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VIBRATION OF A HIGH-PRESSURE PIPING SYSTEM DUE TO FLOW IN A SPHERICAL ELBOW

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During the commissioning of a 400 MW coal-fired power plant with supercritical steam parameters, severe vibration and intense noise at approximately 4 kHz were produced by the high-pressure piping used to by-pass the steam turbine. This vibration caused fatigue damage to several components attached to the piping; including the stem of the main by-pass valve. Since the vibration frequency was within the frequency range of valve noise, attention was first focussed on modifications of the by-pass valve to alleviate its coupling with any other possible sources in the piping system. However, all modifications of the valve design alone *influenced neither the level nor the frequency of vibration*, and therefore modifications in the layout of the high-pressure piping became unavoidable. Tests on a one-third scale model of the piping system without the valve indicated that the strongest excitation source in the system is a sphere-shaped elbow located upstream of the valve. After eliminating this spherical elbow and reinstalling the original version of the by-pass valve, the system operated without any vibration problems and the noise level was reduced to the design value.

1. INTRODUCTION

DURING THE COMMISSIONING OF UNIT 3 at *Vestkraft Power Station*, the high-pressure by-pass piping exhibited strong vibrations and severe noise when the system pressure and valve stroke exceeded certain limits. Shortly after the start of by-pass operation, several small-diameter pipes leading to pressure test outlets were broken. The fatigue life of these pipes was as short as 24 h. Field measurements were therefore conducted to seek the cause of the problem.

The power plant is a modern 400 MW unit with a supercritical coal-fired boiler. Normal operation is at boiler live steam output of 300 kg/s, 250 bar and 560°C. The high-pressure by-pass system is used during the start-up and shut down of the unit to by-pass the high-pressure steam turbine. This process requires the reduction of the steam pressure and temperature from the boiler outlet conditions to those at the inlet to the intermediate superheater. The by-pass system has also a safety function in case of turbine trips, i.e., when disturbances occur which necessitate abrupt disconnection of the turbine.

As illustrated in Figure 1, the by-pass system consists of two parallel lines, each including two pipes that stem from the boiler steam header and lead to a collector. The collector is

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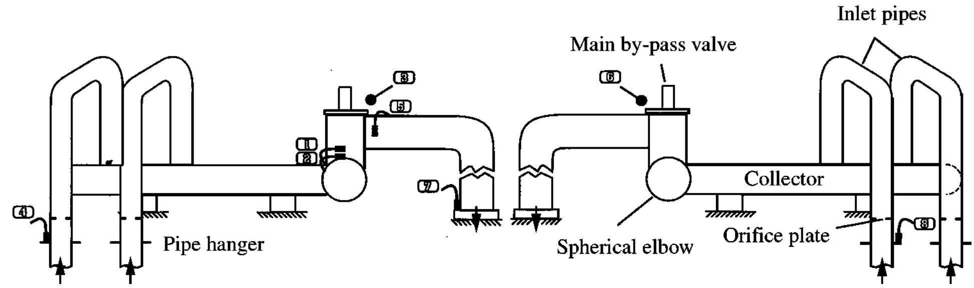

Figure 1. Geometry of the high-pressure by-pass system at Vestkraft Power Plant and locations of vibration and noise measurements. Locations 3 and 6: noise measurements; other locations: vibration measurements.

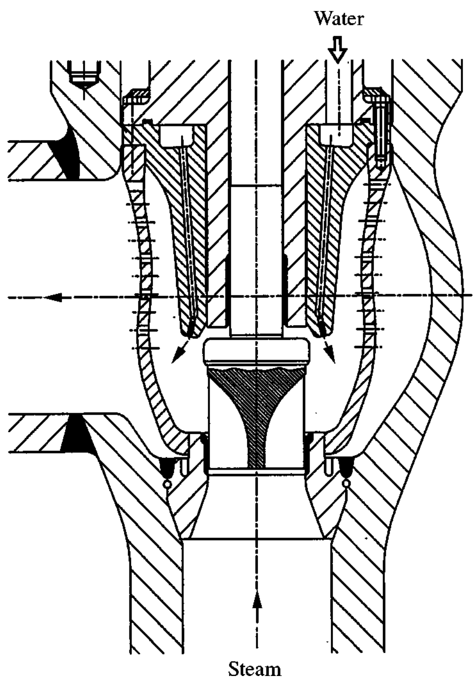

Figure 2. Geometry of the high-pressure, by-pass control valve.

followed by a 90°-elbow, the main by-pass valve and an outlet pipe which leads to the inlet pipe of the intermediate superheater. Due to design considerations related to the safety function of the by-pass valve, a spherical elbow is used between the collector and the valve. With this design, the reaction forces of the valve on the piping system are better managed when the valves are fully opened from the closed position in less than 1 s. This fast opening characteristic is a safety requirement.

The by-pass valve is a standard CCI (former Sulzer Thermtec Ltd) control valve type DRE-125, see Figure 2. It has a built-in water injection system to reduce the steam temperature after the valve *vena contracta.* Typical operating conditions at full load of this plant are 250 bar and 560°C at inlet, and 50 bar and 300°C at outlet, with a maximum flow rate of 150 kg/s per valve. The valve is especially designed for low-noise applications without compromising its robustness and reliable operation. The stem, which is shown in Figure 3, is slotted to form flow channels with different shapes and thereby eliminates

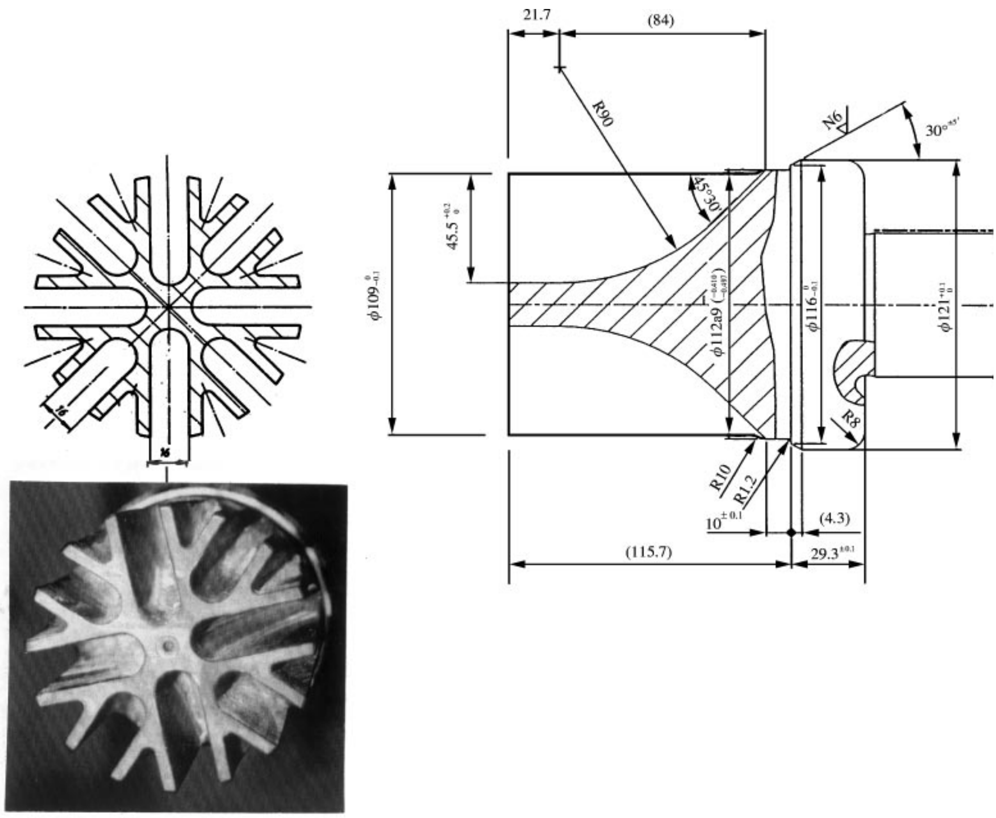

Figure 3. “Low noise” stem of the by-pass control valve. Dimensions in mm.

narrow-band excitations as described by Ziada *et al.* (1989). Water is injected as close as possible to the steam jet to improve atomization of the water jets and reduce the noise generated by the steam expansion (Ziada 1996). This valve is widely used in power plants worldwide without experiencing any vibration or noise problems.

During the commissioning period, an intense sharp tone near 4 kHz was produced when the inlet pressure exceeded 80 bar and the valve lift was increased beyond 65%. Since operation at these conditions endangered the safety of the plant, the operation mode was modified to avoid the production of this tone until the excitation mechanism was diagnosed and suitable counter-measures were formulated. This paper describes the vibration problem encountered with this by-pass system and the investigations performed until an effective counter-measure was implemented. This includes field measurements with the original and modified geometries of the by-pass valve as well as small-scale mode tests of the valve and the piping system.

2. FIELD MEASUREMENTS

Vibration and noise measurements were conducted to gain insight into the dependence of the vibration characteristics on the operating conditions, e.g., steam flow rate, pressure, temperature, as well as the valve lift. The measurements were conducted at several locations, which were kept the same in all subsequent measurements. These locations are shown in Figure 1 and are described in the following: (a) vibration on the short piece of pipe between the elbow and the valve, Locations 1 and 2; (b) vibration on the valve outlet pipe, Location 5; (c) vibration on the hanging brackets welded to the pipes upstream of the collector, Locations 4 and 8, and on the support bracket welded to the pipe downstream of the valve, Location 7; (d) valve noise at 10 cm from the valve, Locations 3 and 6.

All signals were recorded simultaneously on a digital data recorder for subsequent analysis in the laboratory. Since all measurements were taken during normal start-up or shut-down operation, it was not possible, except in very few cases, to change any of the test parameters independently. For example, while the system pressure or flow rate was being increased, the control system adjusted the valve lift automatically to meet the set values of the boiler load.

Typical vibration spectra measured on the pipe hanger at Location 4 are illustrated in Figure 4. Three spectra are shown: one at a low load before the onset of vibration; another at a medium load showing the vibration peak near 4 kHz; and a third at a high load indicating a slight increase in the vibration frequency as the load is increased. In fact, this increase in the frequency is found to be caused by a corresponding increase in the steam temperature as will be shown later.

During the commissioning period, it was possible to record vibration signals over a wide range of steam conditions. For example, tests of turbine full load rejection facilitated measurements at full boiler pressure and temperature, whereas tests at cold starts allowed measurements at part load operation of the boiler (i.e., low pressure and temperature). All data collected during these commissioning tests were analysed to find out which parameter controls the onset of vibration, e.g., pressure level, pressure ratio across the valve, valve lift or flow velocity. The outcome is illustrated in Figure 5. It is seen that although the steam conditions are varied from 80 to 250 bar and from 400 to 560°C, the vibration started always at a constant volume flow rate in the collector. This leads to the conclusion that the vibration is initiated at a constant flow velocity in the upstream piping. It should be noted here that the velocity in the downstream pipe is not constant because it depends on the pressure and the rate of water injection. Moreover, the pressure ratio across the valve is

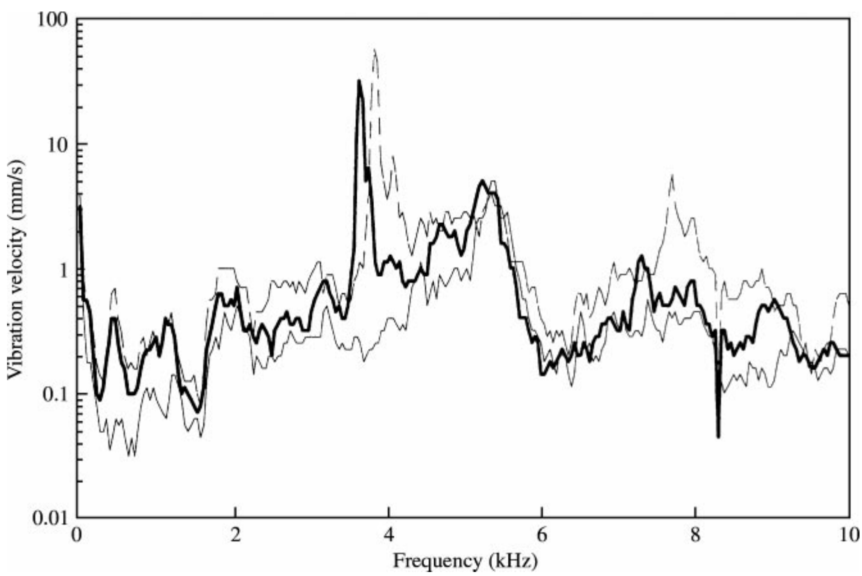

Figure 4. Vibration spectra measured on the hanger of the inlet pipes, Location 4: —, low load; —, medium load; - - -, high load.

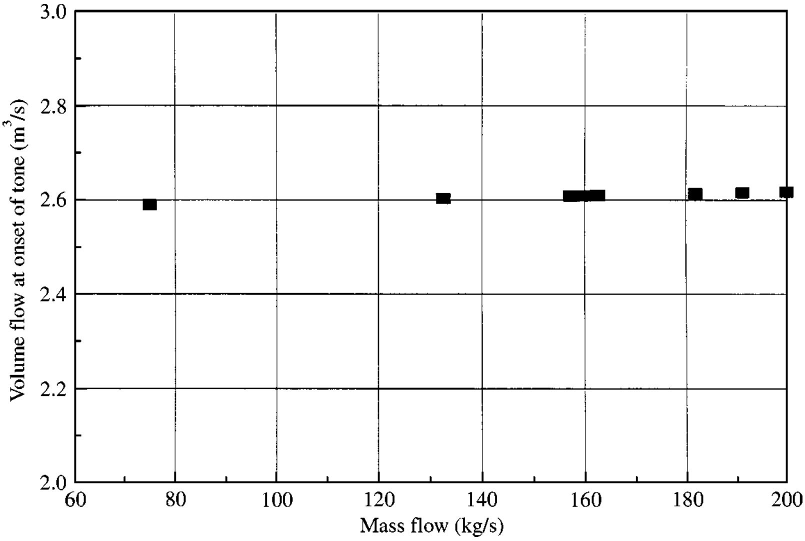

Figure 5. Volume flow rate in the upstream piping at the onset of vibration.

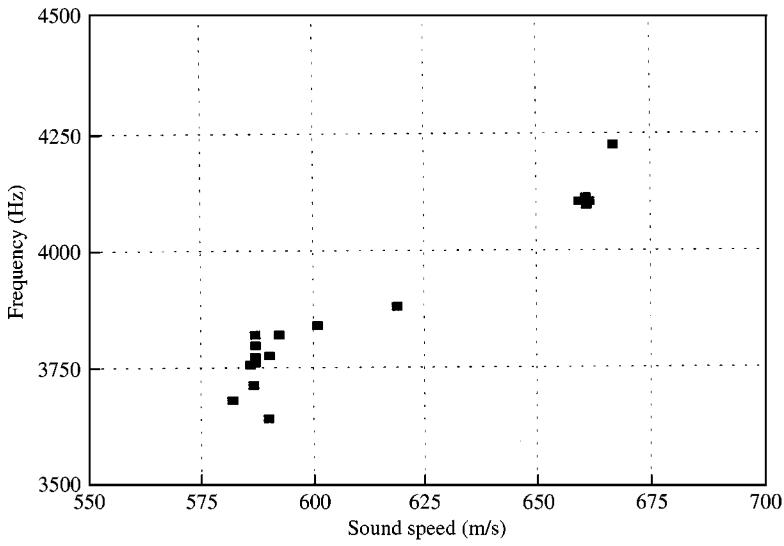

Figure 6. Vibration frequency as a function of the sound speed at the steam conditions in the upstream pipe.

always over critical, which results in sonic steam velocity at the valve *vena contracta* at all test conditions.

A closer look at the test results indicates that the slight variations in vibration frequency (from 3·6 to 4·2 kHz) are related to the steam temperature. Figure 6 shows that the vibration frequency increases almost linearly with the speed of sound, estimated from the steam conditions in the upstream piping. It is therefore very likely that the excitation of an acoustic resonator in the upstream piping is the principal cause of the problem. Mechanical resonances can be ruled out as a possibility because the resonance frequencies do not change with temperature as much as that observed during the tests. This hypothesis was also supported by the observation that the vibration of the upstream piping was stronger than that of the pipes downstream of the valve. Additionally, substantial variations in the rate of water injection inside the valve, up to $\pm 100\%$, at constant steam flow rate did not alter the frequency of the tone.

A simple analysis of the *acoustic* cross-modes (also called higher modes) of the pipes was therefore performed and the results were compared with the test results. The frequencies of the acoustic cross-modes of the pipe can be estimated from (Blevins 1993)

$$f_{mn} = \alpha_{mn}\, C/d, \tag{1}$$

where d is the inside pipe diameter, C is the speed of sound in steam and α_{mn} is the mode eigenvalue; m stands for the number of nodal diameters and n for nodal circles of the acoustic mode. The calculated frequencies are given in Table 1, in which F stands for the collector frequencies ($d = 175$ mm), and f for the frequencies of the inlet pipes to the collector ($d = 149$ mm).

A typical vibration spectrum taken at Location 4 *just before the onset of vibration* is shown in Figure 7. The main peaks which are 10–20 dB higher than the broad-band vibration are strikingly close to the estimated frequencies, see Table 1 also. Moreover, two of these peaks, F_{01} and f_{20}, are very close to the frequency of the observed vibration (3·6–4·2 kHz).

A study was made to assess the susceptibility of the main components to fatigue damage caused by vibrations. Comparison of the calculated stresses with the fatigue limit of X20 CrMoV 12 1 showed that the main components were at no risk of fatigue damage. It was therefore possible to continue the investigation following the normal operation schedule of the unit and all counter-measures were implemented during scheduled shut-downs. The small-diameter side branches that experienced fatigue damage were eliminated.

Although the above test results gave relatively clear evidence that the relevant resonator lies in the pipes upstream of the valve, they provided no clues whatsoever as to the nature of the excitation source! The Strouhal numbers, based on pipe diameters and flow velocities, were at least one order of magnitude higher than those associated with turbulent pipe flows or impinging shear flows (Naudascher & Rockwell 1994). On the other hand, the measured vibration frequency was centred in the frequency range corresponding to the highest noise level of the by-pass valve. This, together with the fact that the valve noise represents the strongest excitation source in the system, leads to the conclusion that the relevance of the valve noise in the excitation mechanism should be investigated first. If the valve noise was to play any role in enhancing the acoustic resonance, then it might be possible to alleviate the vibration by suitable changes in the valve design.

3. ACOUSTIC EXCITATION BY THE VALVE

Model tests were conducted to investigate the noise characteristics of the by-pass valve. A one-fifth-scale valve model was tested by means of pressurized air up to 5 bar. The piping

TABLE 1
Estimated and measured frequencies of the pipe cross-modes

Mode	Estimated (Hz)	Measured (Hz)
F_{10}	1942	1979
F_{01}	4040	3650–4200*
f_{10}	2281	2239
f_{20}	3783	3720
f_{30}	5204	5269

*Indicates the range of measured frequencies.

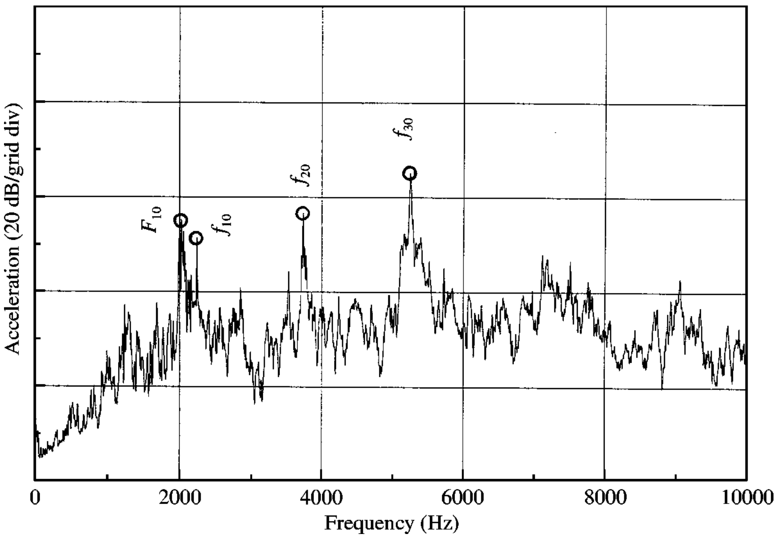

Figure 7. Vibration spectrum taken just before the onset of vibration showing that the dominant frequencies approximate those of the pipe acoustic modes.

system was not modelled because it was desired to know whether the valve by itself can generate discrete excitations. Moreover, it is rather difficult, if not impossible, to properly model the flow-acoustic coupling mechanism between the valve flow and the pipe acoustics by means of a small-scale model. This would require matching of dimensionless parameters involving flow velocity, speed of sound, density of the medium and acoustic impedance at the pipe termination.

Figure 8 shows typical noise spectra measured inside the downstream pipe of the model valve. The spectra at the left side represent the noise generated by a simple valve, which has a plain mushroom-shaped stem without any slots. It is seen that this type of valve generates narrow-band excitations. The Strouhal number of the spectral peaks agrees with that of the tones measured in power plants employing this old version of valve stem. This indicates that if the excitation mechanism (source and resonator if any) is confined to the valve, it would be possible to reproduce it in a small-scale model. A typical example of model testing of control valves has been reported by Ziada *et al.* (1989). They were able to reproduce flow-excited acoustic resonances inside the chest of a turbine control valve in a small-scale model.

The spectra given at the right side of Figure 8 correspond to a valve stem similar to that used in Vestkraft (as well as in many other power plants). This stem was developed to eliminate the discrete frequency excitations generated by the old version of this valve. As can be seen, narrow-band peaks are not produced and the noise spectra have a desirable wide-band character.

As it has been mentioned earlier, the vibration amplitude upstream of the valve in the power plant was higher than that downstream. This contradicts the anticipated pipe response to valve noise excitation as can be seen from the model test results in Figure 9. The maximum sound power level in the downstream pipe of the scale model is up to 30 dB higher than that in the upstream pipe, Figure 9(a). The cause of this large difference is that the pressure drop across the valve is over critical, and consequently the noise generated by the under-expanded jet cannot propagate upstream against the sonic flow velocity in the valve *vena contracta*. It is clear from Figure 9(b) that as the pressure ratio is increased, the

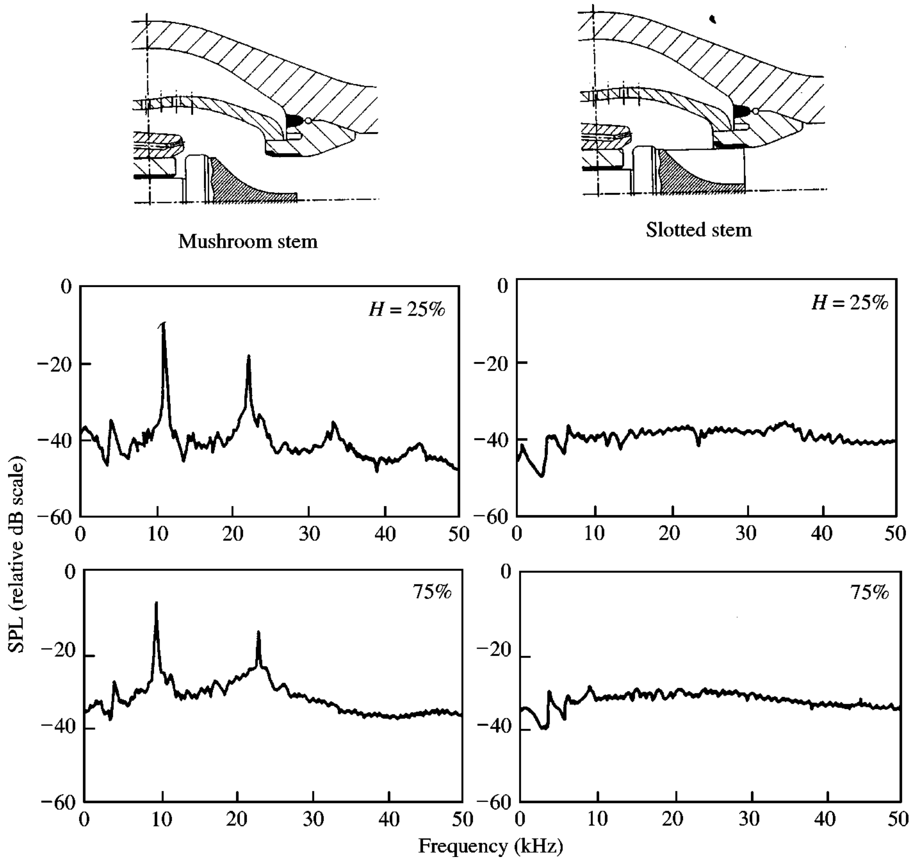

Figure 8. Typical noise spectra measured in the downstream pipe of model valve at a pressure ratio of 3. Left-hand side: valve with a simple "mushroom-type" stem; right-hand side: with a "low noise" slotted-type stem; H is the valve lift as percentage of maximum.

sound power in the downstream pipe also increases, but that inside the upstream pipe decreases. This does not agree with the vibration measurements carried out on the piping system in the power plant.

4. EFFECT OF VALVE GEOMETRY ON VIBRATION

Although the field and model tests indicated that the valve is not likely to be the primary source of excitation, the test results have not clarified the nature of the excitation source or its location within the upstream piping. Three components were suspected to constitute possible excitation sources: the metering orifice plates in the inlet pipes, the T-junctions at the collector inlets, or the spherical elbow between the collector and the valve. However, there was no guarantee that altering any one of these sources would solve the problem. Since these suspected sources are parts of the high-pressure piping, which has a design pressure of 275 bar, modifications to any of them are very expensive and time consuming. The risk involved in modifying any of these components at this stage was very high.

On the other hand, it was argued that even if the excitation source were not in the valve itself, the initiation of acoustic oscillations in the piping system would excite the powerful,

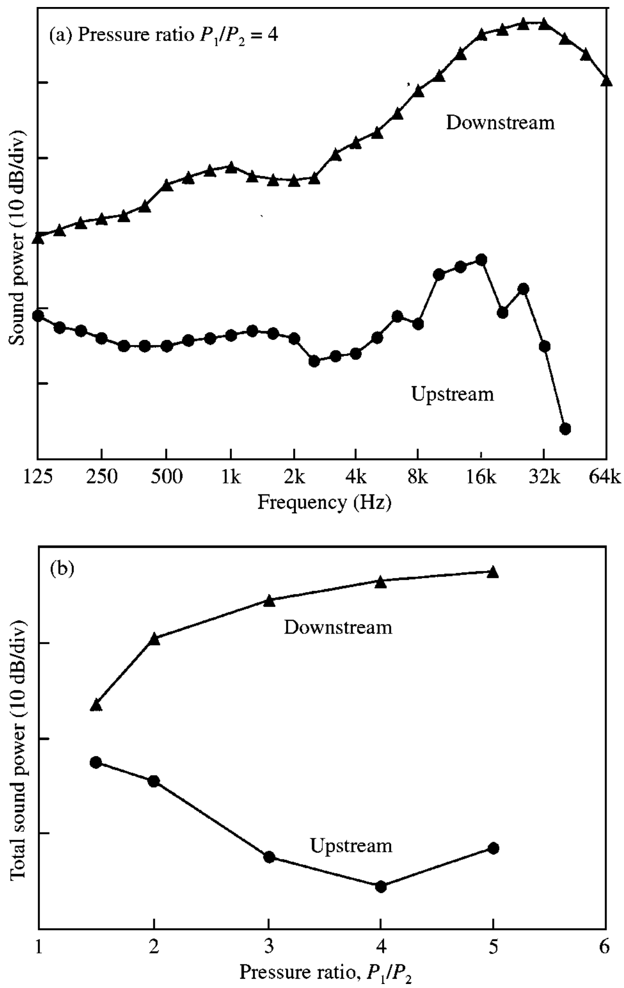

Figure 9. Results of the model valve tests: (a) third-octave spectra of sound power and (b) total sound power in the upstream and downstream pipes.

supersonic flow in the valve, which would then enhance the system oscillations. Additionally, modifications in the valve geometry are drastically simpler, faster and cheaper, compared to altering the high-pressure piping. It was therefore decided to carry out an additional test in which the valve geometry is fundamentally modified. The objective was to check the effect of the valve geometry (if there was any) on the coupling between the valve flow and the excited sound waves in the piping system. This test was regarded as the final check whether altering the valve design only could solve the problem.

Although the supplier of the by-pass valve (CCI- former Sulzer Thermtec Ltd) was not convinced that the valve is the primary cause of the problem, they agreed to undertake a radical modification to the valve geometry. As seen in Figure 10, the right-hand side, a plug inside a perforated cylinder replaced the original valve stem which is shown on the left-hand side of the drawing. Opening the plug increases the flow area by activating more holes of the perforated cylinder. The relatively long slots in the original stem were replaced by 168 short holes that are also smaller in diameter.

After implementing this valve modification, the vibration problem became even more acute. It started at a much lower pressure (20 instead of 80 bar) and a lower flow rate.

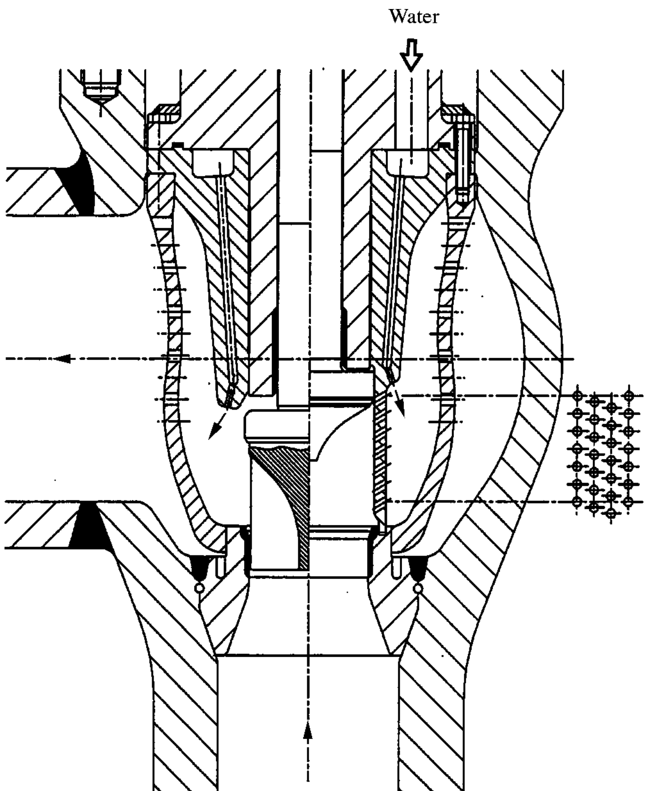

Figure 10. Modified geometry of the high-pressure by-pass valve; left-hand side: original geometry; right-hand side: modified geometry.

Moreover, a new tone was produced in addition to the original one near 4 kHz. A typical noise spectrum is shown in Figure 11. The frequency of the new tone increased continuously from 500 to 1500 Hz as the valve lift was increased. Since the vibration started at a much lower boiler load, it was not possible to start or shut down the unit without operating the system for long periods within the vibration range. Therefore, during the first scheduled shut-down, the original geometry of the valve was reimplemented. No detailed measurements or additional efforts were made to clarify the cause of the additional tone produced by the modified valve. To speculate, it seems that the new valve geometry is more reactive, i.e., it provides more reflection to acoustic waves in the upstream piping, and is probably more susceptible to acoustic excitation as well. Note that the slotted valve stem provides more gradual changes in the flow area than in the case of the modified stem, and that all holes have the same diameter, whereas the slots in the original valve have different geometries (cross-sectional area and length).

The new valve geometry did clearly influence the pipe vibration, the 4 kHz-component started at lower pressure and flow rate in comparison with the case with the original valve. Thus, the supposition that the vibration could be influenced by the valve design was confirmed; however, in this particular case, this influence was not the desired one. As mentioned earlier, the original valve geometry was especially developed to avoid generation of narrow-band noise, which explains its better “resistance” to narrow-band excitation, compared with the new design. Thus, the feasibility of developing another (or a better) valve, which could delay the onset of vibration, seemed to be uncertain, if not unlikely.

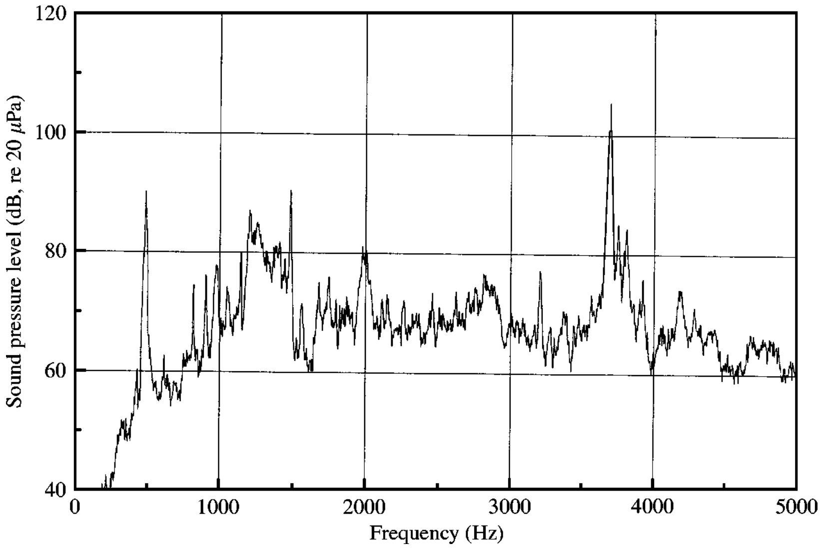

Figure 11. Typical spectrum of the noise generated by the system, with modified valve showing the original tone near 4 kHz and the new tone near 500 Hz.

5. MODEL TESTS OF THE PIPING SYSTEM

At this stage it was recognized that it was not possible to solve the vibration problem without changes in the piping system, but as mentioned before, the risk of any change was very high. It was therefore decided to carry out model tests of the piping system, despite the realization that it might not be possible to reproduce the excitation mechanism in the model. This is because the excitation energy in the model is much smaller than that in the prototype; the model is smaller and the test pressure is much lower. It was hoped, however, that model tests could provide some insight into the relative importance of flow noise generated by each of the suspected sources.

As shown in Figure 12, a one-third scale model was constructed and tested by means of pressurized air. Only the pipe section between the steam header and the valve were geometrically similar to the prototype. Since the valve geometry influenced the onset but not the nature of the 4 kHz component, a simple valve was used to pressurize the system. Later on, tests were continued without the valve, to facilitate tests at high flow velocities but lower pressures. The numbers given in Figure 12 indicate the location of pressure measurements. Vibration of the pipe wall was also measured at the same locations.

The Strouhal number, S, can be used to calculate the resonance frequency in the model, F_m, which corresponds to the vibration frequency in the prototype, $F_p = 4$ kHz. Since we are dealing with an acoustic resonance phenomenon, S is based on the speed of sound C,

$$S_c = (Fd/C)_p = (Fd/C)_m. \tag{2}$$

In this relation, substituting for the model scale $d_m/d_p = 1/3$, $C_p = 580$ m/s, $C_m = 340$ m/s, we obtain a resonance frequency of $F_m = 7$ kHz.

The same relation, but based on the flow velocity, can be used to calculate the flow velocity V_m at which the resonance at F_m would be excited in the model,

$$S_v = (Fd/V)_p = (Fd/V)_m. \tag{3}$$

Using $V_p = 50$ m/s, which is the flow velocity in the collector at the onset of vibration, equation (3) gives $V_m = 30$ m/s. Thus, if the excitation mechanism can be reproduced in the

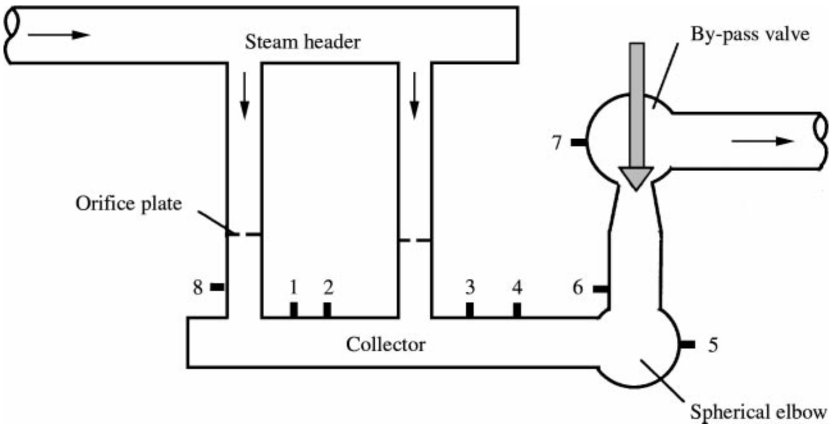

Figure 12. Geometry of the one-third scale model of the pipe.

model, an acoustic resonance near 7 kHz should be excited at a flow velocity close to 30 m/s in the collector.

Although many tests were conducted at several pressures and flow velocities (up to 4 bar and 80 m/s), it was not possible to reproduce the excitation mechanism. In the following, attention is focussed on the main results which helped in identifying the excitation source. The following features can be observed in the typical pressure and vibration spectra that are given in Figure 13: (a) the highest level of pressure fluctuation was measured between the spherical elbow and the valve at Position 6; (b) there was no narrow-band excitation in the piping system, especially near the anticipated resonance frequency (7 kHz)! (c) The pipe vibrations at its main resonance frequencies, near 3·8 and 9 kHz, were very pronounced, and the maximum vibration amplitude was measured on the pipe section between the elbow and the valve at Position 6.

The tests were then focussed on the effect of model geometry on the noise level generated by the whole piping system and the resulting vibration of the pipe wall. First, the valve was dismantled to facilitate tests at higher velocities and also to eliminate the valve noise. The noise generated by exhausting the air directly into the room was then measured. Tests with and then without the metering orifice plates did not show any changes in the flow noise. The T-junctions were then replaced by elbows with a radius of three pipe diameters and the tests were repeated with and without the spherical elbow.

As it can be seen in Figures 14 and 15, replacing the T-junctions with elbows, but retaining the spherical elbow, produced only minor changes in the pressure fluctuation inside the collector and the noise generated in the room (spectra (a) and (b) in Figures 14 and 15). On the other hand, when the spherical elbow was omitted altogether, the flow noise was drastically reduced; resulting in a reduction of 27 dB inside and 33 dB outside the collector (spectrum (c) in Figures 14 and 15). When the T-junctions were then replaced by the elbows, spectrum (d), an additional reduction of about 10 dB was observed at frequencies above 500 Hz. The latter reduction did not occur when only the T-junctions were replaced by the elbows, spectrum (b). This observation is indicative of the dominance of the noise generated by the spherical elbow.

The vibration of the collector wall was also drastically reduced when the spherical elbow was omitted. This can be seen in Figure 16, which shows typical vibration spectra measured at Position 4 with and without the spherical elbow. The total vibration level of the collector wall was reduced by an amount of 24 dB.

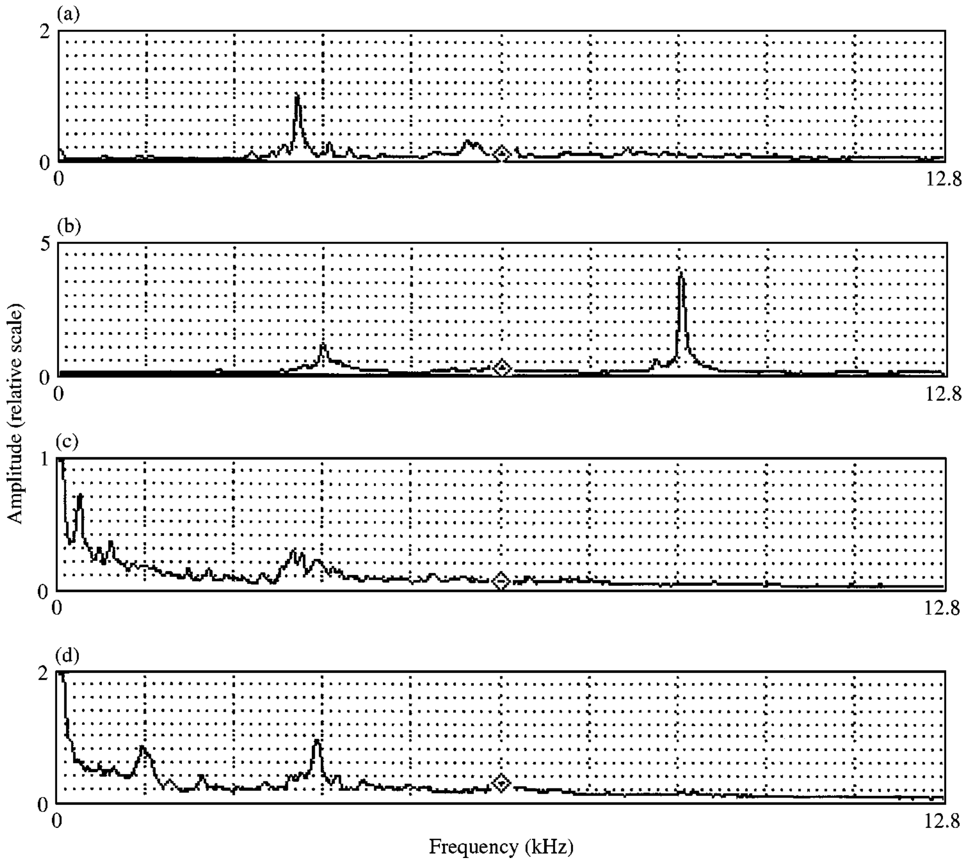

Figure 13. Typical spectra of pipe structural vibration and acoustic pressure measured during the model tests of the piping system. Pipe vibration at (a) Location 3 and (b) Location 6. Acoustic pressure at (c) Location 3 and (d) Location 6; test pressure = 1.6 bar; flow velocity in collector = 65 m/s.

6. MODIFICATION OF THE PIPING SYSTEM IN VESTKRAFT

Although the model tests were not successful in reproducing the vibration problem, it showed beyond doubt that the strongest noise source in the upstream piping system is the spherical elbow. As it turned out, the elimination of this element from the system was the simplest modification that could be made to the high-pressure piping. As shown in Figure 17, the spherical elbow was cut off and the valve was welded directly to the end of the collector. The rest of the piping modifications were made on the low-pressure piping downstream of the valve, which is much more economical than altering the high-pressure piping upstream of the valve. With this modification in the piping system, the vibration problem was eliminated altogether and the valve noise was reduced to an acceptable level. These features can be seen in Figure 18, which shows vibration and noise spectra taken before and after changing the pipe layout. During additional tests, the by-pass system was tested at full boiler load (i.e., 250 bar) and no abnormal vibration or noise levels were observed.

7. CONCLUDING REMARKS

The vibration problem described in this paper would seem at first glance to be caused by valve noise. This is because the valve is certainly the strongest noise source in the system

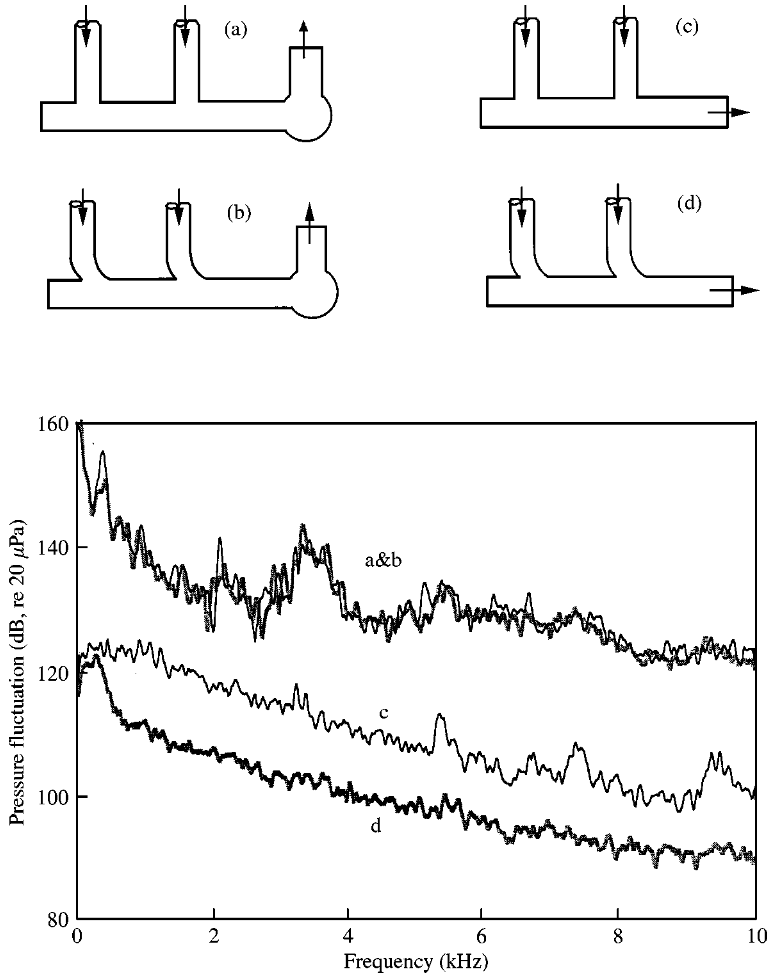

Figure 14. Spectra of pressure fluctuations inside the collector (Position 3) showing the effect of the spherical elbow and the T-junctions: (a) original geometry, total SPL = 165 dB; (b) with elbows instead of the T-junctions, 164 dB; (c) without spherical elbow but with T-junctions, 138 dB; (d) without spherical elbow and T-junctions are replaced by elbows, 131 dB. Flow velocity = 70 m/s.

and also because the vibration frequency is close to that at which the valve noise is at its maximum. However, the field measurements and model tests indicated that neither the system (acoustic) resonator nor the excitation source lie within the valve, but rather in the upstream piping. Despite this perception, it was rather difficult to localize the excitation source because the vibration component at 4 kHz was very strong at all measurement locations (due to structure-borne transmission), and it was not possible to measure the pressure pulsations inside the piping (because safety regulations prohibit drilling pressure taps in high-pressure piping). Additionally, it was not possible to reproduce the excitation mechanism during the small-scale model tests of the piping. These model tests, however, were crucial in clarifying the relative importance of different excitation sources existing in the upstream piping. The spherical elbow was found to be the strongest excitation source, and its removal from the by-pass piping eliminated the vibration problem altogether.

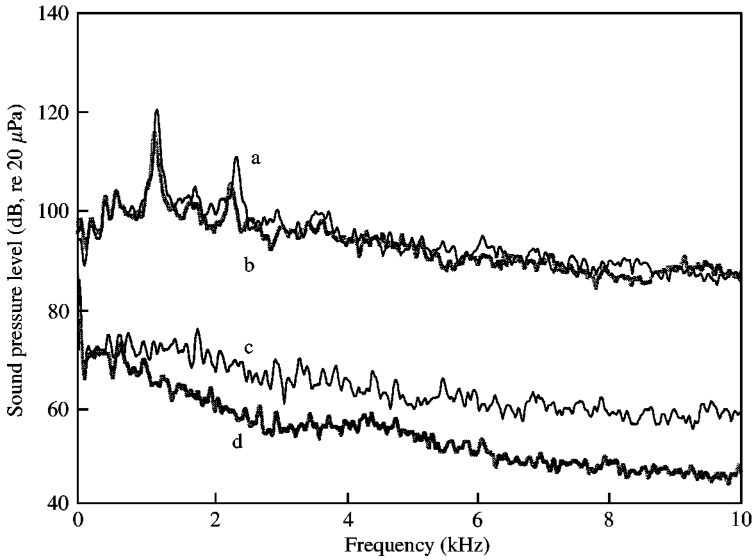

Figure 15. Noise spectra measured in the room showing the effect of the spherical elbow and the T-junctions on the generated noise. (a) original geometry, total SPL = 123 dB; (b) with elbows instead of the T-junctions, 120 dB; (c) without spherical elbow but with T-junctions, 90 dB; (d) without spherical elbow and T-junctions are replaced by elbows, 84 dB. Flow velocity = 70 m/s.

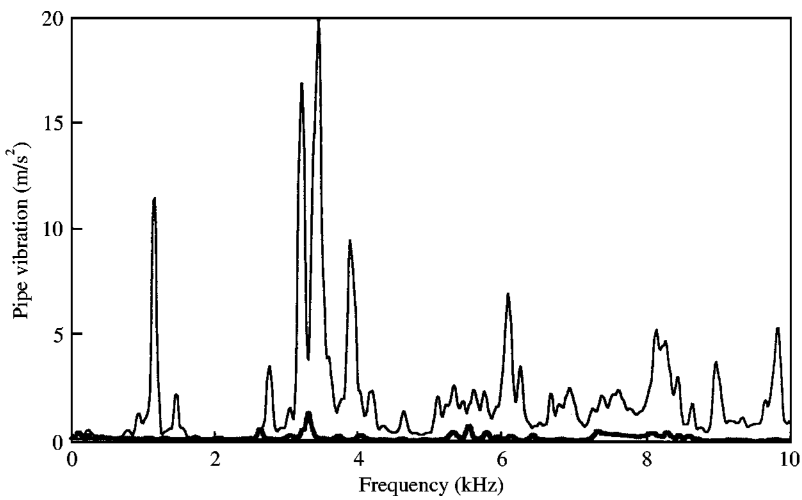

Figure 16. Effect of the spherical elbow on the pipe wall vibration at Position 3: —— with; ▬▬, without the spherical elbow. Flow velocity = 70 m/s.

Although the excitation source has been identified and an appropriate solution to the problem has been found, it must be noted that the nature of the excitation source is not understood. The flow in the spherical elbow has regions of flow separation and the downstream flow approaching the valve inlet is certainly not evenly distributed. It is not clear which location of flow separation is the real cause of the problem, nor which characteristic length should be used to scale the excitation frequency. It is interesting to note

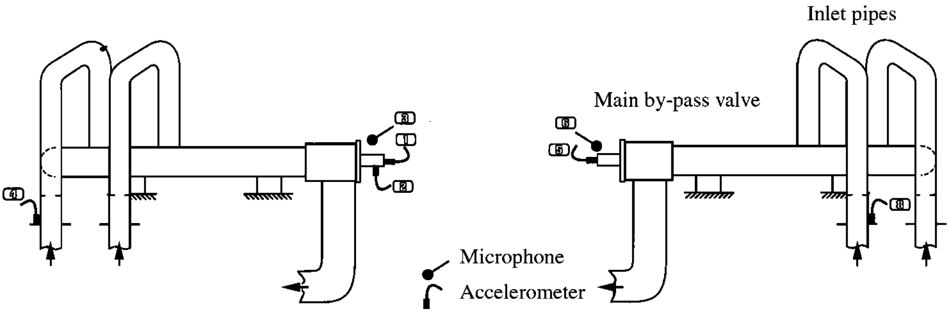

Figure 17. Layout of the high-pressure by-pass piping after removing the spherical elbow.

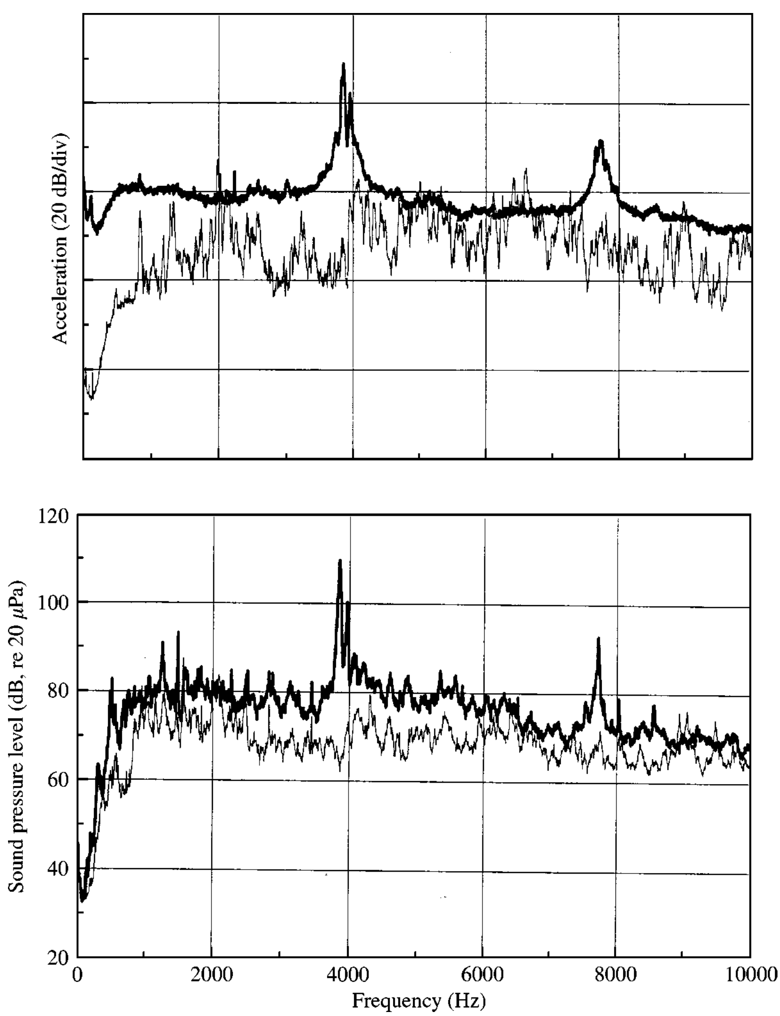

Figure 18. Effect of removing the spherical elbow on (a) the pipe vibration at Location 4 and (b) the noise level generated by the by-pass system. ▬ before; —— after removing the spherical elbow.

that the wavelength of the excited frequency is less than the pipe diameter and therefore the excited acoustic mode must be a complex cross-mode of the piping. This makes the present problem fundamentally different from that reported recently by Schafbuch *et al.* (1997) in which the flow recirculation at the inlet bore of a globe valve was coupled with longitudinal, acoustic modes of the upstream piping. The oscillation frequency in that case was two orders of magnitude smaller than in the present case. Consequently, the Strouhal number in the present case (~ 14) is very much higher than that expected for pipe flows. Schafbuch *et al.*, for example, reported a value of 0.1 for the Strouhal number based on the pipe diameter.

As a final comment, or rather a warning, the usage of such spherical elbows in high-pressure piping systems, which is becoming increasingly common, must be considered very carefully. The simplicity and cost-saving advantages gained by using this design, to avoid excessive pressure and thermal stresses in high-pressure piping, may increase the susceptibility of the plant to vibration problems similar to the one described here or that reported by Schafbuch *et al.* (1997). Further work is needed to better understand the nature of the excitation mechanism and means of its alleviation.

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