (426x19) a recommended maximum span between supports is 12 m. Bending moment corresponding to this span is two times less than the moment M_A, calculated in accordance with (13) As could be seen from the presented diagrams the moment $M_A/2$ has insignificant influence on the service life.
5. The calculations show that in most cases the damage starts from the outside side of the extrados, Figure 8. But in some cases the damage begins on the inside surface and located on the flank of the bend (Figure 9). This fact may provide additional relevant information during the nondestructive examination.
6. Additional set of the calculations was carried out for the studying an effect of the pure bending moment in the absence of the internal pressure. The results showed that the most unfavorable is a closed moment MA. Service Life under such loading is substantially lower than defined by Code and ovality of the section reaches 50%. This fact once again indicates that the maximum allowable by Code bending moment may not be acceptable for piping components working in the creep conditions. This fact should be taken into account for the design of high temperature low pressure sodium piping for NPPs with liquid metal breeder reactor. On the other hand, the thermal expansion moment MC in the absence of pressure does not lead to the failure: the cumulative damage over 316 000 hours does not exceed 0.25.

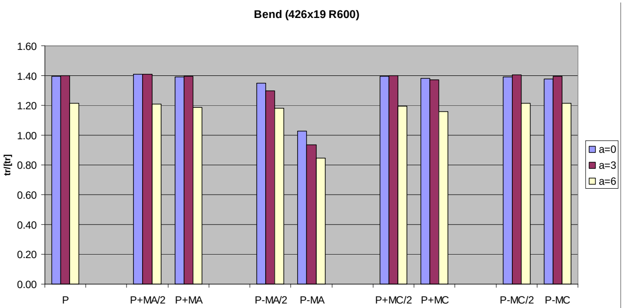


Figure 6. Service Life Ratio for BEND R600

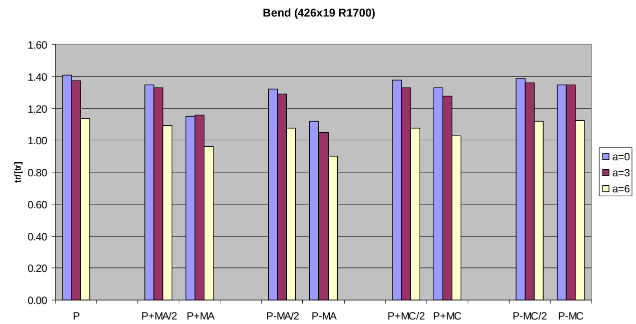


Figure 7. Service Life Ratio for BEND R1700

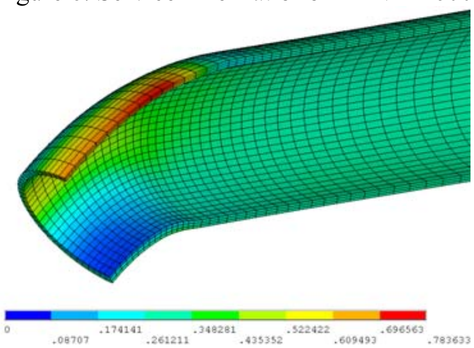


Figure 8. Location of the damage on the Extrados outer surface

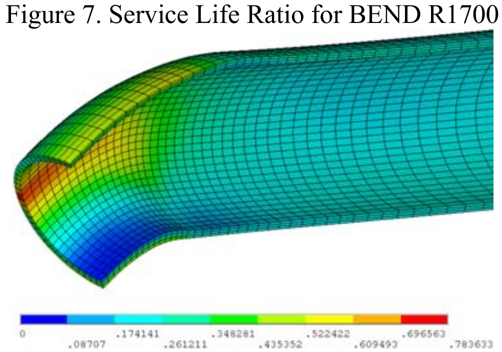


Figure 9. Location of the damage on Flank inner surface

Time-dependent change of ovality

Next set of the results refers to the study for the change of the ovality depending on the relative time to the failure. Figures 10-12 show the time-dependent ovality under different load combination.

The following conclusions are based on the analysis of these results:

1. Under the pure pressure, or pressure in combination with the bending thermal expansion moment M_C, the ovality decreases and tends to zero.
2. Under the pressure in combination with the bending moment from the sustained loads, M_A, ovality depending on the magnitude and sign of the load factor tends to a definite value, regardless of the initial value of ovality.
3. In the case of the absence of the internal pressure ovality can reach a significant values, unacceptable for the normal operation of the piping, Figure 13.
4. It is interesting to note that in the practical assessment of the piping residual life the change of bends ovality during service life is considered as indicator of accumulated damage. As seen from the figures, this approach can give good results when bends are subjected to pressure and thermal expansion bending moment. In the presence of the mechanical bending moment (from the weight) such approach can give unreliable results.

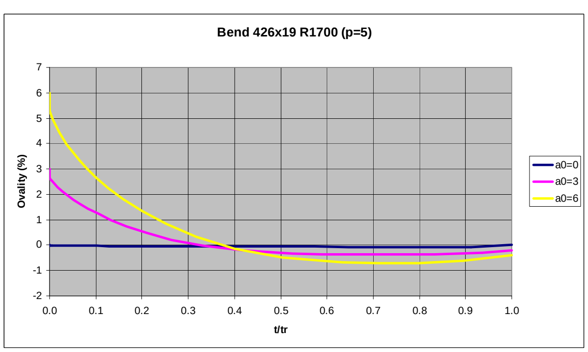


Figure 10. Change of ovality depending on time (P)

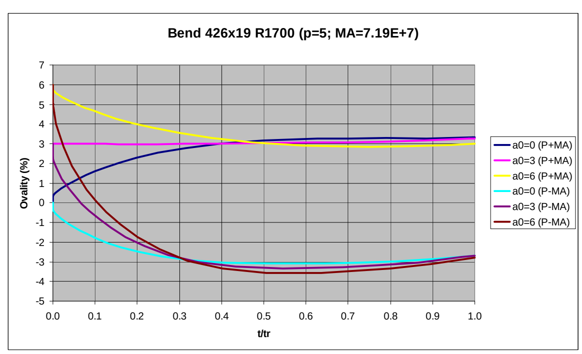


Figure 11. Change of ovality depending on time (P±M_A)

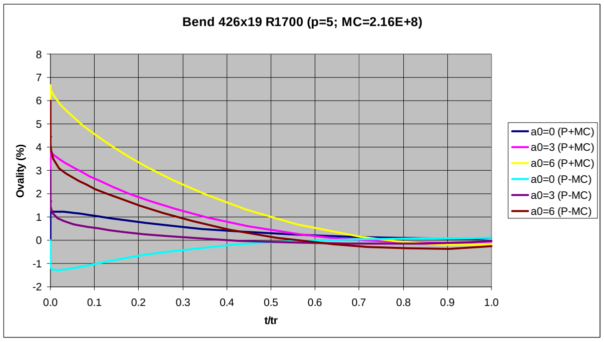


Figure 12. Change of ovality depending on time (P±M_C)

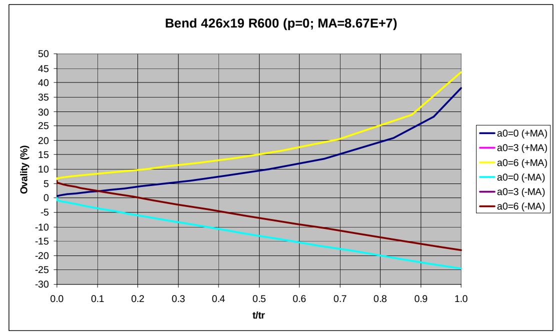


Figure 13. Change of ovality depending on time for the bend without pressure under mechanical bending moment

Change of the damage and stresses during service life of piping components

Figures 14-15 show the graphs of the relative damage parameter and level of equivalent rupture stresses depending on the lifetime of the considered piping components. The results are presented for the locations at which the damage parameter reached his maximal value. The following conclusions could be drawn from these results:

1. After approximately 10% of the service life stresses are redistributed due to the creep despite the fact that the initial level of the elastic stresses is significantly different for the different load combinations. Stresses tend to a constant level in the absence of the initial ovality. At the same time, for bends with 6% initial ovality, stresses continuously change during service life. This observation points to the incorrect use in this case a simplified estimates based on the assumption of steady state of stresses. It should be noted that the curve corresponding to combination of P-M_A, differs from the general trend of stresses (this load combination corresponds to the minimum period of service). This effect is most significant for the bend R600.
2. A curve corresponding to the constant stresses was added in the graphs that show the change of damage. Calculated curves of the damage for the load cases where stresses are constant practically all over the time (initial ovality is zero) are practically coincide with constant stress curve. For those variants for which the stresses change during the lifetime (a = 6%), the constant stress curve of damage is below the calculated curves. This suggests that the accumulation of the damage occurs more rapidly in the initial period of operation and slows down at the end of life.

Damage

Stresses

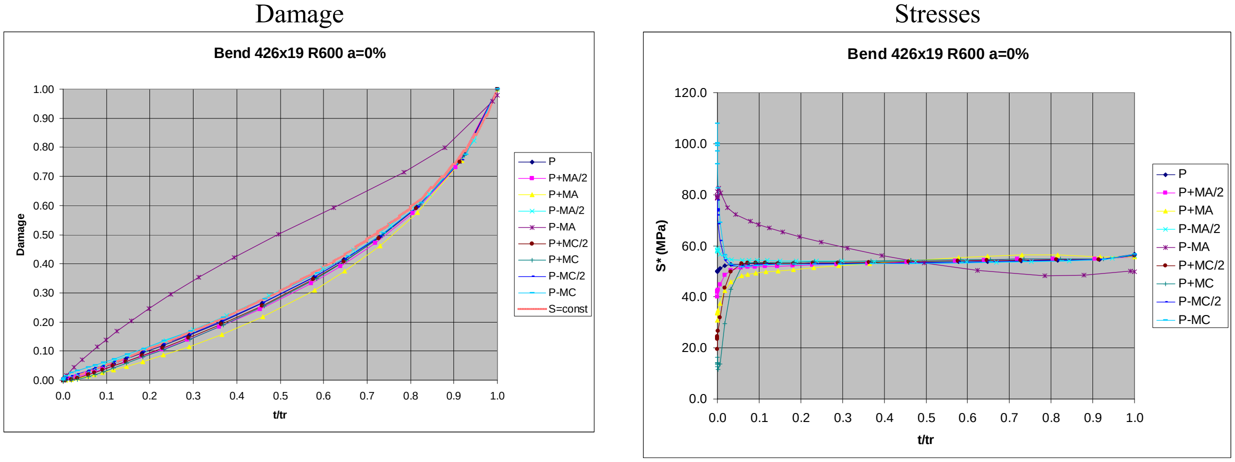
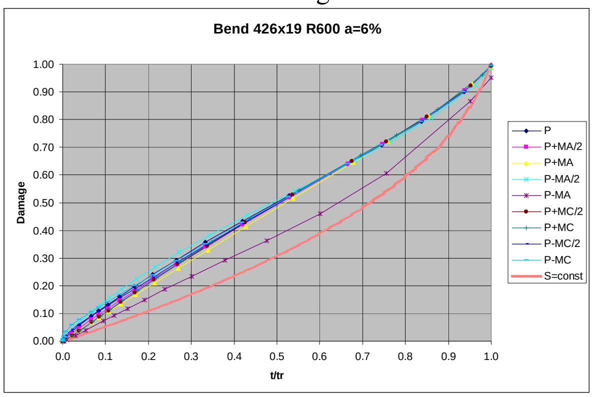


Figure 14. Change of stresses and accumulated damage over the time (a=0%).

Damage



Stresses

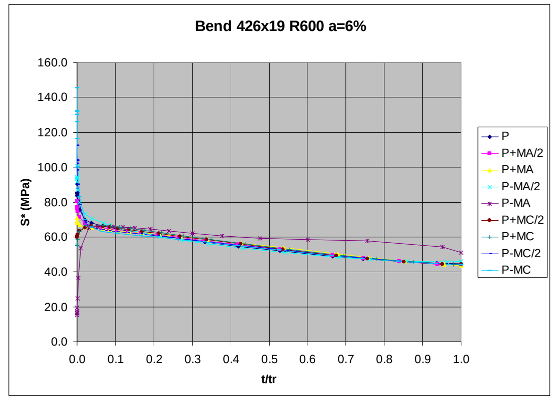


Figure 15. Change of stresses and accumulated damage over the time (a=6%).

CONCLUSIONS

1. Ovality and bending moments in some cases are significant contributors to the stress state of the pipe bends and hence in the service life of bends working in the creep range. Neglecting of these factors could lead to big uncertainties in the remaining life prediction.
2. For the creep damage sustained loads are more important than thermal expansion loads.
3. For the pressure design of the piping bends working in the creep range ovality should be introduced in the Code equations. Alternatively, limits for the permissible initial ovality may need to be reviewed.
4. Results of this study can be effectively used at the design of the new power and nuclear high temperature piping. Moreover, it was shown, that detailed study of the creep damage under actual loads could be decisive on the reliability and safe operation of the high-temperature piping.

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