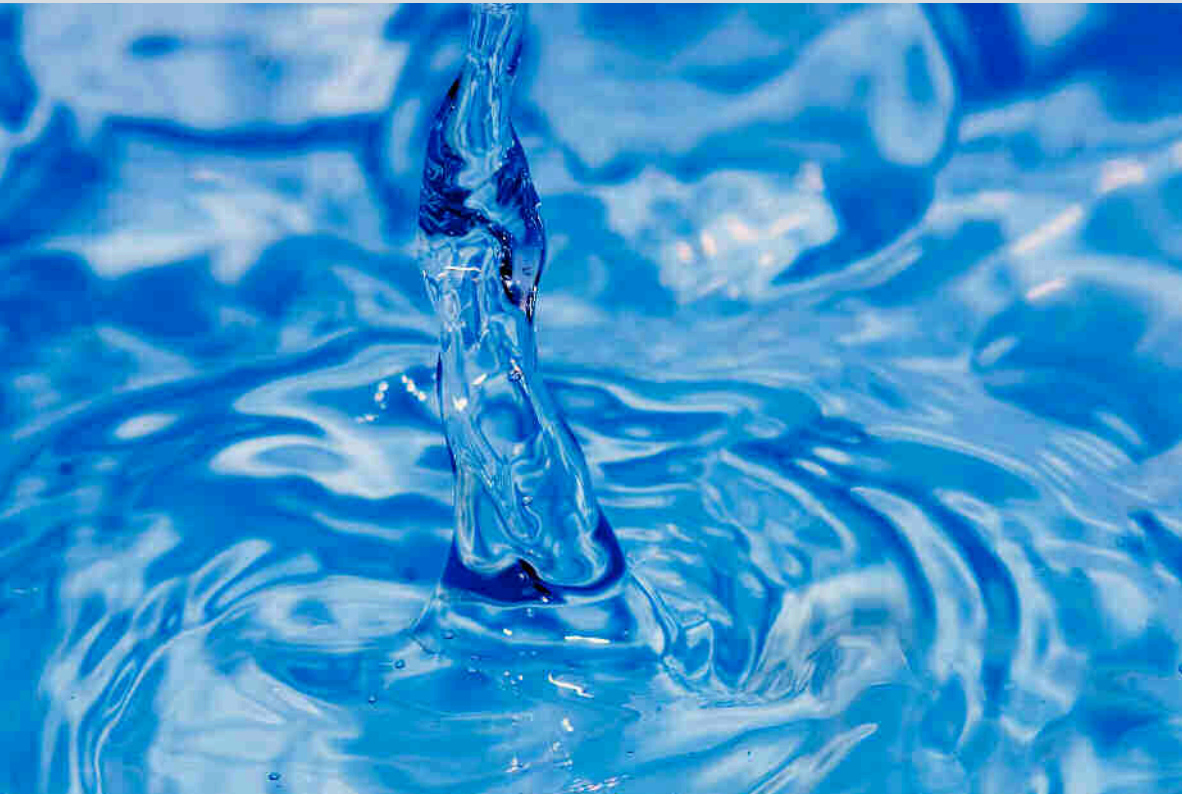


SPECIAL PRINT

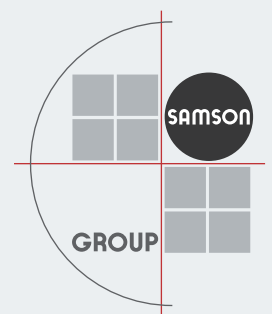
Predicting Cavitation Damage in Control Valves



By:
Dr.-Ing. Jörg Kiesbauer
Dipl.-Ing. Domagoj Vnucec
Dipl.-Ing. Miriam Roth
Prof. Dr. Bernd Stoffel



Special Print from:
Hydrocarbon Processing
March 2006



Predicting Cavitation Damage in Control Valves

Dr. Jörg Kiesbauer, and Dipl.-Ing. Domagoj Vnucec, SAMSON AG
Dipl.-Ing. Miriam Roth, Prof. Dr. Bernd Stoffel, Darmstadt University of Technology

Cavitation can occur in control valves handling fluids, causing loud noise as well as damaging valve components and ultimately leading to additional costs in process plants. A standardized procedure to evaluate the destructiveness of cavitation-induced still does not exist whereas noise emission can reliably be predicted with the new international EN 60534-8-4 standard. This article describes a new method to solve this problem by evaluating structure-borne noise in the ultrasonic range.

1. Cavitation in control valves

The pressure of the process medium changes as it passes through the valve. Key pressures related to the inner flow path include the input pressure p_1 , the pressure p_{vc} at the narrowest point (vena contracta) and the output pressure p_2 .

In other zones around this main flow path, the static pressure can, however, be much higher or lower than p_{vc} . A flow simulation of the pressure field at the vena contracta performed with CFD (Computational Fluid Dynamics) illustrates this behavior (Fig. 1). At a differential pressure of 3.5 bar, the pressure field is stable, likewise at 4.0 bar, except for a small zone at the plug (arrow in the picture at the bottom) where the pressure is clearly lower. Small vapor bubbles start to form when the pressure in this zone reaches the vapor pressure p_v . The medium flow carries these bubbles along downstream to the area of flow exhibiting a higher pressure. At this point, the bubbles implode. This process is defined as cavitation.

Fig. 2 contains four images taken at various differential pressures across the valve. The x_{Fz} coefficient plays a key role in this case, being the differential pressure ratio for incipient cavitation [1]. Together with the input pressure and vapor pressure, it defines the differential pressure for incipient cavitation, that is $x_{Fz} \cdot (p_1 - p_v)$. The first streaks of steam are visible in Fig. 2b just behind the plug on the left-hand side. As the differential pressure increases, the cavitation zone spreads out because the pressure level in this area drops as a result of the increasing flow velocities.

One consequence of this process is noise. Fig. 3 shows a typical graph plotting noise versus the differential pressure ratio $x_f = (p_1 - p_2) / (p_1 - p_v)$. The sharp rise in noise level is a typical indication for the onset of cavitation. The x_{Fz} coefficient can be recorded at this point. A cavitating flow arises between x_{Fz} and 1. When $x_f < x_{Fz}$ the flow is merely turbulent or laminar. When $x_f > 1$, no more vapor bubbles implode and the existing vapor

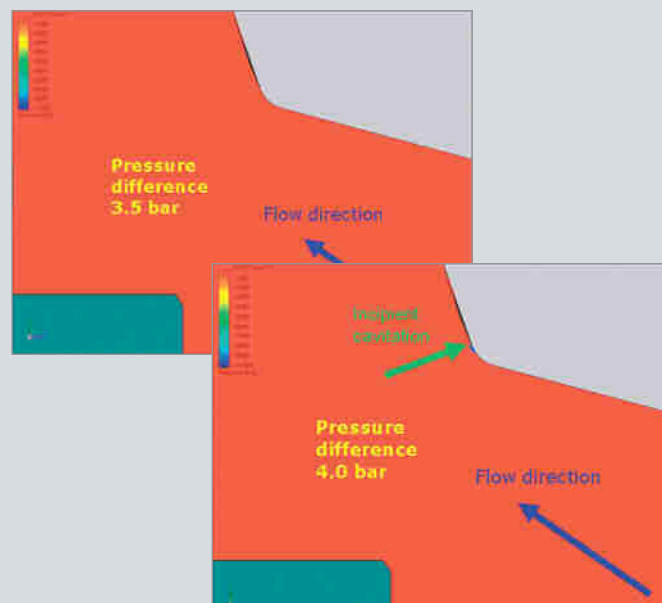


Fig. 1: Flow simulation (CFD) between seat and plug for $p_1 - p_2 = 3.5$ bar and 4.0 bar



Fig. 2: Cavitation process in a control valve as a function of a pressure drop between 5.5 and 9 bar ($p_1 = 10$ bar, $p_v = 0.02$ bar, water)

mixes with the fluid creating a two-phase flow which continues right into the outlet pipe as the pressure p_2 is lower than the vapor pressure at this point.

Noise prediction is to be covered by a new theory-based method laid down in the EN 60534-8-4 standard, which will achieve much more precise results than previous methods [1], [6]. This method estimates x_{Fz} , yet the most precise results are achieved when the coefficient is determined using measured data as in Fig. 4 [1].

The x_{Fz} coefficient should be as high as possible over the whole opening range of the valve to achieve the lowest possible noise level to ensure that no cavitation or just weak cavitation occurs.

However, if the differential pressure ratio x_F is much greater than x_{Fz} , the question arises whether the cavitation can damage valve components. Unfortunately, there are no standardized methods covering this problem. Before discussing some recommended methods based on empirical data, the basic process involved when cavitation damages valve components is to be explained first.

The common theory is based on the microjet [2], which is formed when vapor bubbles implode. This microjet is minute in size, but is associated with a very high flow velocity. On impact against valve components, the microjet can cause plastic deformation of the component surface culminating in material fracturing. The implosion of a bubble depends on the pressure drop between the ambient pressure and the vapor pressure and, in the case of fast moving bubbles, also on the speed that the bubbles are carried along at. As a result, the higher the differential pressure $p_1 - p_v$ is, the stronger the implosions at the same differential pressure ratio x_F are.

According to recent findings [2], [3], [4], the pressure wave that causes the vapor bubbles to implode plays an even greater role.

The rate of material erosion can increase noticeably in the event of additional corrosion.

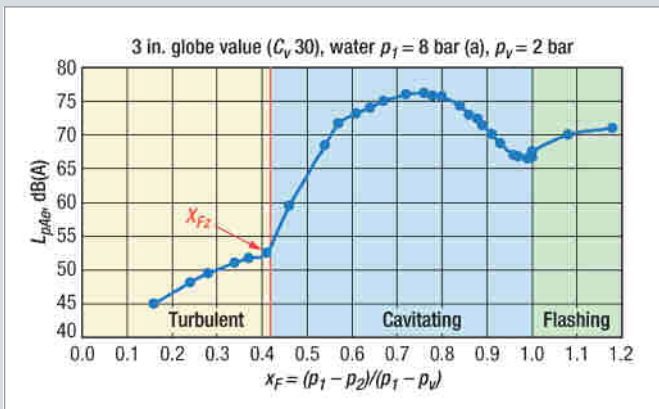


Fig. 3: Typical graph plotting noise vs. differential pressure ratio x_F

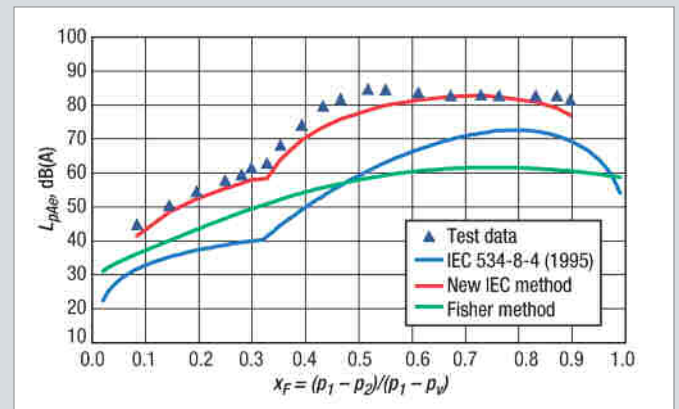


Fig. 4: Improved noise prediction using new IEC method [6]

2. Evaluating cavitation based on empirical data

ISA RP 75.23 (Considerations for Evaluating Control Valve Cavitation) [2] describes the major cavitation phenomena and also defines prediction methods based on calculations. The cavitation index σ is used in place of the differential pressure ratio x_F as the reciprocal. Fig. 5 depicts the same noise level as Fig. 3 plotted against σ . In [2], various distinctive valve σ s were defined based on a similar graph with the structure-borne sound level at the outlet pipe (vibration measurements) instead of the sound pressure level as in Fig. 5:

- σ_i for incipient cavitation ($\approx 1/x_{Fz}$)
- σ_c for cavitation with a constant rise in noise level
- σ_{ch} for incipient choked flow ($\approx 1/F_L^3$, F_L = pressure recovery factor [5])
- σ_{mv} for maximum noise
- σ_{id} for the threshold of material damage

Two variables are determined directly by measuring the sound pressure and flow or are known from standards such as [5] and [6]. This is not readily the case for the other parameters, particularly σ_{id} , and requires numerous and complicated tests, such as erosion tests.

Additional conversion formula are listed, which allow a correction depending on $p_1 - p_v$ and the nominal valve size because the coefficients are mostly determined for smaller valve sizes and low pressures.

Appendix C of [2] specifies the intensity of cavitation numerically as the factor I relating to the condition where cavitation

starts to cause damage for the first time. This condition is identified by σ_{id} and the associated differential pressure $(p_1 - p_v)_{id}$. Without correcting the nominal valve size (SSE see [2]) and in relation to the differential pressure ratio, the following equation applies:

$$I = F_U F_T F_{DC} \left[\frac{1/x_{Fid} - 1}{1/x_F - 1} \right] \left[\frac{(p_1 - p_v)_{id}}{p_1 - p_v} \right]^a \quad (1)$$

x_{Fid} is the differential pressure ratio in the definition of σ_{id} where material destruction first occurs. The factors F take the following influences into account:

- F_U : Velocity factor based, for example, on outlet velocity (1 if smaller than the critical outlet velocity, e.g. 5 m/s)
- F_T : Temperature influence, 2 on average (Appendix C in [2])
- F_D : Influence of the application, e.g. continuous operation 2 (Appendix C in [2])

A maximum exponent $a = 0.11$ is specified for globe valves in [2]. Specific investigations [7] on a needle valve to examine material erosion caused by cavitation at pressures up to 200 bar formed the basis for the material erosion graph plotted versus differential pressure in Fig. 6. The use of the approximation formula specified in Fig. 6 for the material erosion rate to calculate I results in the following equation:

$$I = \left[\frac{1/x_F - 1}{1/x_{Fid} - 1} \right]^{-0.2} \left[\frac{x_F}{x_{Fid}} \right]^{3.9} \left[\frac{(p_1 - p_v)}{(p_1 - p_v)_{id}} \right]^{3.9} \quad (2)$$

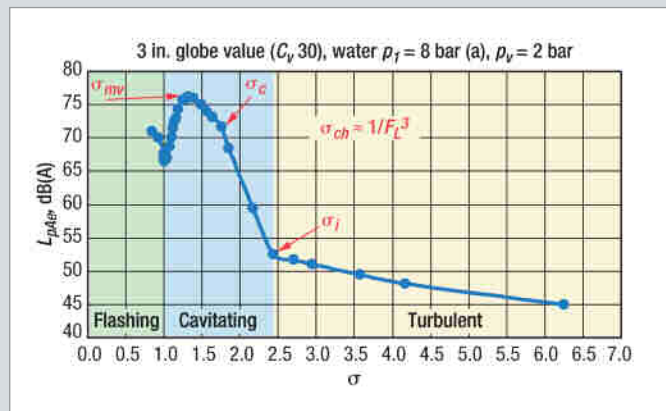


Fig. 5: Typical graph plotting noise vs. cavitation index σ according to ISA RP 75.23 [2]

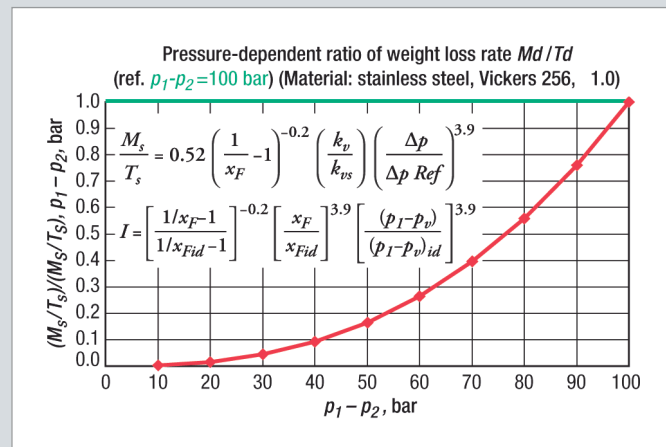


Fig. 6: Material erosion vs. differential pressure (needle valve) [7]

The large differences (Fig. 7) highlight the weakness of the empirical method.

A more general formula for I could be as follows:

$$I = \left[\frac{1/x_F - 1}{1/x_{Fid} - 1} \right]^a \left[\frac{x_F}{x_{Fid}} \right]^b \left[\frac{(p_1 - p_v)}{(p_1 - p_v)_{id}} \right]^c \quad (3)$$

In this case, the exponents a , b , and c are not defined at first and are to be determined using empirical data.

However, knowing x_{fid} is surely much more important than calculating the exact intensity in order to prevent material erosion in the first place.

Another investigation into cavitation erosion on parabolic plugs [8] can be found in valve engineering literature. This investigation and the experience gained by two of the authors of this article, who work for a valve manufacturer, have verified the publicized method to evaluate the risk of destructive cavitation erosion.

A sufficient amount of bubbles need to implode and the intensity of the pressure waves must be strong enough to cause damage worth mentioning. The pressure waves are governed considerably by the differential pressure $p_1 - p_v$. The onset of choked flow for $x_F > x_{fch}$ shows that high concentrations of bubbles exist. Measurements performed by the authors of this paper showed that the differential pressure ratio x_{fch} is approximately F_1^3 . Table 1 contains the x_{fid} coefficients for various valve types that are located just under x_{fch} . In the case of standard globe valves, damage becomes noticeable at differential pressures of 15 bar (Table 1, [10]).

Valve type	x_{Fid}	$\Delta p_{Cav,crit}$ [bar]
Single-stage globe valves	0.7	15
Single-stage globe valves, stellite or hardened plug/seat	0.7	25
Three-stage globe valves	1.0	100
Five-stage globe valves	1.0	200
Rotary plug valves	0.4	10
Butterfly valves and ball valves	0.25	5

Table 1: Coefficients for critical cavitation [10]

Together with the coefficients specified in Table 1, critical cavitation, i.e., the risk of cavitation erosion, exists when:

$$x_F > x_{Fid} \quad \text{and} \quad p_1 - p_2 > \Delta p_{Cav,crit} \quad (4)$$

Example: $x_F = 0.9$, $p_1 - p_2 = 20$ bar, use a globe valve with stellite/hardened material!

3. Acoustic measurements to determine cavitation severity in control valves

The recommendations in section 2, especially at the end of the section, are based on practical approaches, which have mostly been proven in practice. SAMSON AG in Frankfurt and the Turbomachinery and Fluid Power Institute at Darmstadt University of Technology carried out a joint research project to obtain further-reaching and more precise information in this field. The objective of the project was to develop a method of monitoring the destructiveness of erosion in valves in real time. The research involved the acoustic and material methods. This procedure has already been implemented successfully with pumps at the university [4].

The destructiveness of cavitation-induced erosion is not a physical variable that can be directly measured. To identify the negative effect that a cavitating flow has on a component, four parameters for the implosion event are necessary according to [3], [4]: frequency, amplitude (dynamic), pulse duration (kinematic), and distance to the wall (geometric). However, it is not

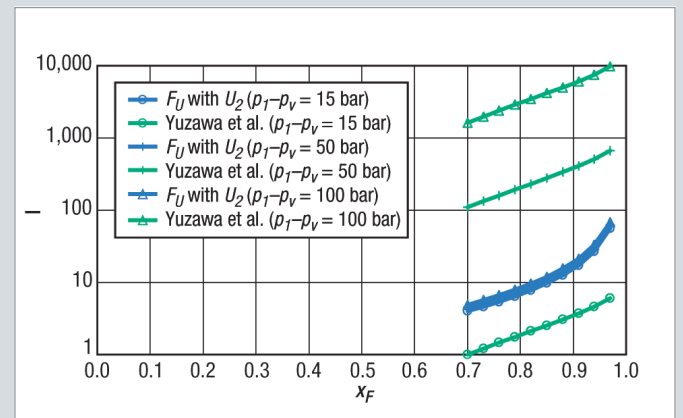


Fig. 7: Comparison of models showing intensity of cavitation at varying differential pressures ($x_{fid} = 0.7$), Yuzawa [7], ISA [2]

possible to determine all four parameters simultaneously, except in very special test rigs. Therefore, two different measuring methods were used in this research project, which provided four independent variables from which these four parameters can be derived. The variables are:

- The number and amplitudes of the acoustic events (acoustic method),
- The number of pits on the surface and their radii (material method).

The acoustic measurements of the cavitation events were performed on a control valve with different types of plugs (e.g. parabolic plug in Fig. 15). In this investigation, the acoustic sensor was directly attached to the plug, and as a result, it is assumed that the measured events are also involved in damaging the surface of the plug.

Histograms plotting the events and a map of the implosion capacity of each inlet pressure were created, although initially it was not possible to determine the threshold value at which the acoustic amplitudes start to cause damage. The material method was required to find this out.

The material damage was therefore analyzed at various operating points using valve plugs made of copper. To achieve this, the plugs were exposed to cavitation for a defined period of time at a certain operating point. Following this, the damaged surfaces were photographed under a microscope and analyzed by a special software (PITCOUNT) developed by Darmstadt University. This allows the number of pits and their radii to be counted.

A correlation of the acoustic and material-related histograms enables the portion of acoustic power to be determined that actually causes the material damage (threshold value required).

4. Acoustic method

The signals caused by cavitation have a special shape (see Fig. 8, signal above oscilloscope). A high-frequency noise at a constant amplitude is observed in the acoustic measurements when no cavitation exists. Much larger amplitudes that trail off fairly slowly arise when cavitation events (implosions) occur. The signal analysis counts the recorded amplitude peaks over a certain period of time and allocates them to amplitude classes (see Fig. 8 histogram on PC as well as Fig 10).

The implosion pressure waves caused by the bubbles are considered to be similar to spherical waves in an approximation. The sound energy E_s of the spherical wave results from the force amplitude \hat{F} exerted on the wall, and the pulse width τ_p :

$$E_s = \frac{\hat{F}^2}{\pi \rho c^3 \tau_p} \tag{5}$$

The transmission performance between the measured structure-borne noise and the force was determined by calibration measurements to allow E_s to be calculated using the structure-borne noise signals. The pulse width τ_p is determined by the correlation with the material-related test results.

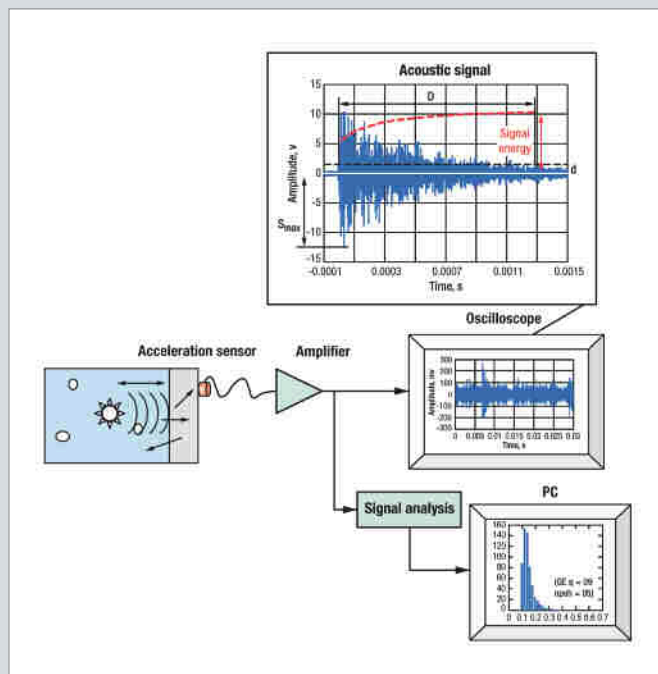


Fig. 8: Signal processing in acoustic measuring method

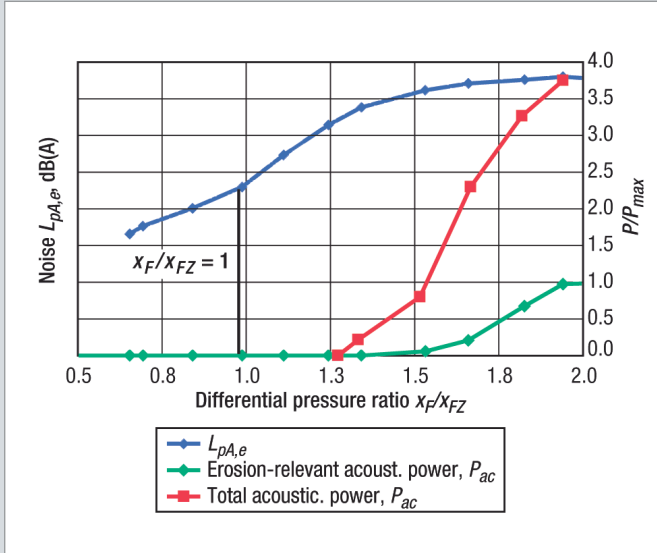


Fig. 9: Comparison of external sound pressure level, entire acoustic power, and damage-relevant analyzed acoustic power for 100 % load (parabolic plug, see Fig. 15)

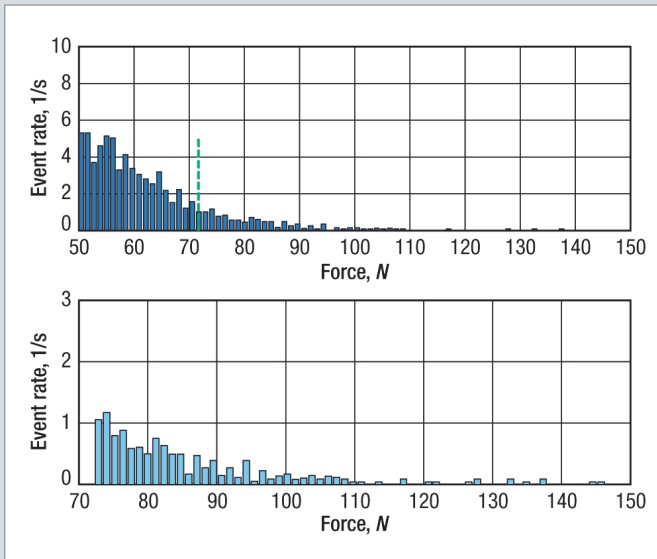


Fig. 10: Acoustic event histogram (without and with 73 N threshold)

5. Material method

The material analysis is generally performed in three steps:

- One specimen each is exposed to a different cavitation operating point
- Photographing the damaged surfaces (Fig. 11)
- Analysis of the photographs (Figs. 12 and 13)

A benefit of this method, in which just the initial stage is investigated, is the relatively short testing time. The time until a surface is damaged is substantially reduced by selecting materials with a fairly low resistance against cavitation. Additionally, a relatively short time helps obtain more constant test parameters. The plugs are made of copper, which is soft and provides a uniform image in undamaged, polished condition in the image analysis procedure.

The images taken under a microscope are analyzed using the PITCOUNT software developed by the Institute (Fig. 12).

The pit count method allows cavitation-induced erosion to be recorded directly. The change in material surface together with

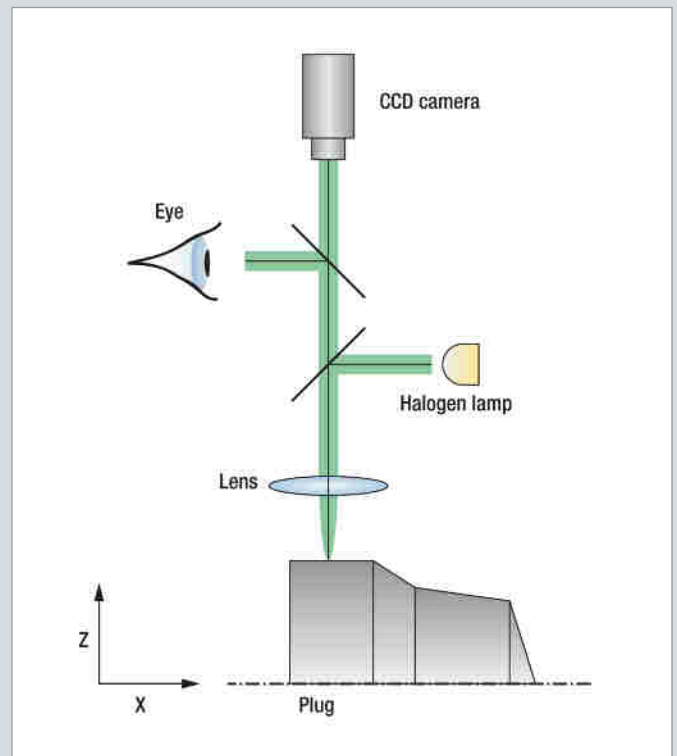


Fig. 11: Schematics of microscope for reflected light to analyse material tests

the duration of cavitation and the type of material used provides a quantitative unit for expressing the destructiveness of erosion.

The analysis results include information about the number, size (radii), and distribution density of the pits in the examined images. This information can be used, for example, to create damage maps which, in combination with the acoustic measurement results and the visual investigations of the cavitation zone, allow statements to be made on the destructiveness of cavitation-induced erosion.

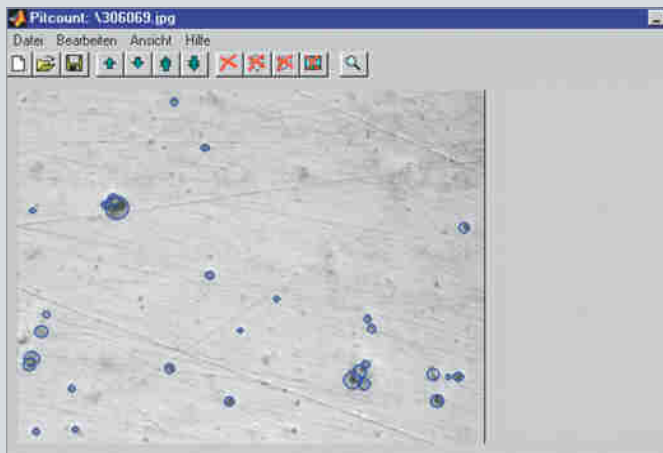


Fig. 12: Software user interface and analysis based on a typical sequence

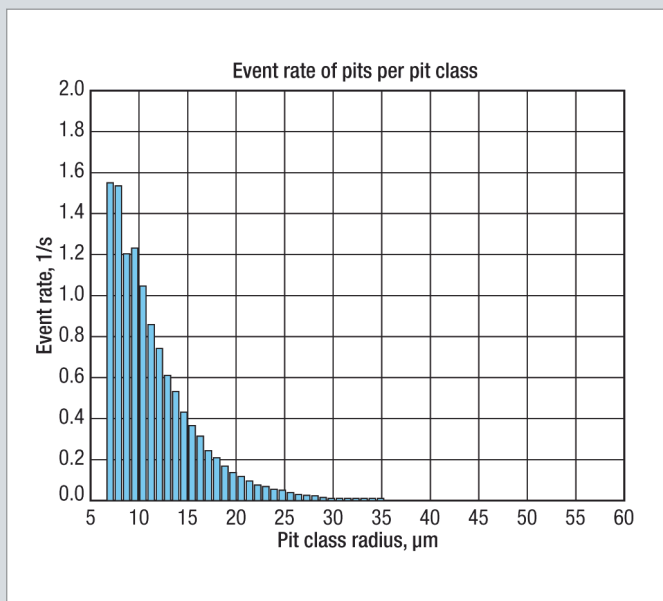


Fig. 13: Pit class radius histogram

6. Correlation between acoustic and material-based test results

The applied model used to calculate the implosion energy E_s using material tests is based on the research performed by Fortes-Patella [9]. The damage caused by erosion consists exclusively of plastic deformation. The shape of the three-dimensional, rotation-symmetrical deformation (pit) may be described as a cone defined by radius $R_{10\%}$ and the depth h . $R_{10\%}$ corresponds to the equivalent radius at a ten percent pit depth, whereby the roundness of the pit is taken into account. The formula for the correlation between damage and acoustic signal takes the pressure wave on bubble implosion mentioned in section 2 as the source of damage. The pressure wave itself is modeled as a Gaussian bell curve, its pulse duration $\tau_p / \sqrt{\pi}$ containing the pulse width from section 3.1.

The implosion energy in relation to the material deformation can be determined with equation (6). The ratio between plastic energy and deformed volume is expressed using a reference radius R_{ref} :

$$E_s = E_{S0}(\tau_p) \left(1 + \frac{R_{10\%}}{R_{ref}} \right)^3 \quad (6)$$

A pit radius histogram (Fig. 13) can be generated for each plug on categorizing the analyzed events as pits into radius classes. These are assigned to the acoustic events with the largest amplitude as the assumed cause of the pits.

With the assumption that just the larger acoustic amplitudes of the acoustically recorded events cause the pits, a threshold value was determined for every operating point to separate the damage-relevant acoustic events from those not relevant for damage. Integrating the pit frequency graph in Fig. 13, for example, results in a value of 11.82 pits per second. To achieve the same sum for greater acoustic amplitudes (Fig. 10, top), a limit of 73 N is required. All amplitudes greater than 73 N summed up also mean a frequency of 11.82 acoustic events per second. This threshold of 73 N is represented in Fig. 10 as a line.

The acoustic implosion energies measured above the threshold are equated with the material-based implosion energies and the pulse width τ_p is determined (Fig. 14, $\tau_p = 3.3 \mu s$).

The discovered parameters, such as pulse width and threshold for mechanical load, differ substantially from those for centrifugal pumps. As a result, cavitation turns out to be much more critical with control valves.

7. Results

After filtering out the acoustic power not relevant for component damage, diagrams were drawn up with the measured external noise level L_{pA} at one meter distance from the valve and the damage-relevant acoustic power plotted against x_F/x_{FZ} .

Fig. 15 illustrates these data for a parabolic plug at various loads by changing the valve opening. Although the cavitation causes audible external noise peaks from 85 to 95 dB around the control valve, damage to the valve is not to be expected. The acoustic power relevant for damage measured at the valve also starts to increase at much higher differential pressure ratios than x_{FZ} . At the point where the damage-relevant acoustic power increases, the pressure differential ratio x_{Fid} for the onset of damage can be determined. In addition, the total acoustic power and thus higher acoustic power at the plug is plotted in Fig. 9 for the maximum valve opening without taking into account the threshold, which would incorrectly indicate the risk of erosion at an earlier stage.

Fig. 16 shows the resulting values in comparison with the value from Table 1 ($x_{Fid} = 0.7$ for single-stage globe valve). The limits for the parabolic plug match well.

In the case of other plug types, different ratios were to be expected, which was also verified by a series of measurements. A clear correlation between these values and the operating point is recognizable when the implosion duration for all plugs is calculated. Shorter signal durations are expected for small valve strokes and likewise at higher pressure drops. This influence is taken into account to determine the final implosion maps. A linear rise in signal duration was found with increasing valve flow coefficients and with decreasing x_F values.

Naturally, the procedure can also be applied to other nominal sizes and valve styles. The measurements of the acoustic power relevant for cavitation erosion can be performed just as quickly as the measurements using the external noise level using a structure-borne sound sensor integrated into the plug. In an initial approximation, the mean threshold, pulse width dependent

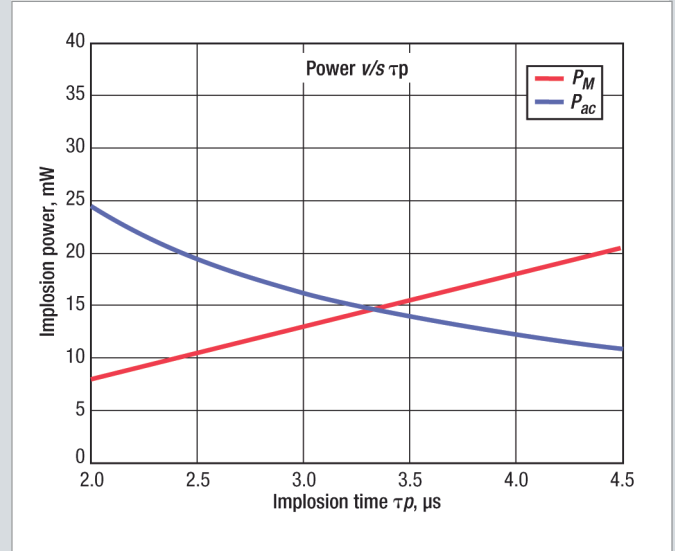


Fig. 14: Implosion power of both methods to determine the pulse width τ_p

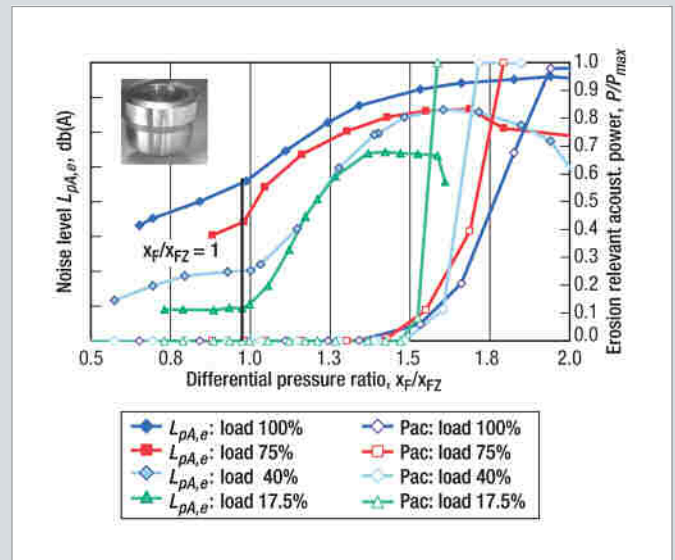


Fig. 15: Onset of cavitation-induced damage for coefficients greater than x_{FZ}

on the k_v coefficient, and differential pressure ratio x_F from existing tests can be used for further analysis.

At the same time, material tests should be performed for very large valve plugs to obtain reliable results. Further series of measurements show whether or how the limits x_{Fid}/x_{FZ} will change.

8. Outlook

It is possible to reliably predict the noise emission created by cavitation in control valves handling liquid flows using the new international EN 60534-8-4 standard. However, a standardized procedure to evaluate the destructiveness of cavitation erosion does not yet exist. An ISA recommendation does exist. However, the most important parameter $\sigma_{i,d}$ ($1/x_{fid}$) for incipient cavitation damage is not specified and can, until now, only be defined in complicated tests to determine the material erosion rate. Thanks to the cooperation between a research institute and industry, an acoustic method was successfully applied to control valves, which allows x_{fid} to be determined quickly.

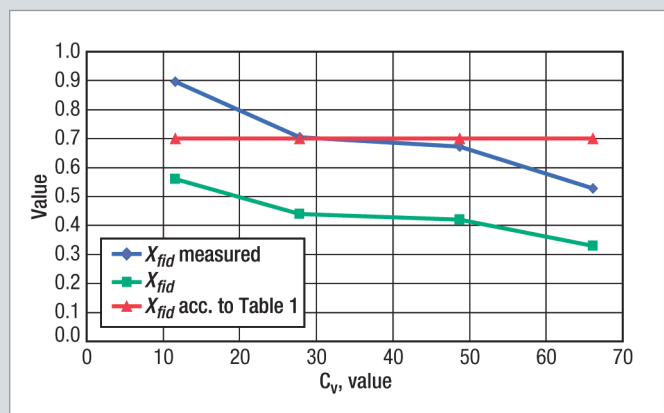


Fig. 16: Limits for damaging cavitation (parabolic plug)

Literature:

- [1] Kiesbauer, J.; Baumann, H.D.: Recent developments in prediction of hydrodynamic noise of control valves, Valve World Magazine Vol. 9, KCI publishing Issue 1, 2004
- [2] ISA Recommended Practice RP 75.23-1995: Considerations for Evaluating Control Valve Cavitation, The International Society for Measurement and Control, Research Triangle Park, NC, USA, 1995
- [3] Lohrberg, H.; / Stoffel, B.; Fortes-Patella, R.; Reboud, J-L.: Numerical and Experimental Investigations on the Cavitating Flow in A Cascade of Hydrofoils, Experiments in Fluids 33 (2002) 578-586, Springer-Verlag 2002
- [4] Lohrberg, H.: Messung und aktive Kontrolle der erosiven Aggressivität der Kavitation in Turbomaschinen Fortschritt-Berichte VDI-Verlag Nr. 942, Series 8, Düsseldorf 2002.
- [5] EN 60534-2-1: Industrial process control valves: Flow capacity: Sizing equations for fluid flow under installed conditions, Beuth Verlag GmbH, Berlin, March 2000
- [6] IEC 60534-8-4: Industrial process control valves – Part 8-4: Noise considerations – Prediction of noise generated by hydrodynamic flow, FDIS – Document 65B/556/ FDIS, 2004
- [7] Yuzawa, S.; Hashizume, T.; Okutsu, R. ; Outa, E.: Lifetime of a Control Valve Plug Related with Cavitation Erosion at Differential Pressure of 20 Mpa, 2002 ASME Fluid's Engineering Division Summer Meeting, Montreal, Quebec, Canada, July 14 – 18, 2002, Proceedings of ASME FEDSM'02
- [8] Stiles, G.F. : Cavitation in Control Valves, Journal of the Southern California Meter Association, Instruments & Control Systems, Vol. 31, Nov. 1964, pp. 2086-2092
- [9] Fortes-Patella, R.: Analyse de l'érosion de cavitation par simulations numériques d'impacts. Dissertation, l'Institut National Polytechnique de Grenoble, France, 1994
- [10] Kiesbauer, J.: Control Valves for Critical Applications in Refineries, Hydrocarbon Processing Magazine, Gulf Publishing Company, Houston, June 2001, Vol. 80, No. 6, pp. 89-100



Dr.-Ing. Jörg Kiesbauer is Director of the R&D division at SAMSON AG in Frankfurt/Main in Germany. His work experience includes R&D in the field of control valves equipped with electric and pneumatic accessories as well as self-operated regulators (flow and acoustical tests, development and optimization of calculation meth-

ods, development and testing of diagnosis tools for control valves etc., development of software tools). Since 1999, he has been involved in the IEC Working Group 65B-WG9 and in the DKE working group 963 as an expert.

Phone: +49 69 4009-1464
E-mail: drjkiesbauer@samson.de



Dipl.-Ing. Domagoj Vnucic is head of development test bench department at SAMSON AG in Frankfurt/Main in Germany. His work experience includes planning and evaluation of flow and acoustic laboratory tests carried out on control valves, application of CFD programs for

the purpose of flow calculation, development and optimization of calculation and sizing methods as well as the implementation of sizing software for control valves.

Phone: +49 69 4009-1796
E-mail: dvnucic@samson.de



Miriam Roth is a scientific assistant at the Institute of Turbomachinery and Fluid Power (TFA) of the Darmstadt University of Technology and a member of the working group hydraulic turbomachines and cavitation.

Her research involves the influence of the installation conditions of centrifugal pumps (experimental and numerical investigations) and acoustic research of cavitation.



Bernd Stoffel has been head of the Turbomachinery Institute of the Darmstadt University of Technology since 1985. He studied mechanical engineering at Karlsruhe University of Technology and was associate at the Institute for Fluid Dynamics and Fluid Machinery at the same university until 1977, earning his PhD. Between 1977 and 1985, Dr. Stoffel was with KSB AG

Frankenthal-Germany, where he had positions in the R&D department. His research in the institute involves cavitation and cavitation erosion in hydraulic machines and plants, dynamics of fluid components and systems, influences of fluid properties (viscosity, gas and nuclei content) in hydraulic machines and plants, the boundary layer behavior on turbomachinery blades and active flow instabilities in turbomachines.

■ Where innovation is tradition



SAMSON AG · MESS- UND REGELTECHNIK
Weismüllerstraße 3 · 60314 Frankfurt am Main · Germany
Phone: +49 69 4009-0 · Fax: +49 69 4009-1507
E-mail: samson@samson.de · Internet: www.samson.de