Thermal-Hydraulic and Structural Analysis of Steam Isolation Valves

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Abstract

In BWR power plants a reliable and quick function of steam isolation valves is necessary for safety reasons. The reliability of the valves can be investigated by performing full scale tests and by using suitable analytical methods. The purpose of the present paper is to describe the analytical methods that have been developed for thermal hydraulic and structural analysis of the valve. These methods have been developed as part of a Finnish-Swedish collaboration project called IVLSP (Isolation Valve Licensing Support Project). The work resulted in a package of computer codes, which can be used together with small-scale experiments for the analysis of valve function. It is felt that the use of the here implemented computer codes is a relatively accurate but still cheap method for the valve analysis.

1. Introduction

The main steam lines of a boiling water reactor are for safety reasons provided with valves, which are designed to close quickly, if isolation of the containment is called for. These valves have to stand many normal closures during the lifetime of the reactor. After perhaps twenty years of operation they still have to stand a closure caused by a possible steam line break.

The functioning of the valves can be investigated by performing full scale experiments or by using a suitable calculational procedure. Full scale experiments are expensive and only a limited number of tests can be performed. In any case a fatigue analysis must be performed in order to ensure the reliability of the valves during the whole reactor lifetime.

The purpose of the present paper is to describe an alternative to full scale testing. The procedure consists of theoretical thermal-hydraulic and structural analyses for the valve (see Fig. 1). Thermal-hydraulic analysis was carried out using the TMOC computer code [1]. The TMOC code has a computational model for the valve function. This model utilizes experimental data, which are obtained from scale model experiments. From the thermal-hydraulic analysis disc velocity (u_{DISC}) and pressure history in the steam line are obtained. These values are then used in structural analysis.

One way to perform the structural analysis is to use advanced computer codes such as ADINA [2]. However, to reduce computer costs a simple but accurate enough procedure was used. The dynamical structural analysis was performed using a lumped parametric model LUMA. The stiffness matrix used in LUMA was calculated by means of a finite element code AXIFEM [3]. LUMA calculates the stress and strain histories used in the fatigue analysis.

2. Thermal-hydraulic Analysis

2.1 Computer Code

The thermal-hydraulic analysis has been performed using the TMOC computer code [1]. In order to describe the steam flow properly, the code was extended to simulate also superheated steam, i.e. a 5-equation model was employed for hydraulics. TMOC utilizes an explicit method of characteristics for the solution of hydraulic equations. This method is advantageous in fast transients and in treatment of boundary conditions.

The isolation valve (see Fig. 2) is described as a boundary between two pipes. At the boundary the total number of unknown variables is ten. Hence, in addition to five equations along the characteristics, five boundary conditions must be specified. Furthermore, the valve location is described as a contraction, for which five additional relations are needed.

Let us assume that the last node (node N) of pipe j is connected to the first node (node 1) of pipe j+1. In addition, the valve location is node N+1. The spatial distance between the points N, N+1 and 1 is assumed to be equal to zero. The following boundary conditions are used:

1) conservation of mass

$$A_N \rho_N u_N = C_D A_{N+1} \rho_{N+1} u_{N+1} = A_1 \rho_1 u_1 . \tag{1}$$

Above A is flow area, ρ is density, u is velocity and C_D is contraction coefficient.

2) static pressure loss

$$p_N + \frac{1}{2} \rho_N u_N{}^2 = p_{N+1} + \frac{1}{2} \rho_{N+1} u_{N+1}{}^2 =$$

$$p_1 + \frac{1}{2} \rho_1 u_1{}^2 + \frac{1}{2} K \rho_1 u_1 \; u_1 \tag{2}$$

where K is pressure loss coefficient for the valve.

3) change in void fraction

$$\alpha_N = \alpha_{N+1} + \Delta\alpha_{N+1} = \alpha_1 + \Delta\alpha_1 \tag{3}$$

4) conservation of energy

$$A_N u_N (\rho h_N + \frac{1}{2} \rho_N u_N{}^2) = C_D A_{N+1} u_{N+1} (\rho h_{N+1} + \frac{1}{2} \rho_{N+1} u_{N+1}{}^2) =$$

$$A_1 u_1 (\rho h_1 + \frac{1}{2} \rho_1 u_1{}^2) \tag{4}$$

Above $\rho h = \alpha \rho_g h_g + (1-\alpha) \rho_\ell h_\ell$, h_g and h_ℓ are gas and liquid enthalpies respectively. The fifth boundary condition is obtained by assuming that the least massive phase follows the saturation line.

In order to solve eqs. (1)-(4) the contraction coefficient C_D and the pressure loss coefficient K are specified as functions of the Mach number and the dimensionless valve position. The position of the valve disc, x, is calculated from

$$\ddot{x} = \frac{F_M + F_P + F_D}{m} \tag{5}$$

where F_M is the magnetic force, F_P is the force acting on the valve piston, F_D is the force acting on the valve disc, and m is the mass of the moving parts. F_M is known on the basis of valve specifications and F_P can be calculated using the chamber pressures. F_D is obtained using the calculated flow variables. A convenient way to calculate F_D is

$$F_D = -k(x,Ma) A_D (p_{up} - p_{down}) + A_R p_{up} \tag{6}$$

Above k(x,Ma) is a force coefficient, A_D and A_R are areas of the valve disc and valve rod respectively, p_{up} is total pressure upstream from the valve, and p_{down} is static pressure downstream from the valve.

The chamber pressures are obtained by solving the mass and energy conservation equations. For a chamber with volume V the mass conservation equation is

$$\frac{d\rho V}{dt} + \sum_k W_k = 0 \qquad (7)$$

where W_k is mass flow in a connecting junction. The energy conservation equation is

$$\frac{d\rho eV}{dt} + \sum_k W_k(h_k + \frac{u_k^2}{2}) + p\,\frac{dV}{dt} = 0 \qquad (8)$$

where e is specific internal energy.

The mass flows in the junctions are in general obtained from the solution of hydraulic equations. It is, however, convenient to model short orifices using only a single momentum equation as

$$\frac{dW}{dt} = \frac{A}{L}\,(p_u - p_d) - \frac{AK}{2L}\,\rho u\,u \qquad (9)$$

where A is the smallest flow area, L is length of the junction, p_d and p_u are pressures at the downstream and upstream end of the junction and K is pressure loss coefficient.

The equations describing the valve dynamics (eqs. (5)-(9)) form a set of ordinary differential equations, which is solved by a two-step Runge-Kutta-method. As a result, the calculated position determines the pressure loss coefficient and the flow area.

2.2 Scale Model Tests

Scale model tests are utilized in order to obtain data for the coefficients C_D, K and k, which can be based on measurements where air is used instead of steam [4]. The coefficients are functions of the Mach number and the disc position. Because the Mach number is a function of p_{up}/p_{down}, k can be expressed as

$$k = k(x, p_{up}/p_{down}) \qquad (10)$$

Measurements can be performed at relatively low pressures, because k is a function of pressure ratio only.

For the present purpose the measurements were performed by the Laboratory of Steam- and Gasdynamics at the Helsinki University of Technology [4]. The forces acting on the valve disc and valve body, chamber pressures and the pressure difference across the valve were measured.

The model was made in the scale 1:2.15 (see Fig. 3). The model had a force-measuring equipment on the spindle. In addition the force was obtained by integrating the pressure profile on the valve disc. The tests were performed with six valve strokes and five pressure ratios. The lowest pressure ratio corresponded to the nominal flow and the highest one to the critical flow. An example of the measured force coefficient is given in Fig. 4.

2.3 Results

The computational model has been tested by calculating test cases [5]. During normal operation the isolation valve studied is closed by evacuating the lower chamber. At full power the velocity in the steam line was assumed to be 40 m/s. The calculated position of the valve disc is shown in Fig. 5 and the pressure histories in the steam line in Fig. 6. The impact is assumed to be elastic, which causes the oscillatory motion before the valve is finally closed.

3. Structural Mechanics

In the field of structural mechanics the analysis is divided into two stages: The first stage includes a static FEM-analysis or experiment to get the force displacement curve for the second stage, which utilizes a one-dimensional lumped mass model called LUMA. LUMA calculates the dynamical structural analysis of the valve. It is possible to perform a dynamical nonlinear structural analysis by using advanced computer codes. However, that is often expensive and nonpractical especially for fatigue calculations. Therefore the FEM-LUMA procedure has been developed.

The modelling of the structure for LUMA is shown in Fig. 7. Each mass m_i is connected with spring k_i to the mass m_{i-1} and with spring k_{i+1} to the mass m_{i+1}. The values of m_i are easily found according to Fig. 7. The equation of motion for a mass point i in the structure is

$$m_i\ddot{x}_i+k_i(x_i-x_{i-1})-k_{i+1}(x_{i+1}-x_i)+c_i(\dot{x}_i-\dot{x}_{i-1})-c_{i+1}(\dot{x}_{i+1}-\dot{x}_i)-P_e^i = 0, \qquad (11)$$

where c_i is a damping coefficient and proportional to the logarithmic degrement of the material. P_e^i is an external force due to friction and pressure. For the structure the following equation is obtained

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{P\}. \qquad (12)$$

The stiffness terms are found by a static FEM-analysis using the AXIFEM computer code [3] or experimentally. In the present case the analysis has been the only possible way to solve the problem. The force F_j (see Fig. 8a) acting on the mass j causes in linear case a deflection $x_{i,j}$ at the point i

$$x_{i,j} = a_{i,j}F_j \qquad (13)$$

The total displacement is then

$$\sum_{j=1}^{n} x_{i,j} = x_i = \sum_{j=1}^{n} a_{i,j}F_j, \quad i = 1,n \qquad (14)$$

or

$$\{x\} = [a]\{F\}. \tag{15}$$

The stiffness matrix $[K]$ is the inverse of $[a]$,

$$[K] = [a]^{-1}. \tag{16}$$

The way to calculate the $[K]$ is valid only before yielding. Hence, an approximate way to form the nonlinear $[K]$-matrix is used: Calculate a nonlinear static analysis with load F_i (Fig. 8b), which gives a force-displacement curve for spring between the points i and i-1 as

$$F_i = F_i(x_i - x_{i-1}) \tag{17}$$

and finally the stiffness k_i equal to the slope of F_i.

The force-displacement curve is approximated by using a piecewise linear curve, Fig. 8c. This is used during the first loading up to the point A. Unloading occurs through the path A-B, using the original stiffness before yielding. The nonyielded stiffness is used when $F_i <$ F(A). After this the stiffness defined by the path A-C is used.

The solution of eq. (12) takes place in time increments with an assumption that the acceleration $\{\ddot{x}\}$ varies linearly during the increment. The calculation begins at the moment of the impact.

One way to check the LUMA procedure is to compare the results with the ADINA calculations. According to the comparisons performed the results of both methods agree fairly well.

The error caused by forming the stiffness matrix with the simplified way described above can also be estimated by calculating the transverse eigenfrequencies of a simply supported beam. An exact value of the lowest eigenfrequency is $\omega = \pi^2\sqrt{EI/m\ell^3}$, where m is the total mass, E is Youngs' modulus, ℓ is the length of the beam and I is the moment of inertia. The data m = 10 kg, ℓ = 1 m and EI = 10000 Nm^2 gives ω = 312 s^{-1}. The eigenfrequency with lumped masses is found from equation

$$[K] - \omega^2[m] = \{0\} \tag{18}$$

Using five lumped masses eq. (18) gives ω = 332 s^{-1}. The error is about 10 %, which is moderate thinking the way of forming the stiffness matrix. The high frequencies are more erroneous, a fact that may have some effect on the results. The high frequencies have some influence on fatigue, but only during a short interval after an impact, because they vanish quickly. Also the correct way to form the stiffness matrix, eq. (16), may not be essentially better, because the high frequencies based on lumped stiffness matrix are always erroneous. At the pipebreak the deformations are mostly plastic and therefore the high frequencies do not have an important role. In Fig. 9 the development of the plasticity in a steam isolation valve of a boiling water reactor with three different closing velocities is shown. The velocity 25 m/s corresponds to the valve closure during a pipebreak near the valve.

4. Conclusions

An analytical procedure for testing of steam isolation valves has been developed. The following computer codes are utilized in the analysis

- TMOC for the hydraulic analysis of the steam line and for the analysis of valve dynamics
- AXIFEM for the calculation of the nonlinear static structural behaviour of the valve disc and seat
- LUMA for the final structural analysis connected with the valve dynamics.

Together with the measured data obtained from small scale tests, the present procedure is felt to be a relatively accurate tool in the safety analysis of isolation valves.

References

[1] SIIKONEN, T., "Computer Program TMOC for Calculating of Pressure Transients in the Fluid Filled Piping Networks", ENS/ANS Topical Meeting on Nuclear Power Reactor Safety, Brussels, 1978.

[2] BATHE, K.-J., "ADINA, A Finite Element Program for Automatic Dynamic Incremental Nonlinear Analysis", Massachusetts Institute of Technology, Cambridge, Massachusetts 1977.

[3] IKONEN, K., "Computer Program AXIFEM for Nonlinear Static Structural Analysis of an Axisymmetric Body", Technical Research Centre of Finland, Nuclear Engineering Laboratory, Helsinki 1978.

[4] WALLEN, G., "Model Test of an Isolation Valve", Laboratory of Steam- and Gasdynamics, Helsinki University of Technology, Report No. 1, 1980.

[5] SIIKONEN, T., "Computational Method for the Analysis of Valve Transients", manuscript submitted for J. of Pressure Vessel Technology.

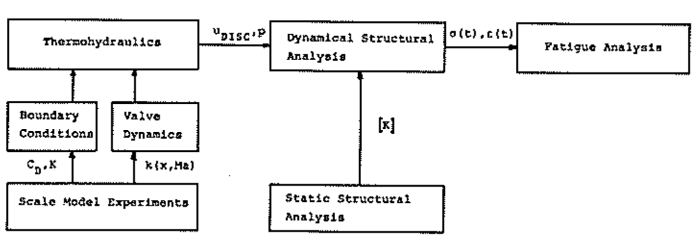

Fig. 1. Test procedure.

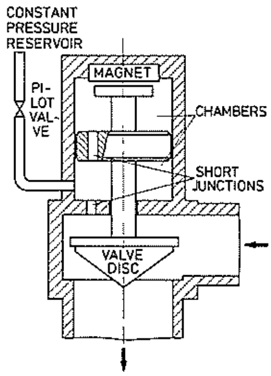

Fig. 2. Isolation valve.

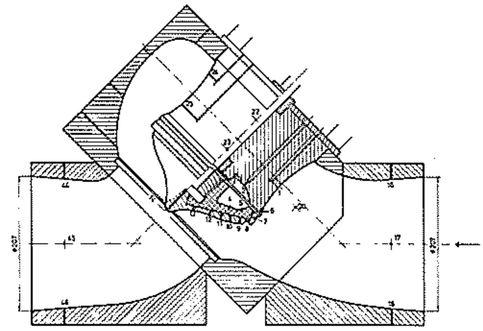

Fig. 3. Scale model of the isolation valve. Numbers indicate the positions of pressure measurements.

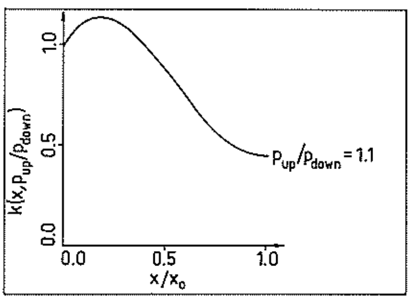

Fig. 4. Force coefficient as a func tion of dimensionless position.

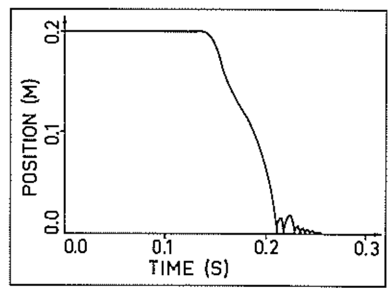

Fig. 5. Position of the valve disc.

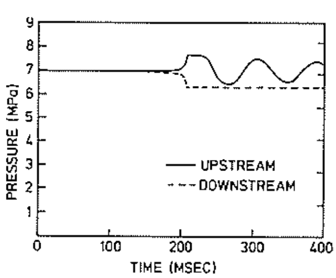

Fig. 6. Pressure history at the steam line.

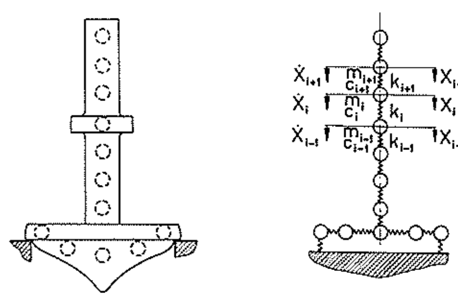

Fig. 7. Lumped mass model.

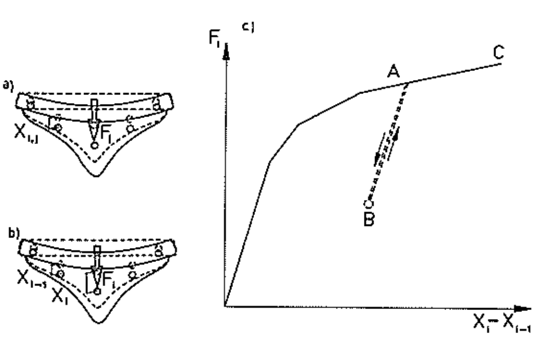

Fig. 8. Derivation of force-displacement curve.

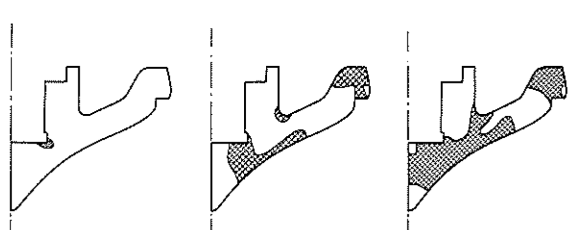

Fig. 9. Development of the plasticity.