ANALYSIS OF THE FOREVER-EXPERIMENTS USING A FINITE ELEMENT MODEL

Eberhard Altstadt, Thomas Moessner, Baj Ral Sehgal [1], Hans-Georg Willschütz

1. Introduction

For future nuclear power plants it is demanded that there are no consequences for the environment and the population even in the closest vicinity of the plant during and after every possible accident scenario. This includes the hypothetical scenario of a severe accident with subsequent core meltdown and formation of a melt pool in the reactor pressure vessel (RPV) lower plenum of a Light Water Reactor (LWR). One accident management strategy is to stabilize the in-vessel debris configuration in the RPV as one major barrier against uncontrolled release of heat and radio nuclides. This strategy also applies to existing plants.

To get an improved understanding and knowledge of the melt pool convection and the vessel creep and possible failure processes and modes occurring during the late phase of a core melt down accident the FOREVER-experiments (Failure Of REactor VEssel Retention) are currently underway at the Division of Nuclear Power Safety of the Royal Institute of Technology Stockholm [1]. These experiments are simulating the behaviour of the lower head of the RPV under the thermal loads of a convecting melt pool with decay heating, and under the pressure loads that the vessel experiences in a depressurization scenario (see Fig. 1). The geometrical scale of the experiments is 1:10 compared to a common Light Water Reactor (LWR).

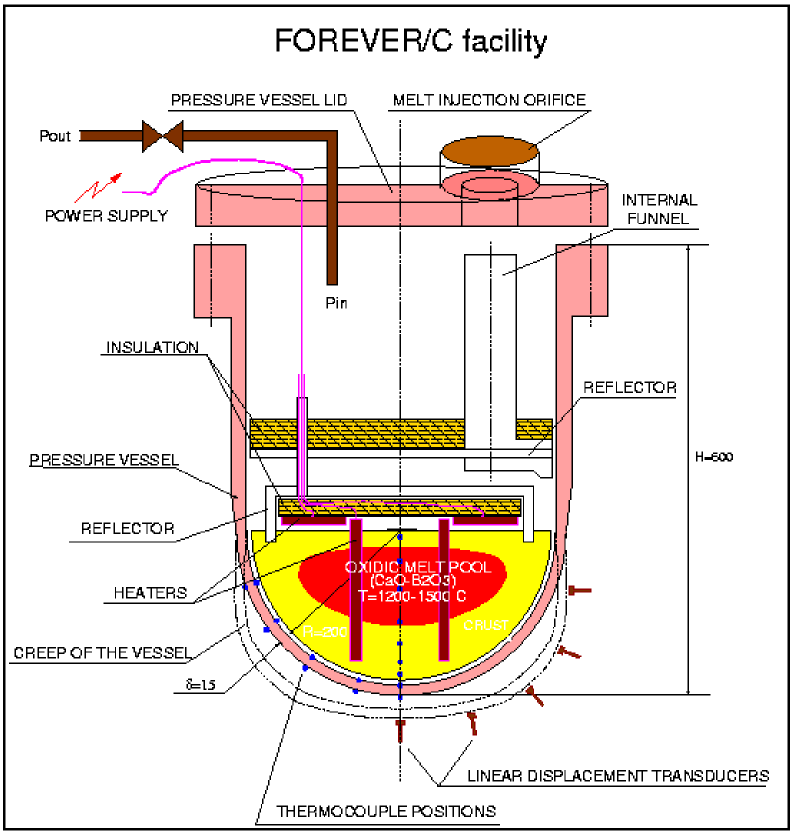

Fig. 1: Principal scheme of the FOREVER/Creep tests. Scheme is not to scale.

During the first series of experiments the creep behaviour (FOREVER-C) of the vessel under the thermal attack of the melt pool and varying internal pressure loads is investigated. It is intended to enforce the creep process until vessel failure. Due to the multi axial creep deformation of the vessel with a non-uniform

[1]Royal Institute of Technology, Division of Nuclear Power Safety, Stockholm, Sweden

temperature field these experiments are on the one hand an excellent source of data to validate numerical creep models which are developed on the basis of uniaxial creep tests. On the other hand the results of pre-test calculations can be used to optimize the experimental procedure with considerations of the uncertainties in the applied models and assumed boundary conditions.

Therefore an axis-symmetric Finite Element (FE) model is developed based on the multi-purpose code ANSYS/ Multiphysics®. Using the Computational Fluid Dynamics (CFD) module the temperature field within the melt pool and within the vessel wall is evaluated. The transient structural mechanical calculations are then performed applying a creep model which takes into account large temperature, stress and strain variations.

Taking into account both - experimental and numerical results - provides a good opportunity to improve the simulation and the understanding of the vessel failure mechanisms. Of particular interest are (i) the time to failure and (ii) the location and mode of failure.

2. Experimental results

The analysis of this work refers to the experiment FOREVER-C2. Due to a rather high heat power input (Q_{max}>40kW) the external temperatures reached more than 900°C in the hot focus region at the upper part of the hemisphere. In consequence a significant creep of the vessel wall was observed. The hemisphere was made of the French RPV steel 16MND5. The applied oxidic melt was a CaO-B_2O_3 mixture (30-70 wt.-%) which has a solidus temperature of T_s=1250K and is a rather aggressive oxide, especially at high temperatures. To model the internal decay heat generation special designed heater rods fixed to an internal insulation-reflector-lid are immersed into the melt from the top. The lid is fixed to the upper part of the vessel. After melt pouring the melt injection orifice in the vessel lid is closed and the vessel inside can be pressurized by Argon. The total duration of the experiment C2 was nearly 10 hours from the start of pre-heating the vessel to the stop of the data recording at the end of the cooldown and relaxation stage. After some 120 min there was a temporarily heater shut down to install additional cooling units to the power supply cables. With the re-gained power supply and temperature level the system was pressurized at t=200min. In Figure 2 the total displacement U_{sum} at different positions of the external vessel surface during the course of the experiment is shown. The thermal expansion of the vessel due to the hot melt pool can clearly be seen. The start of the pressurization indicates

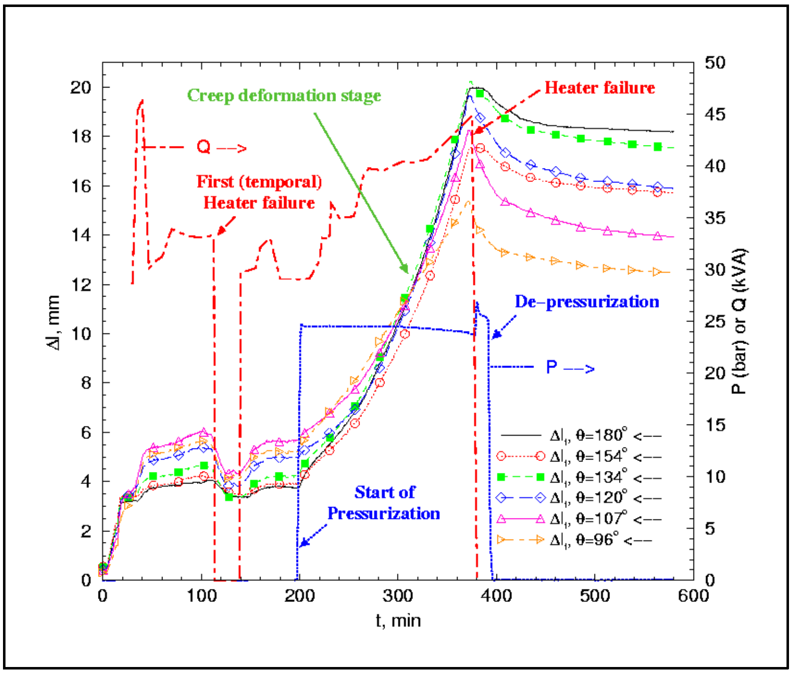

Fig. 2: General course of the experiment C2: power input Q/[kVA], pressure P/[bar] and total displacement U_{sum}/[mm] at different external positions of the hemisphere. $\Theta = 180°$ refers to the very bottom.

the beginning of the creep deformation stage. The creep curves show an acceleration of the creep strain which indicates normally tertiary creep. So far the reason for the observations in C2 is not exactly known.

Due to the creep expansion of the vessel the volume of the hemisphere is increasing to the third power of the increase of the radius. This causes a decreasing melt level and as a consequence in the experiment C2 the uppermost parts of the heater rods were no longer immersed in the pool. Therefore they burned out and the power supply was stopped (for a detailed description of the experiment see [1]).

3. CFD analysis of convection and heat transfer

For the evaluation of the temperature field within the vessel wall the CFD-module FLOTRAN® of the FE-code ANSYS® is used. A 2D-axis-symmetric model with appropriate boundary conditions and material properties is developed. A pure homogenous melt pool is assumed inside the vessel with the surface level set to the welding joint between hemisphere and cylinder. The mesh consist of 1740 elements of which 1400 belong to the liquid region at the beginning of the calculation. Due to some prior estimations it was found that the main heat transfer mechanism at the model boundaries is radiation. Therefore at the vessel outside an radiative heat transfer boundary condition is applied with an ambient temperature of T_{amb}=400K. A surface emissivity of e_{steel}=0.8 has been used. Also for the internal surfaces radiative boundary conditions have been modelled: between insulation and melt pool (cf. Fig. 1) T_{amb} was set to 1200K, above insulation it was 800K. The emissivity was the same as on the outside. A homogenous volumetric heat source is assumed which is applied to the volume within which the heaters are to be found. Especially at the very bottom the distance between the heater and the vessel wall has a significant influence to the crust formation. The internal Rayleigh number for this configuration is calculated in a range of $Ra_i=0.5\ 10^{10}$ to $Ra_i=1.0\ 10^{10}$.

To model the heat transfer within the pool the standard k-ε-turbulence model which is provided by FLOTRAN® is used. Assuming slow temperature changes in the vessel wall and in the lower part of the melt pool a dynamic crust is modelled by stopping the solution every 20 seconds and checking the temperature field. For those elements where the melt temperatures are below the solidus temperature of T_s=1250K at all nodes, the material number is changed so that these elements belong to

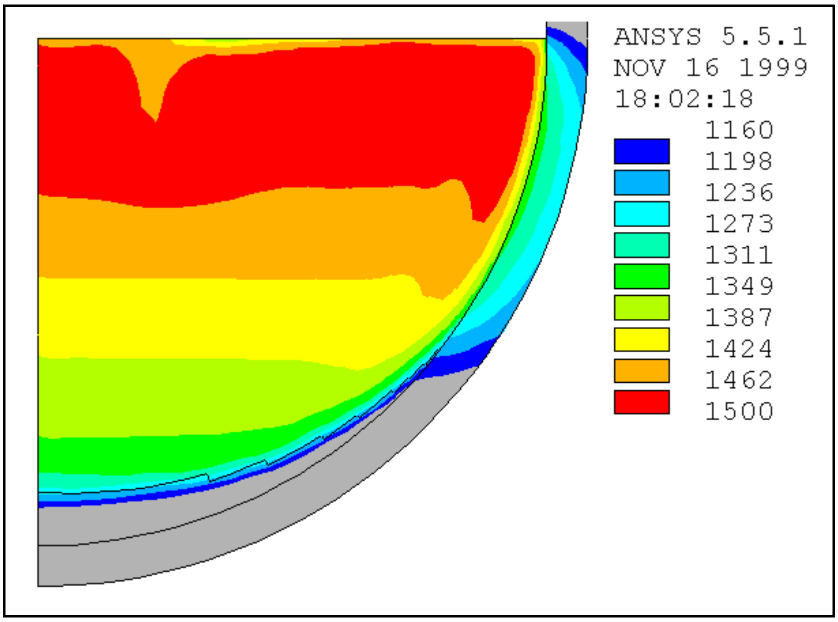

Fig. 3: Calculated temperature field in the hemispherical part with a power input of Q=35kW. T in [K], adjusted scale ranges from 1160K to 1500K, temperatures below 1160K are presented grey.

the solid region of the oxidic crust. So far different power inputs were modelled in transient calculations starting with homogenous initial temperatures within melt pool and vessel wall. The heat generation rates were chosen according to the experimental range which had changed from 30kW to 45kW. In these calculations a thermal steady state for the vessel wall was observed after some 20-30min. The calculated temperature profiles show a good agreement with the measurements in the high temperature region of the vessel. Figure 3 shows the temperature field for the heat generation rate of 35 kW. The melt shows a stable stratification. Therefore, the hottest region of the vessel wall is the upper part of the hemisphere.

4. Structural analysis of the vessel creep

The mechanical 2D-axis-symmetric model of the vessel wall consists of nearly 340 elements and some 410 nodes with 5 element layers over the wall thickness. A sufficient number of elements over the wall thickness is necessary to model the transient body load due to the temperature field which is taken from the CFD analysis.

Because of the large spatial and transient temperature and stress changes within the vessel wall an advanced approach for the numerical creep modelling has been developed. Usually creeping is described by analytical formulas (creep laws) with a number of free coefficients. The coefficients are used to adapt the creep laws to creep test results performed at constant load and temperature. However, it is difficult to achieve a satisfying adjustment for a wide range of temperatures and stresses with only one set of coefficients. Therefore a supplementary tool for the ANSYS code has been developed which allows to describe the creep behaviour of a material for different stress and temperature levels independently. Moreover, it is possible to calculate the creep damage and deactivate elements whose accumulated damage is greater or equal to one. The Digital® Fortran Compiler (Rev. 6.0A) was used for programming and for generating the customized ANSYS-executable on a Windows/NT® platform. The creep data base has been generated using an experimental results analysis performed by [2]. The covered stress range of the data base reaches from σ_{min}=5MPa to σ_{max}=285MPa and the temperature ranges from T_{min}=873K to T_{max}=1373K.

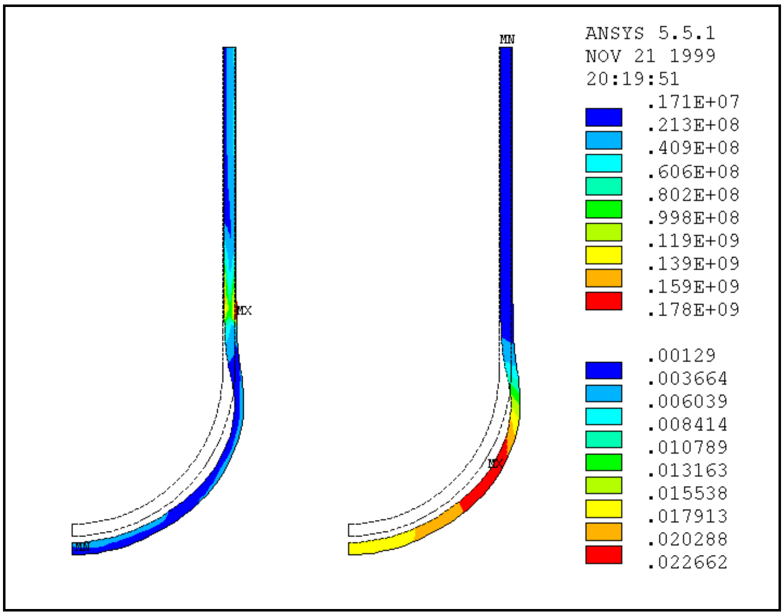

Fig. 4: Von Mises equivalent stress σ_{eqv} [Pa] (left side and upper scale) and total displacement U_{sum} / [m] (right side and lower scale) of the vessel after t=22800s (heater failure in the experiment).

Until pressurization of the system the displacement measured by the linear displacement transducers (LDT) is only due to thermal expansion. Therefore the transient creep calculation starts at the experimental pressurization time of t=12000s (cf. Fig. 5). Figure 4 shows the calculated equivalent stress and the total displacement at the time 22800s (heater failure in the experiment). If a constant temperature field within the vessel wall is assumed for the transient creep calculation a typical creep curve will be calculated by the code (see Fig. 5, blue curve). This curve is characterized by a steep - but decreasing - increase of creep strain (primary creep stage). After that the creep strain rate becomes nearly constant (secondary creep).

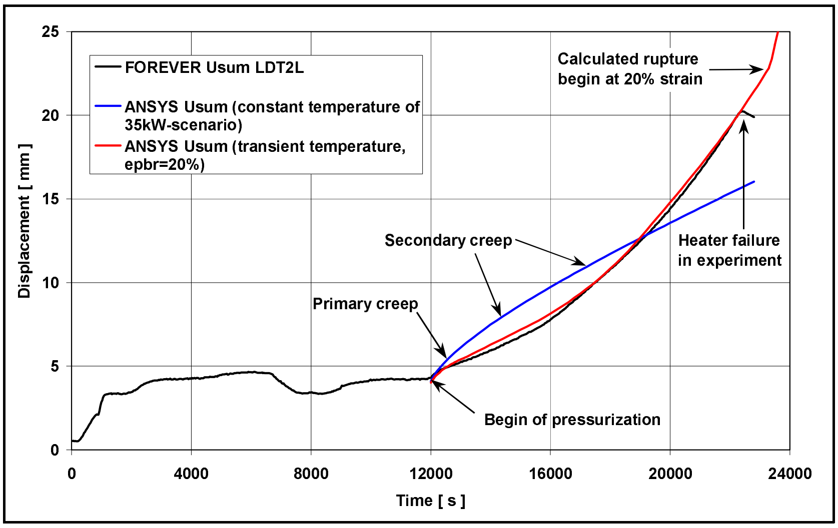

Fig. 5: Total displacement U_{sum} / [m] of the vessel external surface on the left side at position $\Theta = 134°$ over time t / [s].

In fact the total displacement in the experiment (black curve) shows the opposite behaviour to the calculation with constant temperature. The curve looks like a tertiary creep curve, but considering the stress and temperature regimes at this time and the total duration of 3 hours tertiary creep is very unlikely. Performing a transient calculation with a changing temperature field according to the recorded power input (cf. Fig. 2) the calculated creep follows the red curve in Figure 5. This causes a temperature drop at the beginning of the creep deformation stage and a temperature increase after the first half hour until heater failure. Of course this model has to be improved and this will be possible with a temperature field available from a future transient CFD-calculation.

The most interesting question for the next experiments is the vessel failure time. It is not intended to have the vessel failed at the high pressure load. Therefore, different scenarios have to be calculated. In the calculations shown here a slightly increasing heat input is assumed (instead of the unintentional heater failure). Thus the temperature increase has been set to 10K/h and the high pressure load was kept on until vessel failure. With an assumed very conservative creep rupture strain of ε^{frac}=20% for all stresses and temperatures this leads to a vessel failure after t_{frac}=23700s. This means just 15min after the heater failure the vessel would have failed. But considering the uniaxial creep test data of the 16MND5-steel [2] and the results of the Lower Head Failure Tests [3] at the Sandia National Laboratories even a creep

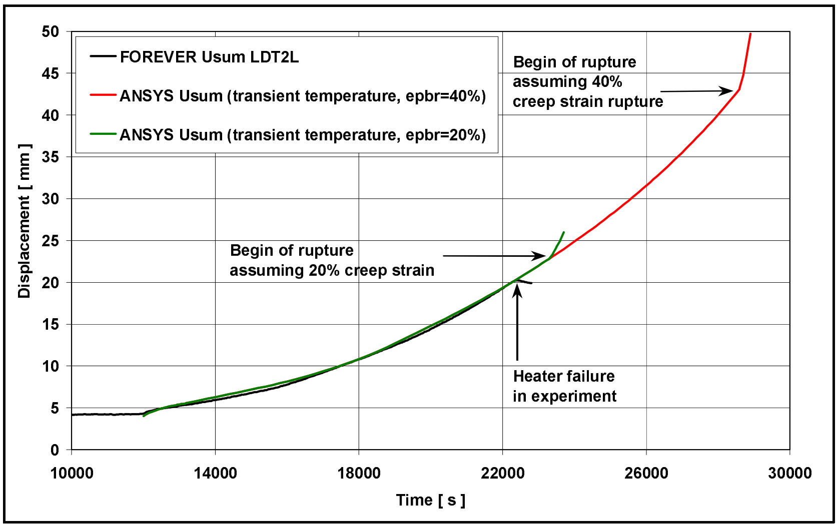

Fig. 6: Estimation of vessel failure time with different creep rupture strains. Total displacement U_{sum} / [mm] of the vessel external surface on the left side at position $\Theta = 134°$ over time t / [s].

rupture strain of ε^{frac}=40% can be stated as conservative. Figure 6 shows the total displacement at Θ=134° for both cases. In the 40%-scenario failure could be expected after t_{frac}=28800s, which means more than one hour after heater failure in FOREVER-C2. Of course, this is a first estimation and further investigations have to be performed.

5. Conclusions

The post test calculations of the FOREVER-C2 experiment show that the behaviour of the vessel - made of French RPV steel - is rather sensitive to temperature changes during the creep deformation stage. Therefore, it seems that the unexpected deformation behaviour during the experiment is caused by the transient temperature field in the vessel wall rather than by a tertiary creep process.

The model will be improved considering the transient thermal boundary conditions which have a great influence to the transient creep calculation. First rupture estimations made clear that one of the main mechanical parameters to be investigated in the future is the creep rupture strain.

References

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