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CHECKING OF SPRING LOADED SAFETY VALVES IN GERMAN NUCLEAR POWER PLANTS

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SUMMARY

Events of safety valve failures in German Nuclear Power Plants forced the utilities to check affected safety valves in their plants. Causes of failures were found to be fluiddynamic interactions during blowoff and inadequate piping support. ABB has developed evaluation criteria for judging valve operation and generated structural loads. By means of these criteria the evaluation task and necessary backfitting can be reduced to a minimum.

INTRODUCTION

Single failures of spring loaded safety valves in fluid systems were identified during the last years in German Nuclear Power Plants. Inspections showed that the damages occurred primarily in the feed pipe to the valves and were caused by unstable operation of the valves.

This situation forced German Utilities to perform a preventive examination of affected valves in their plants.

Bayernwerk AG and other utilities along with ABB decided to develop a program to establish criteria for the identification of the single endangered valve, to minimize backfitting and impact on plant availability as well as on in-service inspections. The examination method is primarily based on calculations and analyses, which offer common criteria for stable valve operation and inoffensive stress behaviour for safety valves of different suppliers with different design.

CAUSES FOR DAMAGES

Causes of failure in particular could be:

- Pressure drop in the valve feed pipe:
  if the ratio of pipe length to nominal diameter exceeds a certain value, then the resulting pressure drop will be higher than acceptable, and will result in an unstable opening and closing of the valve.

- Interaction of reflected pressure waves with the spring-mass-system of the valve: if the feed pipe to the valve is too long, resonant vibrations can be induced in a frequency range up to 30 Hz.

- Inadequate process engineering:
  oversized design of e.g. full-lift safety valves can result in valve motions in a frequency range of approximately 5 Hz.

- Inadequate supports of piping:
  if the piping is not supported adequately, a blow off operation of the valve can result in unacceptable stresses, causing damages to the pipe.

EXAMINATION METHODS

To establish consolidated evaluation methods, a series of fluid and structural dynamic analytical tests were performed on a number of representative full-lift and proportional safety valves of different suppliers and for a feedpipe diameter range of DN 15 to DN 100. For all tests, it was taken for granted that in respect to an effective pressure limit, the pressure rise velocity in the system will not exceed 20% of the set pressure per sec.

The fluid dynamic analysis was performed using the code INROS, developed by ABB. This code was designed for solutions of transient pipe flow problems. Typical applications are determination of pressure waves and structural loads caused by valve actuations, pump trips or pipe rupture.

For the actual checking procedure of safety valve setting characteristics the code was supplemented with an exact mechanical valve model which permits the calculation of the valve spindle lift x and fluid discharge as a function of governing forces expressed by the equation:

$$m_{Spl} \cdot \ddot{x} = + F_{Fl} \pm F_{Add} - F_{Spr} \pm F_{Fric} - F_W$$

The reaction forces on the valve spindle are shown in Fig. 1.

The fluid force ($F_{Fl} = A_{Seat} (P_I - P_0) \cdot a(x)$) is a function of the pressure difference between inlet and outlet and an amplification factor a(x) which must be evaluated experimentally. This amplification factor is strongly dependant on the seat geometry besides the lift and therefore differs significantly between the various types and suppliers of valves.

Potential additional forces (F_{Add}) caused i. e. by sealing foils or by installed damping devices can be taken into account by actual specification.

For the given type of spring the spring force (F_{Spr}) is a linear function of the lift x. The mechanical friction force (F_{Fric}) may be up to 4% of the spring force.

Finally, the weight of spindle ($F_W = m_{Spl} \cdot g$), comprising all moving parts and 50% of the spring mass, must be included.

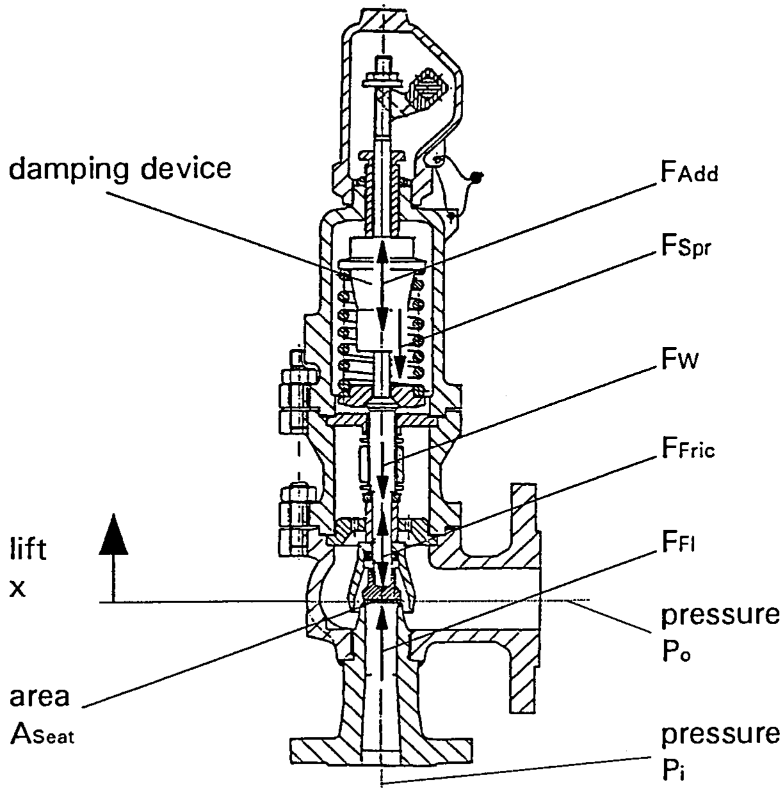


Fig.1: Valve model with acting forces

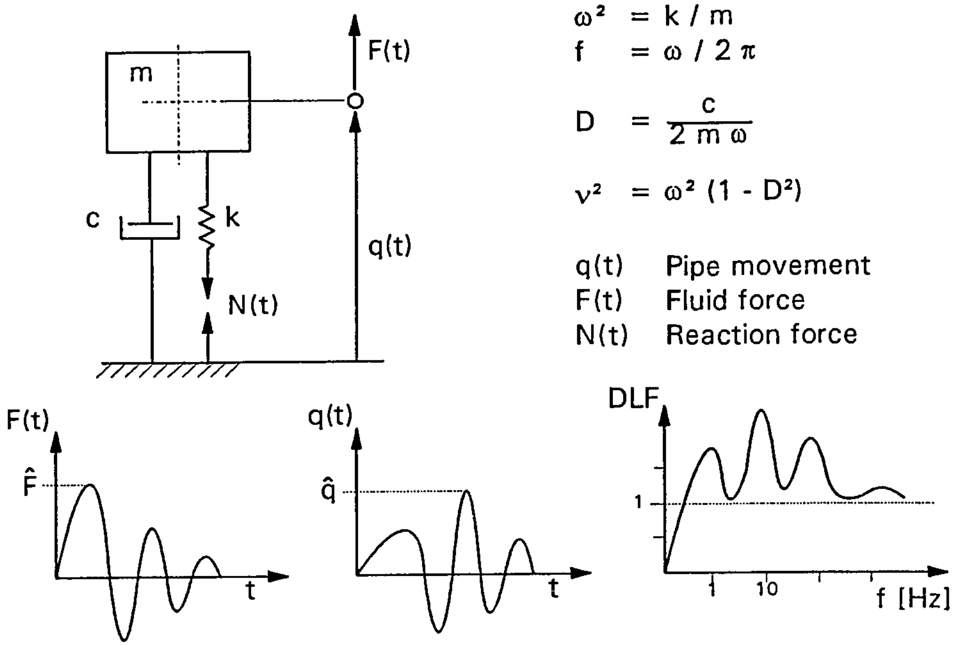
The fundamental results are process data (pressure, temperature, massflow) and, concerning structural analysis, the time history of fluid forces and corresponding Dynamic Load Factors (DLF) for predefined pipe sections.
As shown by Fig. 2 the DLF is defined as the ratio of the (maximum) reaction force of a one degree-of-freedom oscillating mass to the (maximum) excitation fluid force. Hence the calculated DLF-spectra represents the distribution of frequency content versus a specified range of frequency for the resulting fluid force. From the reference calculations the DLF turned out to have a maximum value of approx. 5,0 depending on feed pipe length and valve setting characteristics.

In addition to these fluid dynamics evaluations, structural dynamics calculations were performed with the finite element code ANSYS, using models of feed pipes with different lengths. The nominal diameter and support points of the pipes were also varied. Fluid loads and pressure pulsations, calculated with INROS for unstable operating valves, were applied as load spectra.

CRITERIA FOR STABLE OPERATION

A summary of the established evaluation criteria is given in Fig. 3. Test series with various parameters resulted in proven criteria for stable operation of the valves:

- Valves with set pressure $p_o \leq 20$ bar:
  stable, if feed pipe length doesn't exceed 0,9 m for DN < 50; 1,5 m for DN $\geq$ 50.



$$DLF = \frac{\hat{q}}{\hat{q}_{Stat}} = \frac{\hat{q}}{\hat{F} / k} = \frac{N(t)_{max}}{\hat{F}}$$

Fig.2: Definition of Dynamic Load Factor (DLF)

- Valves with set pressure $p_o > 20$ bar:
  stable under the same conditions as for $p_o \leq 20$ bar, if the maximum flow velocity c in the feed pipe doesn't exceed 3 m/sec.

CRITERIA FOR HARMLESS UNSTABLE OPERATION

As a result of the dynamic time response analysis, performed with the finite element program ANSYS, it could be proven that no damage will be caused, even during unstable operation of valves, if the following set pressures are not exceeded:

- $p_o \leq 10$ bar for feed pipe $\leq$ DN 50
- $p_o \leq 5$ bar for feed pipe > DN 50

Regarding allowable stresses, these criteria however are only met as far as significant bending loads from pipe displacements are not exceeded. So further stress criteria for pipe bending must be considered.

A typical feed pipe with fixed points at the main pipe and the safety valve is shown in Fig. 4. During blowoff pressure pulse generation, oscillating forces and roughly translatory displacements of the pipe will occur. Then a fatigue analysis for the pipe section at location A must be of interest.

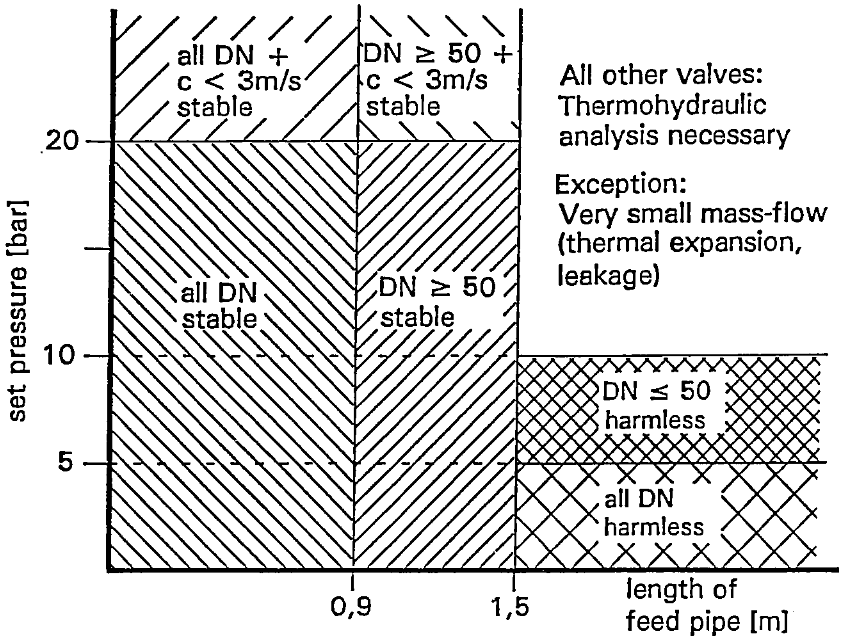


Fig.3: Criteria for stable valve operation and for harmless loads in case of unstable operation

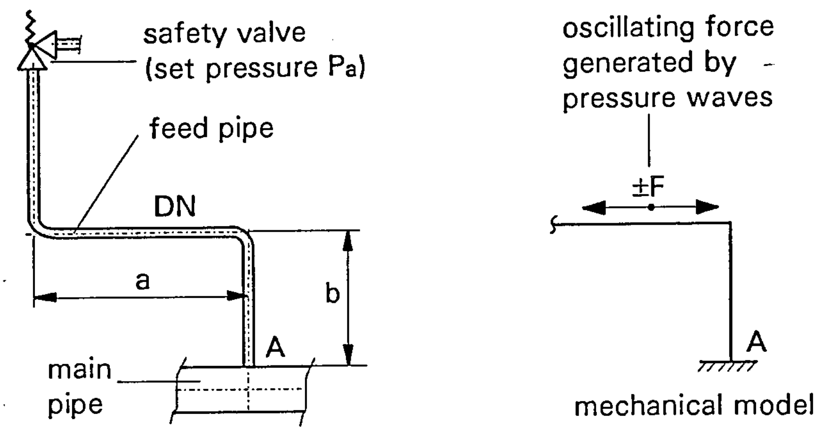
From fluid dynamic calculations and fatigue analysis based on the KTA-Regulatory-Guide for reference pipe and material data the stress of 160 N/mm^2 can be shown to be a reasonable upper limit of allowable stress values. This is in correspondence to 72000 cycles/hour (fatigue curve for ferrit).
Considering pressure pulse frequencies of 20 Hz as a rule, there is about one hour for detecting and stopping the valve blowoff. On this stress limit a criteria number $z = 3{,}8$ is derived for arbitrary pipe geometry and set pressures $\leq$ 10bar.
That means that in all cases $z \geq 3{,}8$ the stress limit for fatigue is met. Otherwise, for cases $z > 3{,}8$ the pipe geometry must be changed or backfitting with an additional support has to be done.
This criteria number is conservative for austenitic material with enhanced fatigue capability.

CONCLUSION

When all input data are available, the criteria as given in Fig. 3 and Fig. 4 will be applied to the single valve. No further activities are necessary for those valves which comply with the criteria for stable operation and harmless unstable operation.

For all other valves a specific individual fluiddynamic analysis has to be performed, except for those valves with small blow-down, e.g. for the load "transient thermal expansion with chambered volume", or valves for providing safety against small leakage from connected systems with higher pressure.

By means of the derived criteria the evaluation task can be reduced considerably. For all affected valves in a plant only approx. 20 analytical evaluations have to be performed, but resulting only in isolated cases with a real backfitting necessity.



evaluation base:

- fatigue analysis cross section A (KTA 3211.2, 72000 cycles, $F_{max}=600$ N, DLF=5)
- stress result only by bending, other loads can be neglected for $P_a \leq 10$ bar
- stress evaluation
  - force $F \sim a \cdot DN^2$
  - moment $M \sim F \cdot b$
  - inertia moment of resistance $W \sim DN^{2,7}$
  - stress $\sigma = M / W$
- criteria for stress limit (Reference material: St 35.8)

$$\frac{a \cdot b \cdot P_a}{DN^{0,7}} = z > 3,8$$

a, b: [m]
DN: [mm]
Pa: [bar]

Fig.4: Check criteria for typical feed pipes in case of unstable valve operation on base of fatigue analysis for set pressures $P_a \leq 10$ bar

Backfitting activities include:
- Change of length, nominal diameter or pipe routing for feed pipes and blow-off pipes
- Improvements for the support system of pipes and valves
- Change of the lift limit of the valves
- Exchange of valves

and if possible
- Installation of damping systems to change the valve friction factor characteristics

In any case, all resulting backfitting activities are adjusted to the different valve designs, and supported by analytical evidence.