IMPACT LOADING OF A BWR CONTROL ROD DURING BRAKING

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SUMMARY

In an emergency case the control rods of a boiling water reactor are shot into the RPV from below against the dead weight of the rods with drive motors. According to the position of the control rods between the fuel elements the rods can reach in that case velocities up to 4 m/s. The moved masses of the control rods and of the pistons (both of them are connected by a coupling) are braked through a cup spring which transfers its forces to the RPV-bottom sphere. The spring has to be designed that in this case the complete kinetic energy of the control rods of about 1000 Nm can be taken up. The spring power and the inertia of the moved masses cause extremely high loadings during and shortly after the impact onto the spring. The shock-like loading propagates along the whole rod at the speed of sound, and this is also the reason why the weaker cross-sections have to endure considerable short-term stress peaks.

Because of the dimensions (small width, great length) the calculation could be done one-dimensional on the base of the difference method. For the determination of the decelerations exact mass-modeling was necessary. Thus the stresses can be easily calculated from the forces with the appropriate cross-section. The whole model was divided into 145 mass zones (equation of state for austenitic steel), with which a precise survey could be determined about critical positions. The spring was modeled as a mass zone, which equation of state coincided with the spring characteristics. Apart from the spring zone all mass zones had the initial velocity of 4 m/s. It could be seen that by the impact the first zone had a shock-deceleration of almost 40,000 g, followed by an oscillation at higher frequency with $\pm$20,000 g. This effect is of continuous-mechanical nature as one can see with the result of the mean initial deceleration of the whole rod which is about 100 g. During the problem time of about 5 ms the whole rod vibrated with $\pm$50 g at a frequency of more than 10,000 Hz. After 5 ms the velocity decreased to 3.65 m/s and the spring had absorbed the work of 150 Nm over a compression length of 0.015 m. The stresses reached values up to 12.E8 N/m^2 in the first mass zone during the impact, in the further process the values vibrated at $\pm$5.E8 N/m^2. At some points the strains reached maximal values of 0.1%. With this calculation the results of some tests with control rods could be confirmed. Additionally this calculation may give prospects for a fatigue analysis of the used material by a determination of loading cycles. It has to be emphasized that the calculation had been done with static material constants, which can not be exactly applied to shock dynamic loadings.

Nevertheless the demonstrated way of calculation shows that it is possible to registrate the complicated, continuous-mechanical method well and to transfer them to global effects.

1. Introduction

At the Boiling Water Reactor (BWR) the control rods are shot from below into the vessel because of nuclear steam generating in the upper part of the vessel. The control rod drive has to perform two functions:

a) to move the rods slowly for the control of the nuclear reaction during operation (Fig.1a). This happens with an electric motor with axial threaded spindle. This process is unimportant for the design of the mechanical strength of the rods.

b) to insert the rods extremly fast with pressurized water at emergency conditions (Fig.1b). This process is connected with high accelerations and decelerations, which results in high loadings because of the inertia of the masses.

In the following only the process at emergency conditions will be treated, as this is important for the design of the control rods and of the hollow pushrod.

2. Description of the process during impact

The control element is pushed upwards together with the hollow pushrod (both are connected with a coupling) by pressurized water, which comes into the control rod drive housing. According to the position of the control rods between the fuel elements and according also to the positioning time the rods can reach velocities up to 4 m/sec, which may hit the throttle bush. The moved masses are braked by a Belleville spring, which transmits the deceleration forces to the bottom sphere of the pressure vessel. The spring, which has a non-linear characteristic, has to absorb the total kinetic energy of the rod of about 1000 Nm. The spring force and the inertia of the rod masses cause extremly high loadings during and short after the impact onto the spring. The shock - like loading propagates along the whole rod at the speed of sound, and this is also the reason why the weaker cross - sections have to endure considerable short term stress peakes.

3. Determination of the stresses

The knowledge of the stresses during the shock dynamic loading is necessary for the design. Therefore a lot of experiments have been run with original rods. The measurements were done with strain gauges and acceleration instruments. The results however were valid only for a special case. Only with reservation one can transmit the results to general statements. Analytically the impact effects are calculated by computer codes, especially for continuous - mechanical items. For this purpose a calculation was done with the code PISCES, which integrates the finite difference equations under the terms of spacial and time - dependant coordinates. The result considers the energy-, the momentum- and the continuity - equations. The stability of the time integration is given by automatic cycle control /1/, /2/.

4. Model

The geometry of the control rods (small thickness, long rod)(Fig.2a) allows an one-dimensional calculation. For this purpose the real geometry was changed to a rotational-symmetric model. Finally the density of the different parts of the rod had to be recalculated to relative densities. This changes the real frequency of the loading along the rod, but in order to get the magnitude of the frequency this effect is neglectible.

The spring was modeled as a mass zone (Fig.2b). This gives the chance to calculate the spring deformation and the real impact. Though there is a gap in the coupling this part was modeled as a rigid connection, because the resulting disturbation changes the dynamic response only in the upper frequencies. For the same reason the gap between the absorber tube and the upper resp. the lower connecting section was neglected. The mass of the absorber tubes was taken into account.

The whole rod was devided into 145 mass-zones to get some local effects as local strain and local stresses. The parameters of the equation of state and of the yield model were taken from austenitic steel. Only in the spring zone the equation of state was the spring characteristic and the yield model was simplified as hydrodynamic; all the other zones got the initial velocity of 4 m/sec. No damping was included.

5. Results

The experiments have shown, that the maximum loading is given mainly right at the impact of the hollow pushrod onto the throttle bush. Therefore the problem time was limited to 4.26 msec to calculate only the initial effects. The mean velocity, i.e. the mean value of the single zone velocity powered by its mass relation to the total mass, decreases within this time to 3.65 m/sec. The kinetic energy decreased from 1050Nm to 920 Nm (Fig.3). The various local effects are the reason for the oscillating curve. The spring force reached already the value of $1.6 \cdot 10^4$ N, which means that the force-displacement relation of the spring characteristic is still linear. At higher spring displacements (20 mm) the non-linear characteristic begins. It should also be mentioned, that in this calculation the water compression was taken into account by prestressing the spring.

The first zone of the throttle bush got a local initial deceleration of the magnitude of $2.5 \cdot 10^5$ m/sec^2; this shows the time-history plot of Fig. 4. Compared with this result the mean deceleration of the whole rod had values of 130 g. The frequency was approximately 15000 Hz.Fig.4 shows the propagation of the wave caused by the impact.

After 2.2 msec the shock-wave, which was reflected in the meantime at the free end of the control rod, reached again the initial part. The second peak results in the new reflection at the spring mass. As there is no damping nearly no decrease of the peak values was found in the range of 4 msec (Fig.5)

No part of the rod had reached the yield stress. As expected the highest stress-values were found on two parts:

a) the lower part of the hollow pushrod, where the impact took place
b) the lower connecting section, which is the weakest part of the cross-section of the control rod /Fig.6/ shows that the peak value of 1.25 10^8 N/m^2 is still far below tolerable stresses /3/ therefore in this case there is no danger of failure.

Nevertheless a fatigue analysis will be necessary with respect to the endurance strength /3/. With the frequency and the number of specified emergency conditions one can determine the total cycles, which the material has to endure.

6. Comparison to experimental results

Internal measurements with original rods have shown accelerations and decelerations, which correspond to the analytical results within a few percent. It has to be emphasized that the calculations have been done with static material constants and that for the comparison with the experimental results only some transformations in the initial and boundary conditions have been necessary.

The good accordance of the results, which could be seen inspite of the above mentioned restrictions, demonstrates the sufficient design of the control rods. As the measured and calculated frequencies have been in the same order of magnitude, it can be stated that the analytical model with the used material description gives sufficient knowledge to high dynamic processes like this impact.

7. Conclusion

The demonstrated calculations show, that this analysis combined with the experiment is a practical way to registrate the complicated, continuous-mechanical effects and to transfer the results to global behaviour. It is now easy and economical to investigate other parameters like higher velocities of the control rod, other materials and other spring characteristics in order to optimize the construction.

References

/1/ PISCES 1 DL. Manual A "General Description and finite - difference Equations" Physics International Comp. San Leandro USA 1972

/2/ PISCES 1 DL. Manual B "Input Manual" Physics International Comp. San Leandro USA 1972

/3/ ASME Code Section 3 Edition 1974

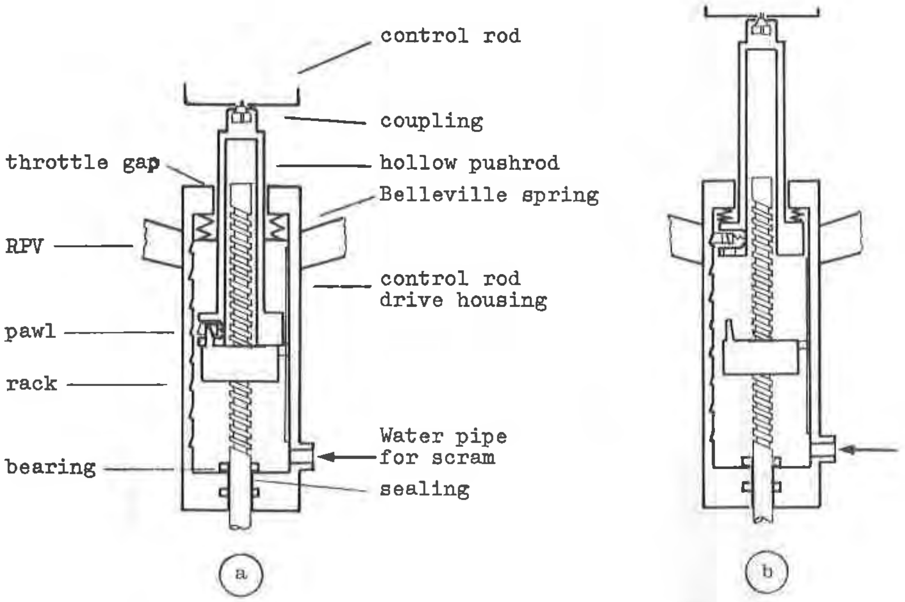

Fig. 1: Scheme of control rod drive

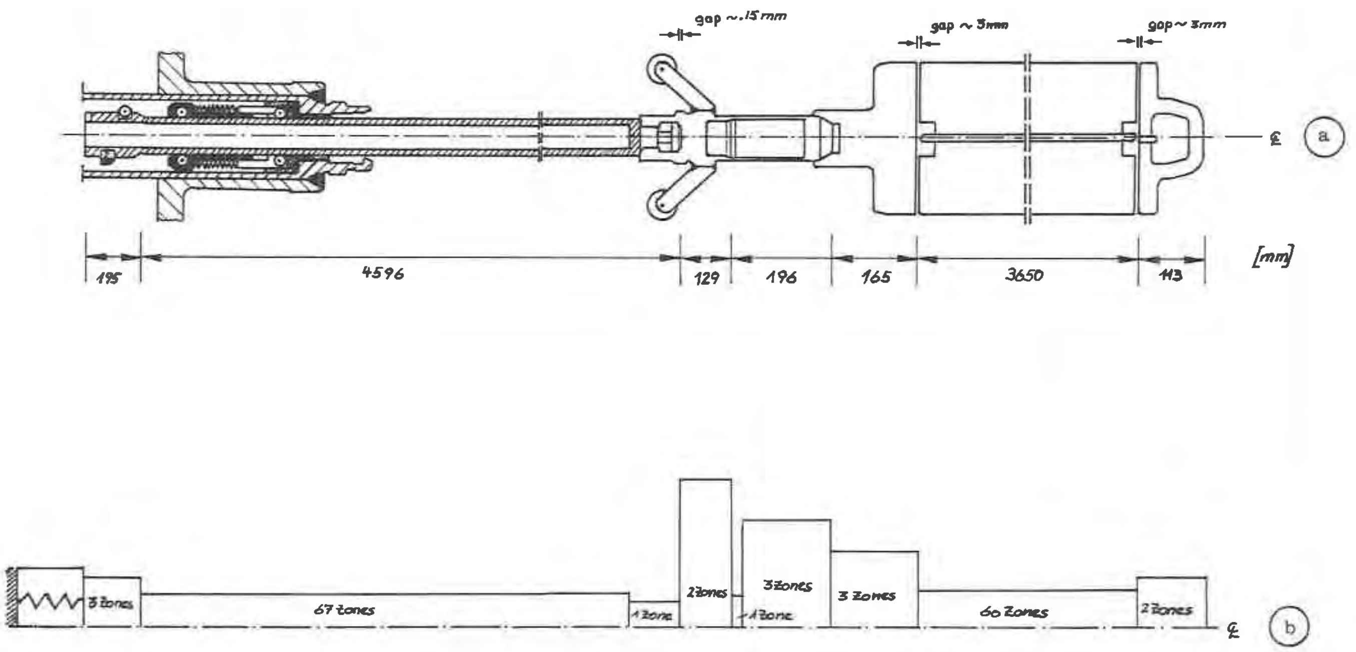

Fig. 2: Control rod   a) real geometry
b) scheme of PISCES - model

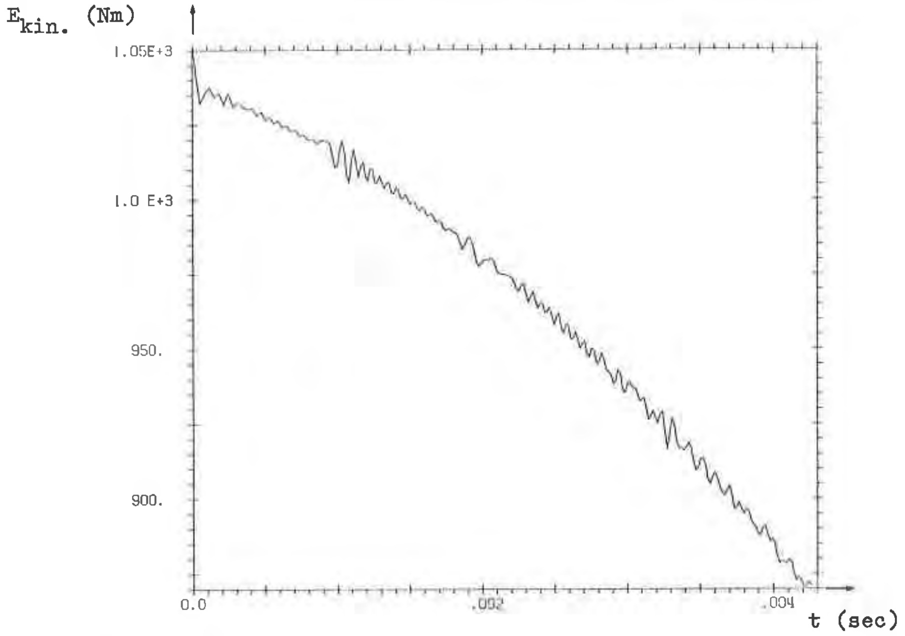

Fig. 3: Time History of total kinetic Energy of the control rod

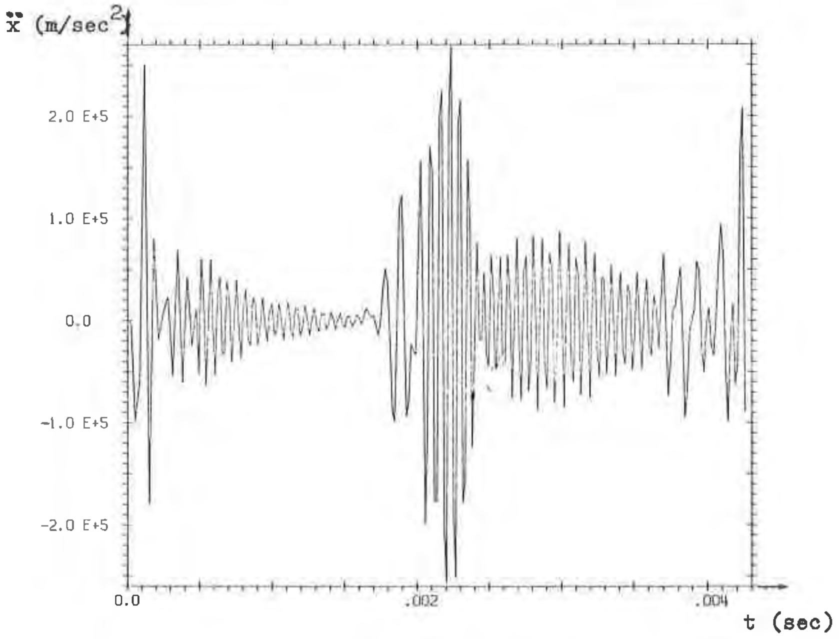

Fig. 4: Time History of local acceleration of the throttle bush

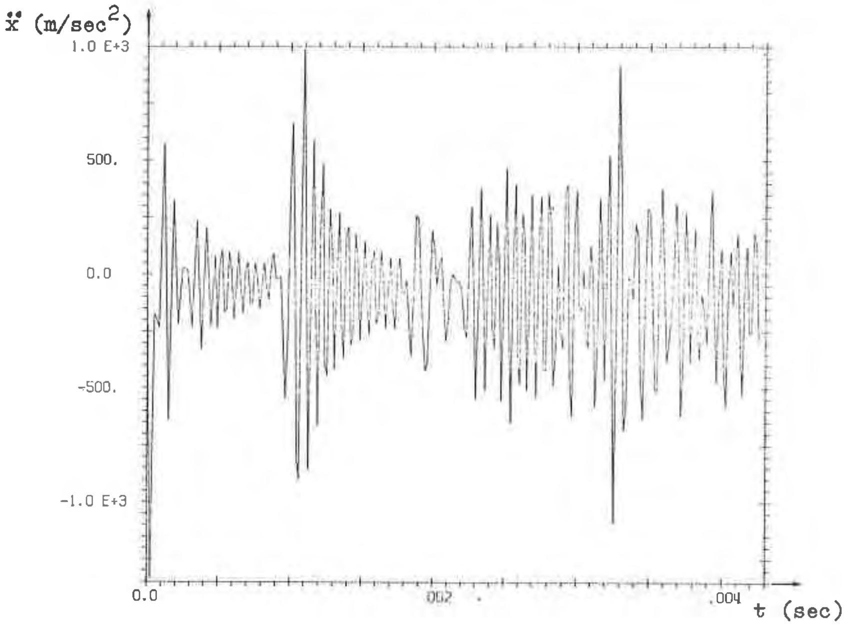

Fig. 5: Time History of mean acceleration of the control rod

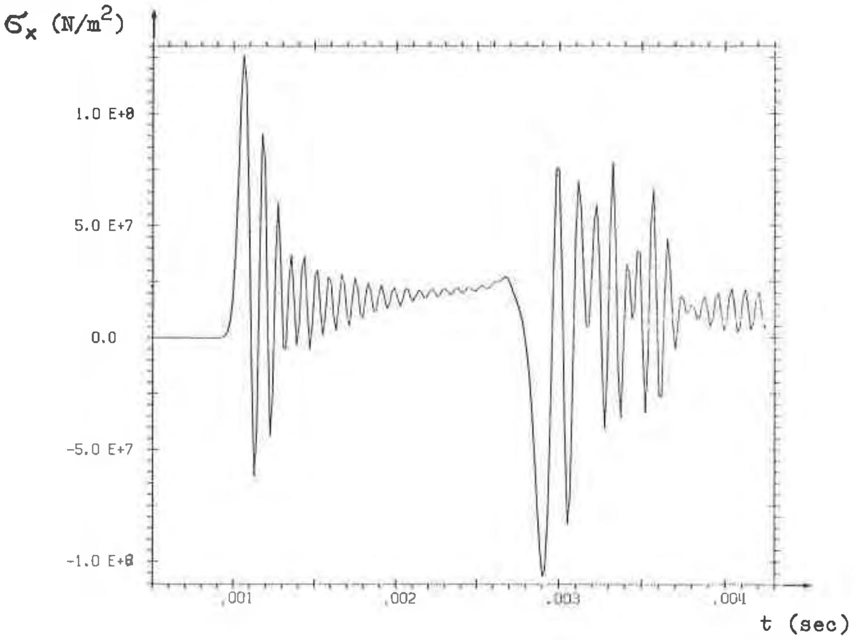

Fig. 6: Time History of the stress in the lower connecting section