PREVENTING NOISE CAUSED BY VORTEX SHEDDING IN GATE VALVES AND ORIFICES

B.A.W. Smith, B.V. Luloff and J.H. Tromp
Atomic Energy of Canada Limited
Chalk River Laboratories
Chalk River, Ontario
Canada, K0J 1J0
Tel: (613) 584 8811 Ext.3887
Fax: (613) 584 4523
E-mail: smithba@aecl.ca

ABSTRACT
Gate valves and high-energy orifices can both act as sources of noise. This noise can represent a simple environmental problem or, in severe cases, can cause piping failures within hours. This paper presents the results of two studies, one to eliminate noise in the main steam lines of a new reactor, and, the other, to develop design guidelines for preventing noise in multi-stage, high-energy orifices. Both the valves and orifices were found to have a common noise generation mechanism, namely, an unstable fluid shear layer (e.g., vortex shedding) coupled with a fluid-resonant condition (i.e., an acoustic resonance). The main steam line noise was found to be caused by periodic vortex shedding across the seat cavities of the main steam isolation valves. The orifice noise is thought to be due to vortex shedding within the orifice holes themselves. This paper reviews the findings of both studies and presents measures that can be taken to eliminate noise problems.

NOMENCLATURE

c	speed of sound
d	depth of cavity
D	inside diameter
f	frequency
k_r	real part of wave number (see Equation 1)
L	length of cavity in axial direction
m	vortex shedding mode number
M	Mach number (U/c)
q	dynamic pressure ($\frac{1}{2}\rho U^2$)
Re	Reynolds number ($\rho UD/\mu$)
S	Strouhal number (fL/U)
U	average flow speed
Δp	pressure loss
λ	acoustic wavelength (c/f)
μ	fluid dynamic viscosity
ρ	fluid density

Subscripts

p	pipe
t	valve throat

INTRODUCTION

There have been a number of cases where the steam lines of nuclear generating stations have produced loud tonal noises that, although not a threat to the pipe integrity, pose an environmental hazard for nearby workers. AECL took part in a measurement program to identify the source of the noise at the CANDU station that was experiencing this problem. The outdoor noise level was found to be as high as 92 dBA and to have a dominant tonal component at about 500 Hz. Fig. 1 shows an example of a noise spectrum that was measured outdoors at site. These characteristics made the noise sufficiently annoying that modifications were considered to eliminate the noise source.

The series of noise and pipe wall vibration measurements was able to confirm that the Main Steam Isolation Valves (MSIVs) were the source of the pressure pulsations in the steam lines. The most important confirmation that these inline gate valves (shown in Fig. 2) were the source of the noise was that the pipe wall vibration frequency increased when the MSIV was closed slightly. This indicated that the noise source is related to the flow velocity in the valve throat and not to flow velocities elsewhere in the piping system.

Since the MSIVs were installed in an operating plant, any modifications would have to involve a minimum shut down period and would preferably be made in-situ. A short-duration, scale-model test program was undertaken to develop the optimum means of reducing the noise source in the gate valve. The modifications developed by this test program were implemented and were found to eliminate the steam-line noise. This paper describes the model test program and its findings as they relate to gate valves in general.

The high-energy orifices that are used in the pressure and inventory control system of CANDU stations are also prone to the production of intense noise, which, in some cases has caused piping damage in a matter of hours. Recently, intense noise was generated by a feed orifice at low heat transport system pressures during commissioning. This and other current designs consist of a series of one- or four-hole orifice plates spaced out to allow pressure recovery between plates. Although these multi-stage orifices have been optimised to avoid cavitation, the recent experience has demonstrated the need for revised design guidelines to avoid noise and vibration problems. There is also interest in reducing the size of these orifices and avoiding degradation caused by erosion. In order to develop the improved design guidelines, a literature review has been performed and an experimental program planned. This paper briefly reviews the current findings and outlines an experimental program planned to develop solutions.

VALVE TEST PROGRAM

Model Design

The VIBFLO air supply system at Chalk River Laboratories (CRL) was selected to provide a measured flow of air to the model. This system can provide at least 150 g/s of air on a continuous basis and up to about 530 g/s for short periods. To maximise the Reynolds number, *Re*, at a given Mach number, *M*, the highest possible air pressure and, hence, density must be used. Based on the capacity of this system, the available pipe supply, and the maximum pressure of the air, 3 inch plastic PVC piping was chosen to represent the steam line. The length scaling factor was therefore set to 0.115, the ratio of the inside diameter of the model pipe to the inside diameter of the steam line.

Figures 3 and 4 show a drawing and a photograph, respectively, of the valve model installed in the pipe. The model consisted of three main sections made of acrylic: an upstream reducing section, a downstream diffusing section, and a central disk-shaped insert to model the valve seat cavity. Four bolts were used to clamp the model and PVC pipe flanges together. Machined aluminium sleeves around the bolts were used to align the pipe and model sections. O-rings provided sealing. The model valve seats were designed to be easily interchangeable.

Figure 5 shows the five central inserts representing the different valve seat cavities that were tested. Insert 1 represents the existing, reference valve design, that has a cavity (larger diameter region between the two seats) that varies in depth around its circumference. The guide bars and stop bar that are located in the middle of the cavity are used to help position the disk in the real valve. Insert 3 represents a design modified to reduce the cavity depth to the minimum allowed by the existing disk. Inserts 1 and 3 both connect the valve throat to a volume representing the bonnet cavity that was machined into the main upstream and downstream model components. Inserts 2, 4 and 5 have circular holes that result in uniform cavity depths. Insert 2 has the same inside diameter as the reference seat and hence eliminates the valve seat cavity. Inserts 4 and 5 result in cavities that have length (*L*) to depth (*d*) ratios of 5.8 and 1.7, respectively

The central inserts were designed to be tested with one of the three disk designs shown in Fig. 6. These disks may be described as follows: Disk 1, the existing circular disk; Disk 2, a D-shaped disk designed to minimise the cavity volume; and Disk 3 an extended disk designed to minimise the cavity volume for use should the stop bar be removed.

The thirteen axisymmetric valve seat designs used in the testing program are illustrated in Fig. 7. A minimum sealing face height of about 1.27 mm at model scale was considered necessary to ensure reliable valve sealing. For simplicity, only axisymmetric seat designs were considered. Possible solutions involving flow-deflecting vanes were not tested because of concerns about parts coming loose.

The model steam line consisted of a 3-inch PVC pipe connected to the existing VIBFLO loop air supply. The downstream end of this pipe was connected to a 2-inch PVC pipe containing a valve for controlling the pressure in the test section. The initial phase of testing employed a gate valve for pressure control. This gate valve was replaced with a ball valve for the second phase because the gate valve produced its own tonal noise under certain conditions.

Although the exact length of the model steam pipe and the position of the gate valve model in this pipe were not considered important, the total length of the PVC pipe was scaled to represent the approximate length of a steam line to ensure the high axial acoustic modal density that is representative of the long steam lines. The gate valve model was placed about 120 diameters (9 m) from the inlet end of the PVC pipe to ensure that the turbulent flow in the pipe was fully developed.

The VIBFLO air supply has a flow control valve controlled by a microprocessor-based system connected to flow, pressure and temperature sensors. The system allows for either manual or automatic operation of the valves and displays the mass flow, absolute pressure and temperature of the air in the measuring leg.

The model was instrumented to measure unsteady pressures, average static pressure, and pressure drop across the valve. Five quartz-based dynamic pressure transducers were flush mounted in the wall of the plastic pipe. Three were mounted upstream from the valve and two were mounted downstream. All of these pressure transducers were at the same circumferential position on the pipe. The axial positions were chosen to ensure that at least one of the transducers would be close to an antinode of any standing wave pattern in the pipe at the frequency of the tonal noise. Inserts 1, 4 and 5 were fitted with a sub-miniature quartz based pressure transducer recessed at the side of the cavity.

The model Reynolds number was limited to about 0.52 million while the original steam line had a Reynolds number of about 30 million at 100% flow. At these Reynolds numbers and considering the upstream flow path in the model, turbulent flow would be expected in both the full-scale and the model valves. Boundary layer estimates indicate that the velocity profile upstream of the valve seat cavity was well represented in the model test, especially at the highest model Reynolds numbers achievable.

Experimental Procedures

Thirty different valve configurations that were tested to arrive at an optimum solution. Each valve configuration was tested at between 7 and 19 different conditions. The test section conditions were determined from the mass flow and temperature measurements displayed by the flow control system and the test section pressure measured with the strain gauge pressure transducer. Typically, an initial test series was performed where the mass flow was set to give Mach numbers that generally ranged between 0.05 and 0.10 at a Reynolds number of about 0.145 million. The Mach number of the full-scale steam line was 0.073 at 100% flow. Often, a few more short duration tests were done with the highest possible air flows to achieve Reynolds numbers up to 0.52 million.

During each test, signals from the dynamic pressure, static pressure and pressure drop transducers were recorded simultaneously on a tape recorder and analysed for spectral content. The band root mean square (RMS) unsteady pressures quoted were determined over a 1000 Hz frequency band that was centred about the dominant tonal noise frequency if one was present (as in the spectra in Fig. 8). When tonal noise was not evident (as in the spectra in Fig. 9), a frequency band from 3000 to 4000 Hz was used. The highest of the RMS pressures measured in the pipe for each condition was taken as the pipe unsteady pressure. The unsteady pressures were normalised by the dynamic pressure in the valve throat ($q_t = \frac{1}{2}\rho U_t^2$). This dynamic pressure was calculated from the average throat velocity, U_t, which was based on the minimum cross sectional flow area of the valve configuration being tested.

Valve Noise Generation Mechanism

The scale model test was able to reproduce the essential features of the noise problem observed in the steam line. Figure 8 shows the spectra of the upstream pipe pressure, P_1, measured with the reference valve configuration. The pipe spectra show a dominant narrow band noise peak that varies from about 3320 Hz (S=1.12) at Mach 0.090 to 2833 Hz (S=0.93) at Mach 0.064.

When the valve seat cavity was filled in with Insert 2 the tonal noise disappeared and the broadband noise decreased. The corresponding spectra decreased monotonically with increasing frequency as happens with random turbulence.

The broad noise peaks, characteristic of vortex shedding in the absence of acoustic resonance, are seen in the pressure spectrum shown in Fig. 8. At Mach 0.079, for example, these peaks are centred on Strouhal numbers of about 0.52, 1.05, 1.55, and 1.97, respectively.

Much work has been done to understand and control vortex shedding over cavities in walls and external cavities in axisymmetric bodies of revolution (e.g., Naudascher and Rockwell 1994, Lucas et al., 1997, Blevins 1990). Because of the inherent instability of a free shear layer, small disturbances at the upstream edge of the cavity result in the formation of discrete vortices in the shear layer over the cavity. Each vortex will convect downstream until it impinges on the downstream edge of the cavity causing a pressure perturbation there. This pressure perturbation will then be acoustically transmitted back to the upstream edge where it can initiate the formation of another vortex. The time required for the vortex to move across the mouth of the cavity plus the time required to transmit the pressure back to initiate the formation of a new vortex will determine the preferred vortex shedding frequencies.

Block (1976) developed the following equation to estimate the Strouhal number of vortex shedding over a relatively shallow cavity at low subsonic Mach numbers.

$$S = \frac{fL}{U} = \frac{m}{\dfrac{1}{k_r} + M\left(1 + \dfrac{0.514}{L/d}\right)} \qquad (1)$$

where k_r is the real part of the wave number that has a value of 0.57, d is the depth of the cavity and m is the mode number. For a representative average depth, Equation 1 indicates that Strouhal numbers of 0.51, 1.01, 1.52 and 2.03 would be expected for the first four vortex shedding modes. These Strouhal numbers match the Strouhal numbers that were observed in the valve model helping to confirm that the broad peaks are due to vortex shedding over the cavity.

The narrow band noise peaks near 3240 Hz in Fig. 8 appear to be due to a fluid resonant condition caused by a coupling of an acoustic resonant mode of the valve with a vortex-shedding mode of the cavity. Frequency response functions (FRFs) of pipe pressure with respect to cavity pressure were taken with the four possible valve orientations. These FRFs indicated that, at the main tonal frequency, the pipe pressures inline with the sides of the valve were three or more times those inline with the top (stem) or bottom (stop bar) of the valve and that the high pressures on the opposite sides of the pipe were about 180° out-of-phase with each other. This pattern is very similar to that of the lowest frequency acoustic cross mode (Blake 1986) that has a mode shape characterised by a single sine wave around the circumference of the pipe and a model-scale cut-off frequency of 2800 Hz. This mode is consistent with out-of-phase vortex shedding from opposite sides of the valve seat cavity coupled with an acoustic mode across the throat. At the frequency of the highest noise peaks, the throat diameter consistently represents about 0.45 wavelengths.

Since the deep axisymmetric cavity (L/d=1.7) produced by far the loudest tonal noise, it appears that the bonnet cavity of the valve is not required for the production of tonal noise, although it may have a small influence on its frequency. Fig. 10 shows that the shallow axisymmetric cavity (L/d=5.8) greatly reduced the tonal noise production, possibly by stabilising the shear layer.

Figure 10 shows curves of normalised pipe RMS pressures versus Mach number for a selected set of model configurations tested at Reynolds numbers of less than 0.25 million.

Disk modifications were not found to be effective at reducing the noise levels while cavity modifications of the full-scale valve were judged impractical. The test program therefore focused on valve seat modifications.

Various authors have reported that vortex shedding instabilities can be significantly reduced if the flow is prevented from impinging on the downstream edge of the cavity either by moving the downstream edge out of the flow or deflecting the flow at the upstream edge (Naudascher and Rockwell 1994, Rockwell and Knisely 1979, Ethembabaoglu 1978, 1973, Blevins 1990, Willmart et al. 1978). Pressure variations have also been reduced by rounding and inclining the downstream corner (Ethembabaoglu 1978, 1973, Heller and Bliss 1975). To test these approaches, Seats 3, 2 and 4, were installed in the downstream position while a standard seat was used upstream. These downstream seats had rounded or chamfered corners with the minimum sealing face height (see Fig. 7). These three configurations all reduced the maximum unsteady pressure coefficients, but only slightly.

Franke and Carr (1975) found that ramps or chamfers upstream and downstream of a cavity greatly reduced the pressure fluctuations in the cavity as long as the flow was separated over the upstream ramp. To test this approach, Seats 2, 3 and 4 were tested in both the upstream and downstream positions. These configurations all produced significant reductions in the tonal noise but Seat 3, with a 7.7° chamfer, produced the lowest RMS pressures and performed well over the full range tested from Mach 0.053 to 0.084. Changing only the upstream seat to Seat 4 was almost as effective at reducing the noise as changing both the upstream and downstream seats but caused higher pressure losses.

Because of the encouraging results for the double ramp configurations, Seats 5 to 12 were developed and tested in configurations with identical seats upstream and downstream. Seat 13 was used in the upstream position in a configuration with Seat 10 downstream.

Of the seats with a rounded corner (radius>0.13 mm), Seat 4 performed the best. Figure 10 shows that below a pipe Mach number of about 0.079, Seat 4 reduced the noise to background levels in the pipe. Above this flow, the tonal noise levels rose with increasing flow to levels similar to those observed with the reference design at Mach numbers of 0.090 and above.

All of the chamfered seats with a sharp corner (0.13 mm radius) showed significant improvements over the reference configuration. However, with the exception of Seats 7 and 10, all eventually produced a loud tonal noise as the Mach or Reynolds number was increased. Up to a point, the Mach number at which the noise began to rise increased with decreasing chamfer angle. At Reynolds number of less than 0.25 million the Mach numbers at which the noise level rose were: Seat 5 (0° chamfer) - 0.071; Seat 6 (4° chamfer) - 0.079; and Seat 3 (8° chamfer) - 0.086. At higher Reynolds numbers, the increases in noise level tended to occur at lower Mach numbers for these seats.

While, the full and medium length 15° chamfers on Seats 7 and 10, respectively, were almost equally effective at reducing noise, the short 15° chamfer on Seat 12 was generally noisier and produced an increase in noise level above Mach 0.090. Seat 13 was designed with a 12° chamfer that begins with a step to ensure flow separation. The performance of Seat 13 was similar to that of Seat 12 except that its noise level did not rise rapidly until above Mach 0.097.

Figure 9 shows a multiple spectra plot of the pipe pressure obtained using Seat 10. This seat did not produce any significant tonal noise at any of the conditions tested including those at higher Reynolds numbers. Unlike Seat 7, this seat also had a sealing face high enough to ensure reliable valve sealing. Seat 10 was also very effective at reducing the intense noise generated by the deep axisymmetric cavity. This is an indication that the effectiveness of Seat 10 is not dependent on the details of the cavity such as its depth or the presence of the guide or stop bars. Because of its effectiveness over a wide range of conditions, the Seat 10 design was selected for implementation in the steam line valves

Implementation

A custom-made machine was used to reshape the seats of the MSIVs at the CANDU station to give them the profile of Seat 10. This modification successfully eliminated the noise coming from the Main Steam Lines.

HIGH-ENERGY ORIFICE STUDIES

Based on the limited evidence available, the noise produced by the Pressure and Inventory Control feed orifice was found to be due to a vortex shedding from the square upstream edge of the orifice holes. As in the MSIV, fluid-dynamic (due to intermittent reattachment) and fluid-resonant (i.e., acoustic resonance) feedback mechanisms were found to be acting together to cause the intense noise. Acoustic resonance is thought to occur in the volume between one orifice plate and the next.

To avoid the problem experienced with the feed orifice mentioned above, it is recommended (a) that the thickness-to-diameter ratio of orifice holes be less than 0.1 or greater than 2.0 to avoid intermittent reattachment of the free shear layer, or (b) that the upstream edge be given a radius that is at least 20% of the hole diameter. By rounding the upstream orifice edge the orifice will be made far less prone to degradation caused by erosion of the square upstream edge.

To avoid the possibility of noise problems it is recommended that the range of vortex shedding frequencies be kept below the lowest expected acoustic resonant frequency. Until tests can be done, it is suggested that 1.0 be used as the maximum Strouhal number of the vortex shedding.

It is recommended that the inter-stage distance not be reduced to less than 10 times the diameter of the holes in the upstream orifice plate to avoid the possibility of fluid-dynamic feedback from the jet striking the downstream plate.

A test program has been designed to develop more reliable and compact multi-stage orifices. The tests will examine the possible noise production mechanisms, evaluate the effect of inter-stage distances on pressure drop, and examine the cavitation issues in more compact orifice designs. A full-scale test section has been designed and certified, and the components procured. Although the test section is designed to accommodate heat transport system pressures, the planned tests will use water and air at pressures of less than 1 MPa.

To avoid erosion damage it is conservatively recommended that flow velocities be kept to below 50 m/s. Inspection of orifices that have been used in the P&IC system is critical to developing revised design guidelines regarding erosion.

SUMMARY

A 0.115 scale model using air to represent steam successfully reproduced the mechanism causing a tonal noise in a large high-pressure steam line and to develop a practical means of reducing this noise. The source of the narrow band tonal noise in the steam line was a second mode vortex shedding across the valve seat cavity coupled with an acoustic mode in the valve. The acoustic mode resulted in out-of-phase pressures on opposite sides of the valve seat cavity. This excited an acoustic cross mode in the steam piping.

To avoid excessive noise caused by gate valves in high-pressure steam lines the throat velocity should be kept less than 50 m/s or the valve should be fitted with an eyepiece to eliminate the valve seat cavity when the valve is fully open. These measures will also help to reduce pressure losses. If an eyepiece is not used then the valve seat cavity should be kept as shallow as possible while still allowing for adequate sealing face height. In venturi-type valves, high contraction ratios should be avoided to keep the velocities as low as possible and the boundary layer as thick as possible.

The model-testing program indicated that 15° chamfers added to both the upstream and downstream valve seats eliminate tonal noise production. It is estimated that these chamfers should each extend axially no less than 20% of the width of the cavity. The model-testing program showed that rounding and/or chamfering the upstream and downstream valve seats in other ways can also be effective depending on the flow conditions.

If a scale model of new valve designs is being tested prior to production (e.g., to confirm pressure drop estimates), it is recommended that model tests be done at the full scale Mach number and that pressure measurements be made in the model to ensure that no tonal noise source is present. As a minimum, a new valve design should be studied to confirm that the expected vortex shedding frequencies do not coincide with any obvious acoustic modal frequencies of the valve.

It is now thought that the current multi-stage high-energy orifice deigns are prone to generate intense and potentially damaging noise. This noise is thought to be due to vortex shedding along the separated shear layer in the orifice holes combined with acoustic resonance between subsequent orifice plates. Tests will be performed to confirm these conclusions and develop new design guidelines for multi-stage orifices. These guidelines are expected to permit the design of quiet, more-compact orifices that are not subject to degradation by erosion.

ACKNOWLEDGEMENTS

The authors wish to thank R. Hartlen from Plant Equipment Dynamics, Inc. for many useful discussions and some good advice during the course of this work.

REFERENCES

Blake, W.K., 1986, Mechanics of Flow-Induced Sound and Vibration, Vol. 2, Academic Press.

Blevins, R.D., 1990, Flow-Induced Vibration, Van Nostrand Reinhold.

Block, P.J.W., 1976, "Noise Response of Cavities of Varying Dimensions at Subsonic Speeds," NASA-Langley Research Centre Technical Paper D-8351.

Ethembabaoglu, S., 1978, "Some Characteristics of Unstable Flow Past Slots," Journal of the Hydraulics Division, ASCE, Vol. 104/HY5, pp. 649-666.

Ethembabaoglu, S., 1973, "On the Fluctuating Flow Characteristics in the Vicinity of Gate Slots," Thesis for Division of Hydraulic Engineering, The University of Trondheim, The Norwegian Institute of Technology.

Franke, M.E., and Carr, D.L., 1975, "Effect of Geometry on Open Cavity Flow-Induced Pressure Oscillations," AIAA Paper 75-492, AIAA Second Aero-Acoustics Conference, Hampton, Virginia, 24-26.

Heller, H.H., and Bliss, D.B., 1975, "The Physical Mechanism of Flow-Induced Pressure Fluctuations in Cavities and Concepts for Their Suppression," AIAA Paper 75-491; also Progress in Astronautics and Aeronautics, Vol. 45, pp. 281-296.

Heller, H.H., Holmes, D.G., and Covert, E.E., 1971, "Flow-Induced Pressure Oscillations in Shallow Cavities," Journal of Sound and Vibration, Vol. 18, No. 4, pp. 545-533.

Lucas, M.J., Noreen, R.A., Sutherland, L.C., and Cole III, J.E., 1997, Handbook of the Acoustic Characteristics of Turbomachinery Cavities, ASME.

Naudascher, E., and Rockwell, D., 1994, Flow Induced Vibrations: An Engineering Guide, A.A. Balkema Publishers.

Rockwell, D., and Knisely, C., 1979, "The Organized Nature of Flow Impingement Upon a Corner," Journal of Fluid Mechanics, Vol. 93, pp. 413-432.

Willmart, W.W., Gasparovic, R.F., Maszatics, J.M., McNaughton, J.L., and Thomas, D.J., 1978, "Management of Turbulent Shear Layers in Separated Flow," Journal of Aircraft, Vol. 15, No. 7, pp. 385-386.

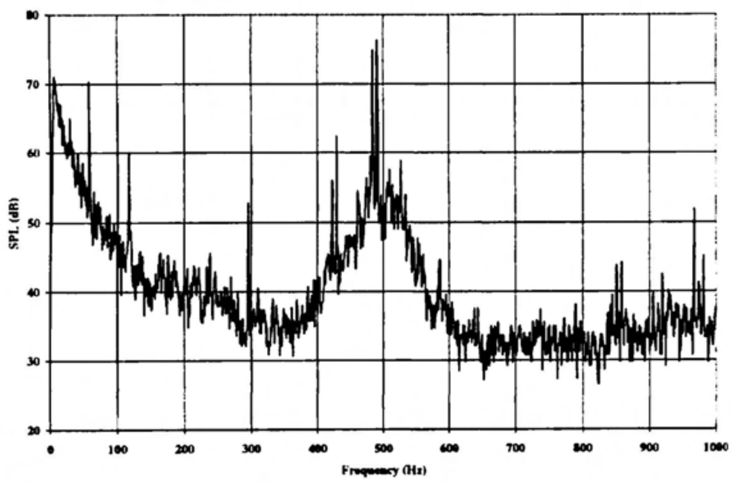
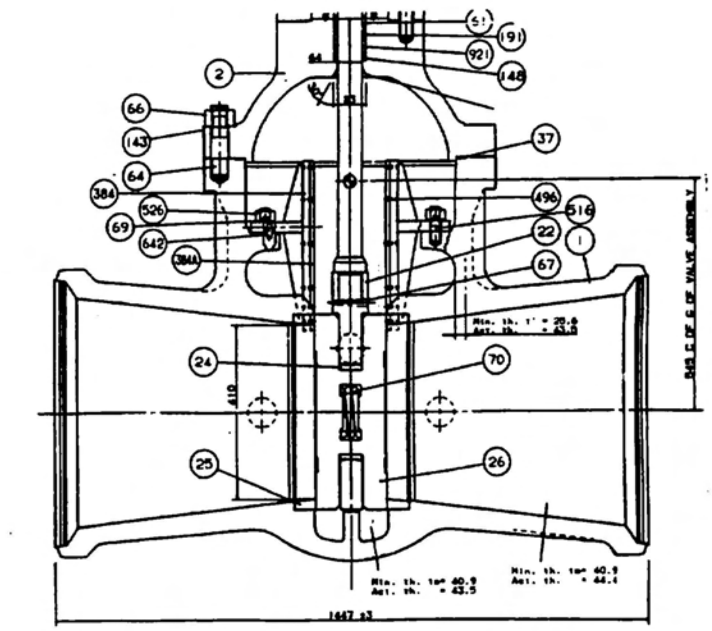


FIGURE 1: NOISE SPECTRA TAKEN FROM THE ROOF OF THE MAIN STEAM LINE METER ROOM



Legend: 2-Bonnet, 25-Seat, 26-Disk, 384-Disk Guide   Dimensions in mm

FIGURE 2: DRAWING OF MAIN STEAM ISOLATION VALVE

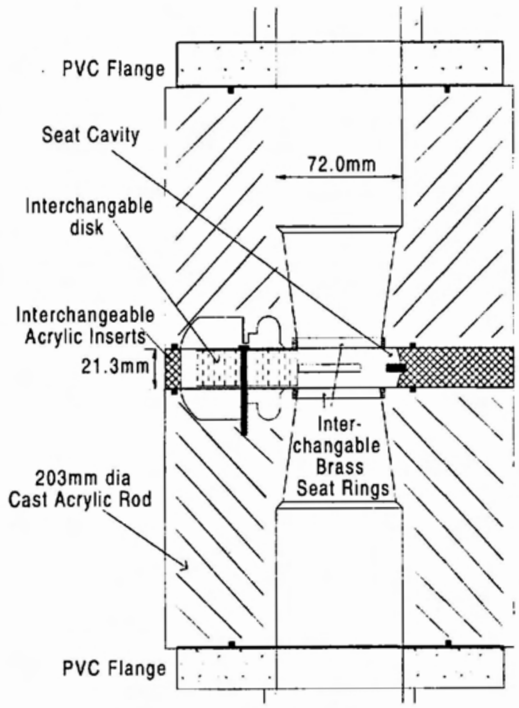


FIGURE 3: DRAWING OF VALVE MODEL

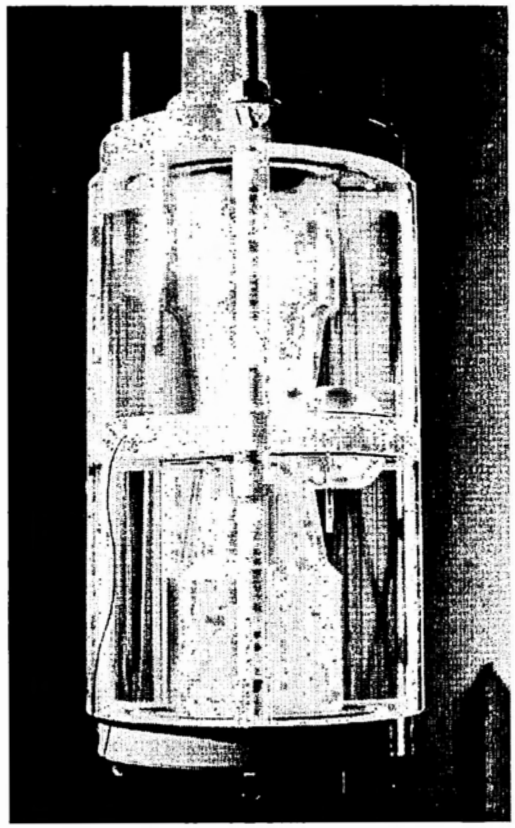


FIGURE 4: PHOTOGRAPH OF VALVE MODEL

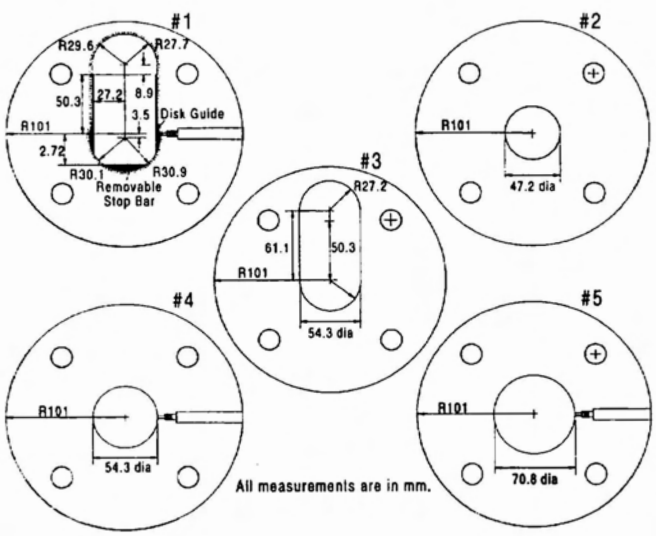


FIGURE 5: MODEL CAVITY INSERTS

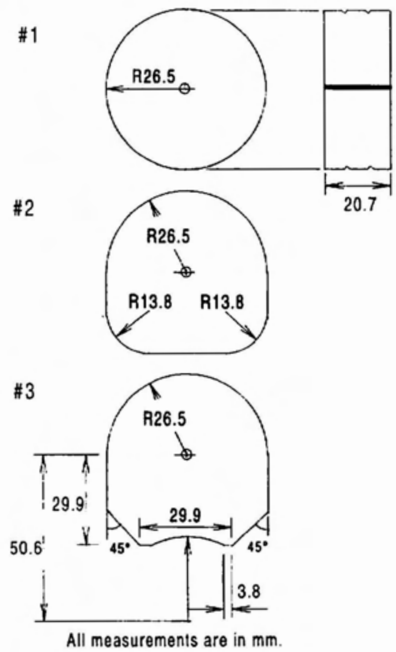


FIGURE 6: MODEL VALVE DISKS

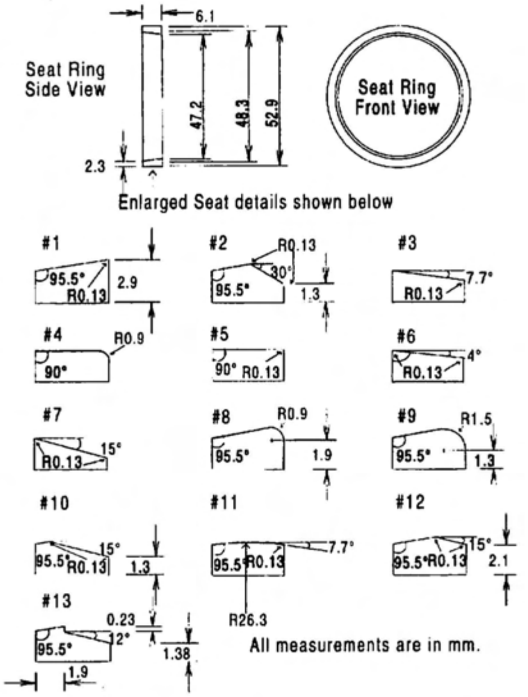


FIGURE 7: MODEL VALVE SEATS

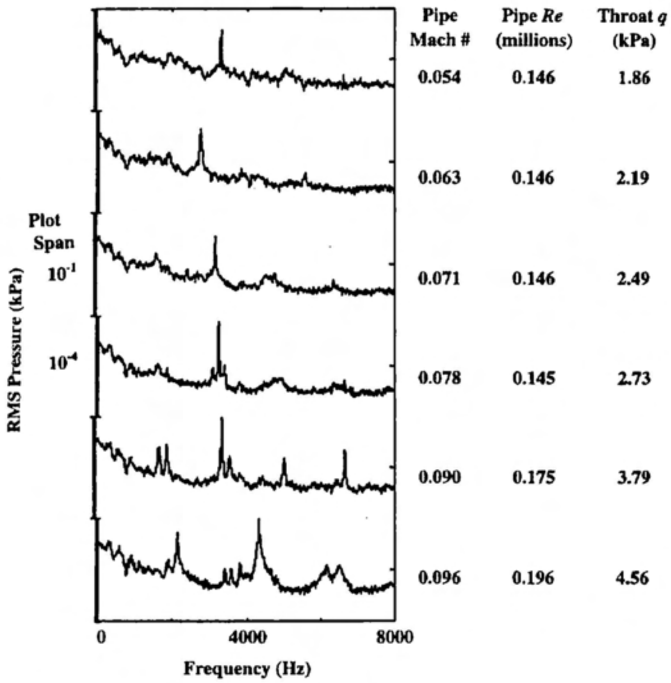


FIGURE 8: PIPE PRESSURE SPECTRA MEASURED WITH REFERENCE VALVE CONFIGURATION

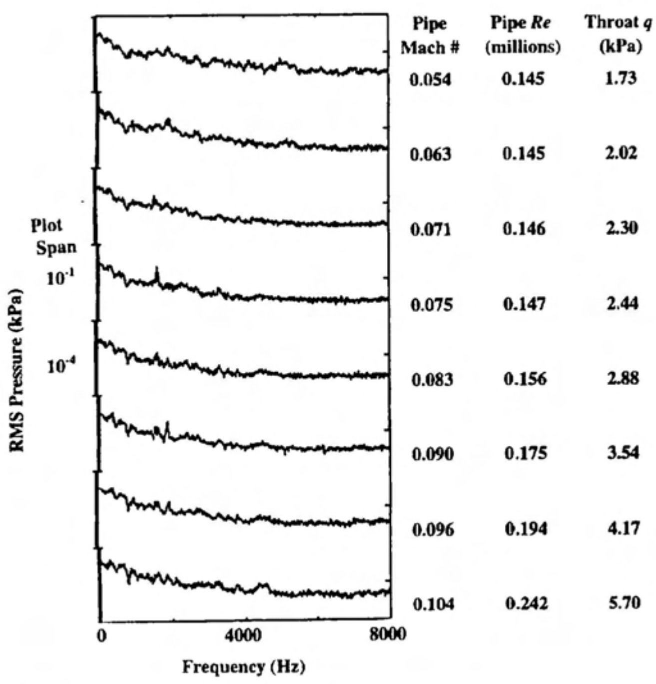


FIGURE 9: PIPE PRESSURE SPECTRA MEASURED WITH SEAT 10 UPSTREAM AND DOWNSTREAM

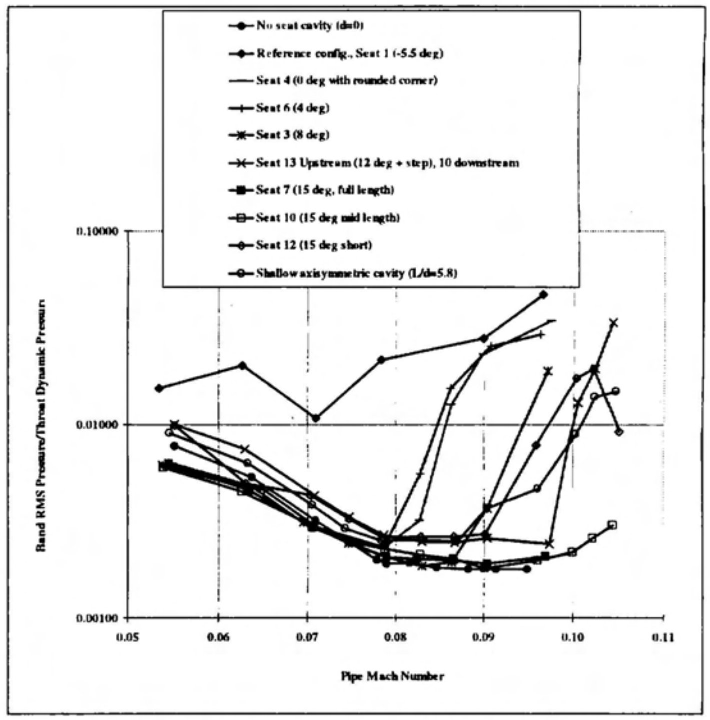


FIGURE 10: EFFECT OF MACH NUMBER ON NORMALISED BAND RMS PRESSURES IN PIPE: SELECTED RESULTS AT REYNOLDS NUMBERS LESS THAN 0.25 MILLION