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Experimental study of pressure fluctuations and flow perturbations in air flow through vibrating pipes

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ABSTRACT

This paper discusses the results of an experimental study of the effect of pipe oscillations on the wall pressure field and flow rate through a metallic pipe with air flowing through it. The data presented in this paper show that the frequencies of pressure oscillations in a non-oscillating pipe are identical to the natural structural modes of the pipe suggesting the influence of structural properties on the fluid dynamics of the flow. The results presented in this paper also show that the wall pressure undergoes both a temporal as well as a spatial oscillation if the pipe is forced to oscillate periodically. The pressure oscillations are found to be harmonics of the pipe oscillations. There is a drop in the mean pressure when the pipe is subjected to periodic oscillations. The flow rate through the pipe is seen to undergo a periodic change over a range of almost 7 percent variation when the pipe is oscillated. The study presented in this paper elucidates the dominant effect of system dynamics on determining the flow behavior through a rigid pipe. The adverse effect of flow oscillations, induced by pipe motion, can lead to departure of the flow from the intended design conditions and can render the fluid supply system inadequate.

1. Introduction

Vibration or agitation conditions imposed on supply system lines and associated components is probably the most ignored design consideration that could render as inadequate an otherwise good system. Pipes for carrying fluids are present in various industries and vehicles. Vibrations in pipes are excited by external excitations and/or flow perturbations caused by pumps and by turbulent perturbations [1]. These pipe vibrations may cause severe alteration to the flow, causing periodic transition to turbulent flow and re-lamination as reported by Hino et al. [2] in a purely oscillating pipe flow. If the pipe carrying the fuel starts to vibrate at certain frequency and amplitude, it may lead to pressure oscillations in the pipe, giving rise to an unsteady fuel flow rate.

The studies on fluid–structure interaction with emphasis on the structural effects of a piping system under vibration conveying fluid or such kind of problems involving structural aspects of a fluid carrying pipeline are aplenty. Paidoussis [3] has discussed the phenomenon of flow induced vibration of pipes conveying fluids and has provided detailed analytical framework for studying such class of problems. He has also summarized various experimental studies related to flow induced oscillations in pipes (internal flows) and slender bodies (external flows). Paidoussis and Li [4] have considered the problem of a pipe conveying fluid a model dynamical problem due to its rich dynamic behavior. They have discussed the dynamics of pipes with supported ends, cantilevered pipes, continuously flexible pipes, incompressible as well as

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compressible fluid flows through pipes with steady and unsteady velocities considering linear, nonlinear and chaotic dynamics. Wang and Ni [5] have shown that the dynamics of a standing pipe conveying fluid is much richer than the hanging system. Qiao et al. [6] have modeled a curved pipe conveying fluids, restrained with nonlinear constrains and have studied the possible existence of chaotic motions. They have shown the presence of chaotic motions in certain ranges of parameter space via doubling-period bifurcations. Jin [7] has studied the dynamics of a cantilevered pipe conveying fluids with the motion-limiting constrains and a spring support. He has concluded that the system is fairly stable at low flow velocities. However, at larger flow velocities, the pipe loses its stability either through divergence or through flutter. He has also shown that chaotic motions can occur in this system in certain range of parameter space. Paidoussis and Moon [8] have investigated the dynamics of a flexible cantilevered pipe conveying fluid with motion-limiting constrains and have reported that with increasing flow velocity, beyond the Hopf bifurcation, there are regions of period-doubling and chaos. Panda and Kar [9] have studied the internal resonance of nonlinear pipes conveying fluids with harmonic velocity pulsations and have reported that such a system can exhibit principal parameter resonance of second mode and combination parametric resonance due to excitation related to flow pulsations. Furthermore, they have reported pitchfork bifurcation, Hopf bifurcation and saddle mode bifurcation for variation of control parameters.

Significant amount of work has been carried out on pulsatile pipe flow in which the axial velocity component has an unsteady, often periodic component imposed on it [2,10,11]. Ahn and Ibrahim [12] have numerically simulated oscillating gas flow inside Stirling engine heat exchangers. Mishra et al. [13] have solved the incompressible Navier–Stokes equation in two dimensions in the vorticity-streamfunction formulation for the unsteady flow of an incompressible viscous fluid in the entrance region of a parallel-sided channel whose walls pulsate in a prescribed manner where the walls move symmetrically about the central axis. They have shown that for large values of frequency of pulsation of the channel walls oscillating sinusoidally with time, back flow occurs near the channel walls at a certain location far downstream from the entrance section at a certain instant of time. It is also found that at a subsequent time there is flow reversal at the same location across the entire cross-section of the channel. However, they have shown that for sufficiently small values of frequency, such back flow does not occur. Kehr et al. [14] have numerically simulated a fluid–structure interaction problem with significant focus on the effect of fluid of a moving structural boundary as in a cross-connection control valve. They have shown that the movement of the structural components, which were treated as moving boundaries in computation, has a large bearing on the fluid medium. Mateescu and Venditti [15] have developed techniques for effectively simulating flows with moving/oscillating boundaries using time dependent co-ordinate transformation and usage of artificial compressibility thereby solving the moving boundary problem in a static domain. Mateescu et al. [16] have used similar formulation to the simulation of laminar three-dimensional flows between annular passages with the outer cylinder oscillating and the inner cylinder remaining fixed in cylindrical coordinate systems. They have reported that the nonlinear unsteady fluid forces calculated with this method depend on both the displacement and the velocity of the oscillating structural boundary, in agreement with physical reality, but in contrast with the linear ones obtained by mean-position analyses based on the small amplitude assumption, which are only velocity-dependent. Frendi [17] has numerically investigated the effect of structural oscillation on fluid dynamics using a fully coupled model that solves the unsteady flow equations as well as the dynamic equations of the structure. In all the flow regimes studied, i.e., laminar, transitional and turbulent, he has reported that the structural vibration introduces significant changes in the flow field. While investigating the flow geometry of a backward facing step with an immediately following oscillating bottom wall, the size and shape of the various re-circulation bubbles is found to be strongly affected and oscillate at the same frequency as that of the structure. In addition, the reattachment point oscillates at the same frequency as that of the structural vibrations. Benhamou et al. [18] have carried out experimentations to visualize the developing flow (transition to turbulence mainly) in a pipe forced to oscillate in a horizontal plane. The visualization results shows that pipe oscillations induce a secondary transversal flow as indicated by the helicoidal form of the streak lines. At relatively high oscillation frequencies, the intensity of this transversal flow increases and may destabilize the flow structure. In fact, large mixing is induced in the flow and leads to the development of turbulent vortices even when the flow Reynolds number is lower than its critical transition value for a stationary pipe ($Re<2000$). This earlier transition to turbulence, attributed to the Coriolis force, occurs at lower values of Re as the oscillation Reynolds number, indicated as $Re\omega$, increases. The flow structure can become fully turbulent if Re is further increased. They have developed a chart indicating the limits of the transition and full turbulent flow structures. Hino et al. [19] have experimentally studied oscillating turbulent pipe flows for various Reynolds numbers and frequencies. They have reported that the turbulent energy production becomes exceedingly high in the decelerating phase, but the turbulence is reduced to a very low level at the end of the decelerating phase and in the accelerating stage of flow reversal.

As the fluid flow over a structure is known to give rise to phenomenon like flutter, it is but imperative that structural vibration will alter the flow field inside fluid filled piping systems and, sometimes, the changes can adversely affect the performance of a fuel feed systems. It can alter the mean fuel flow rate and can give rise to fuel flow oscillations, which can give rise to combustion instability in engines. This paper describes an experimental study of the effect of pipe vibration on the flow properties of air flowing through a pipe in an attempt to quantify the fluid mechanics aspect of fluid–structure interaction in a vibrating pipe system. It should be pointed out that unlike most of the previous relevant studies in this field, this study uses air as the flowing fluid through the pipe instead of a liquid. Owing to the compressibility of air, an additional damping term is included in the fluid mechanics of the flow and is more complex than a liquid flow through the pipe.

2. Experimental procedure

A schematic illustration of the experimental setup used in this study is shown in Fig. 1. Compressed air, at known and regulated pressure, was introduced into a stainless steel pipeline, having an internal diameter of 9.5 mm, through a calibrated rotameter that measured the flow rate of the air. The air supply pressure was controlled using a pressure-regulating valve and the airflow rate was controlled using a needle valve located after the rotameter. The volume flow rate fluctuations were measured using a thermal flow meter (measurement range of 0–50 LPM). Just before the entry to the pipe examination section through flexible plastic tubing, a robust AutoTran® model 860 piezo-resistive pressure sensor was placed. The purpose of usage of this sensor was to have knowledge of the pressure in the line before the flow enters the main pipe section, so that the sensitive sensors downstream could be protected from overpressure. Six calibrated Kulite® pressure sensors (Model XCS-062, with build in signal conditioner, Full scale: 0.35 bar, Resolution: infinitesimal, Natural frequency: 150 kHz, Acceleration sensitivity: 1.5×10^{-3} percent of FS/g, Pressure sensing principle: fully active four arm wheatstone bridge, dielectrically isolated Silicon on Silicon) were used to measure the wall static pressure at six locations along the length of the pipe, placed at equal distances with each other, as shown in Fig. 2. The pressure sensors were mounted on a vibration isolated stand and fixed at the end of flexible rubber tubes of 0.2 m lengths connected to the pipe so that they can be protected against the vibration of the pipe from being transmitted to them. Enough margins were given to these flexible tubes so that they do not stretch or compress when the pipe was set into motion and contaminate the measurements. The pipe was clamped at the middle to an electromagnetic shaker driven by a power amplifier. The input to the amplifier was given from a function generator that provided sinusoidal DC input to the shaker. The displacements were tracked using a calibrated Keyence® displacement sensor (Model LK-2503 CCD LASER displacement sensor, linearity ± 0.1 percent of F.S., resolution 1-μm). A general quantification of the displacement is given in Table 1.

The flow sensor and the inlet Auto Tran pressure sensor were connected to a PC via a National Instruments Data Acquisition (DAQ) card. The Kulite sensors and the displacement sensor were connected to another PC through another DAQ card. In each PC the data were acquired using an eight channel, 16 bit data acquisition card into a computer using LabVIEW® data acquisition program. The data from all the sensors were acquired simultaneously with a typical scan rate of 200 samples/s for 10 s. The stored data were processed using MATLAB® on a personal computer. The tests reported in this paper were carried out for an air flow rate of $3.3 \times 10^{-4}\,m^3/s$ (20 LPM, which is almost at the middle of the measurement range of the thermal flowmeter leaving enough room to accommodate oscillations) through the pipe, which corresponds to a flow Reynolds number of 3020 based on the pipe diameter. The pipe was oscillated at different combinations of amplitude and frequency and the effect of these oscillations on the wall pressure and the flow rate were studied.

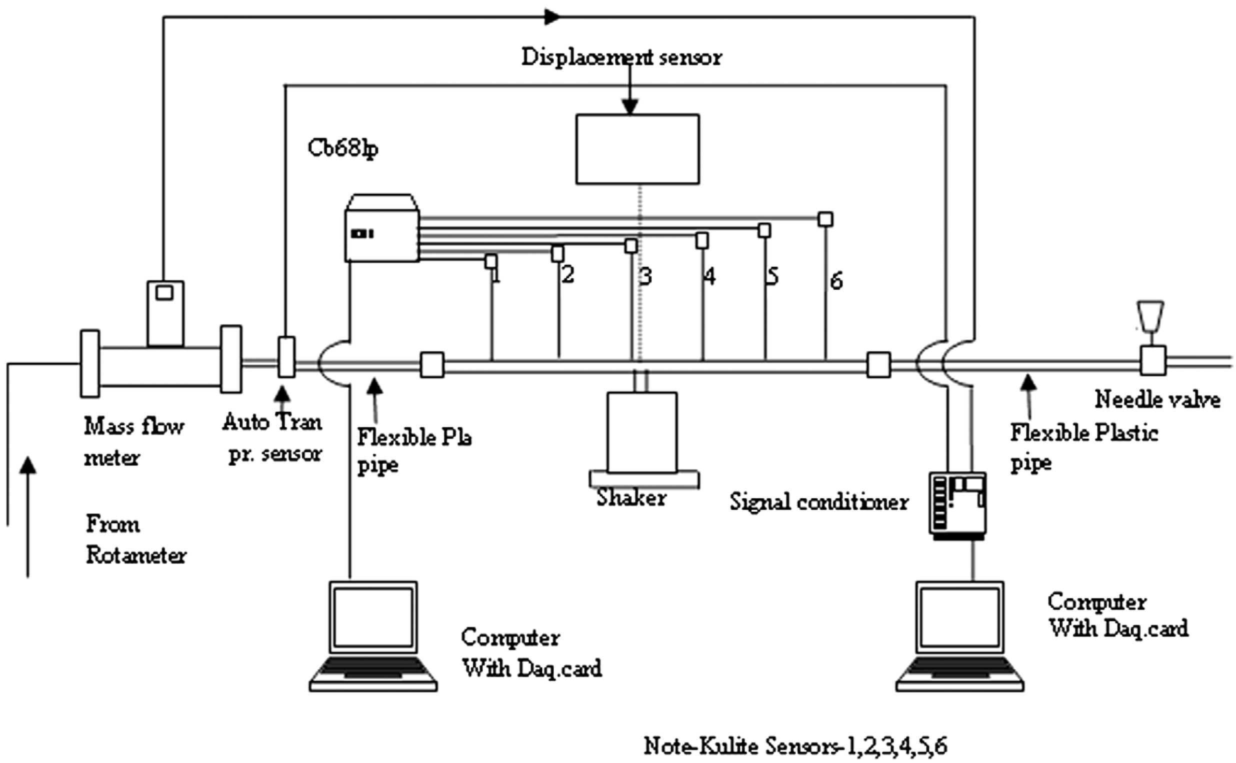

Fig. 1. Schematic of the experimental setup.

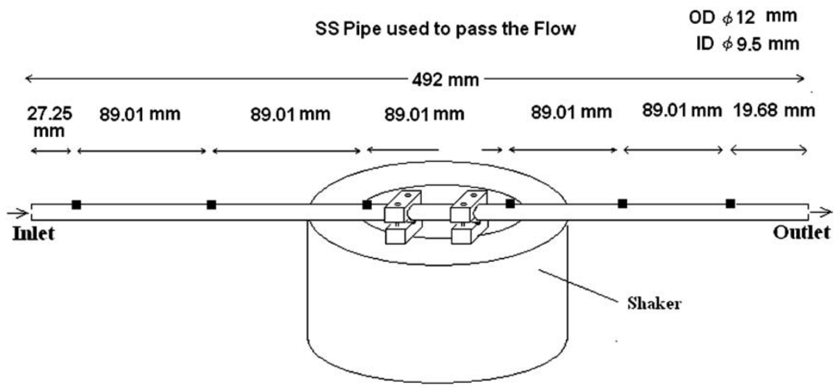

Fig. 2. Pipe dimensions and measurement port locations.

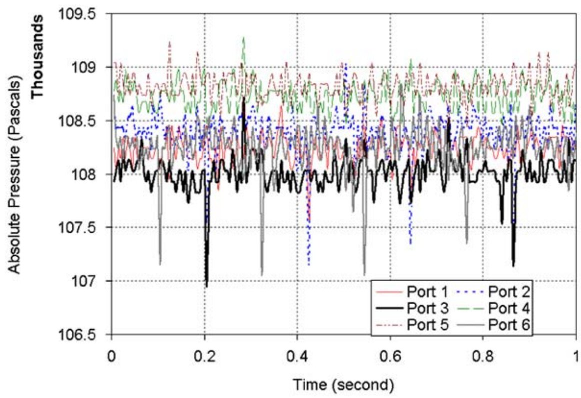

Fig. 3. Temporal variations of wall pressure at different port locations for unforced pipe.

All the pressure sensors were calibrated by connecting the sensors to a small chamber filled with compressed air. A mercury manometer was used as the standard sensor. The calibration curve was linear for all the sensors with R^2 value >0.99. In order to estimate the measurement errors and uncertainties in the mean pressure, 10 000 pressure readings were taken at a given chamber pressure and the standard deviation was found to be <0.03 percent, with <1 Pa measurement uncertainty (with 95 percent confidence level) [20].

3. Results and discussions

This study focuses on the effect of vibration amplitude and frequency on the wall pressure field inside the pipe and the corresponding effect on the flow rate. Both the average and transient behaviors were measured and the data are presented in this section.

3.1. Pressure oscillations without forced pipe oscillation

While looking at the results of experiments in general, it is very important to keep in perspective the results obtained in the no-oscillation case. The results of the no-oscillation case serves two very important purposes, namely, providing the basis for comparing the results obtained with oscillations and pointing out the importance that inherent structural properties of the pipe will have in dictating the results as obtained with oscillations of the pipe. Fig. 3 shows the general temporal variation of pressure across the various pressure scanning ports along the length of the pipe. The above pressure distribution shows up a very interesting phenomenon. Though in general, there is not much variation in the average values of the pressures noted at various ports, there are pressure oscillations clearly visible at all the ports. The observation is augmented by a fast Fourier transform (FFT) analysis of the oscillating component of the pressure data. In plotting of the FFT spectrum, a 5-point moving averaging has been done and a plot of the same is given in Fig. 4. There are a couple of features that are very prominent in Figs. 3 and 4. Firstly, the peaks occurring for the pressure values of the port 6 are above the peaks of every other ports, which indicates larger values of oscillating component for the port 6, even though a plot of

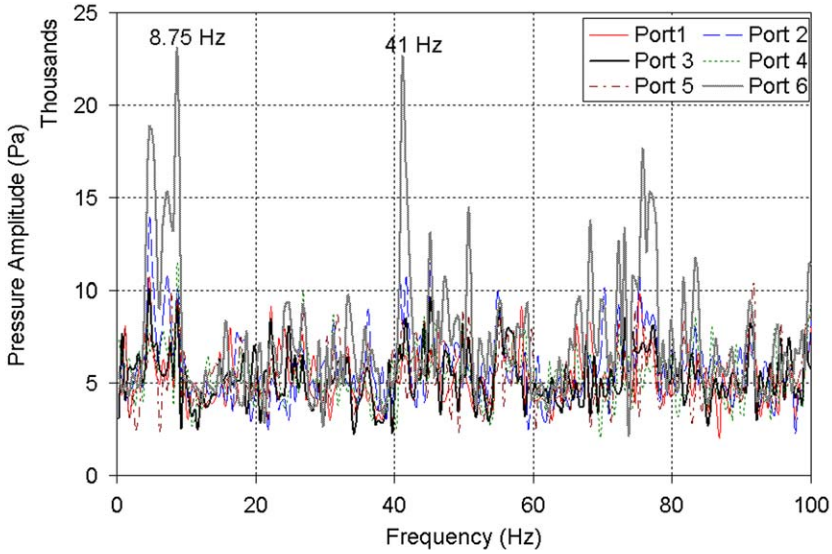

Fig. 4. Frequency spectrum of the pressure oscillations for the unforced pipe.

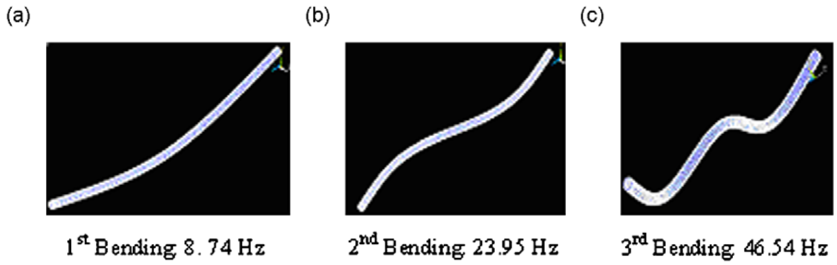

Fig. 5. Simulation results for the natural mode shapes of the pipe assuming free–free boundary condition.

the pressure values at any time instant along the length of the pipe shows that the maximum values of pressure is occurring at port 5 as seen in Fig. 3. The other very important feature that can be observed from the FFT plots is the spectral distribution of the peaks. The largest peak for port 6 occurs at a frequency of 8.75 Hz and another major for port 6 occur at around 41.25 Hz, which can be considered to be a harmonic of the previous one. Other than that we observe peaks around 46 Hz and then another at around 75 Hz. The peaks for other ports are also seen to be clustered around those above-mentioned values. Since there is no oscillation of the pipe in this case, one would tend to think that these pressure oscillations must have been reflecting either the electronic noise in the measurements or the inherent structural behavior of the pipe. A determination of fundamental frequency of the pipe with the same material and dimensions as outlined in the experimental set up descriptions were carried out using ANSYS® (assuming a free–free boundary condition, because the stainless steel pipe is much stiffer than the flexible tube supplying the air) and the mode shapes from those simulations (for half the pipe length) are shown in Fig. 5. The first few vibration modes of the pipe with given boundary conditions were obtained to be equal to 8.74, 24 and 46.5 Hz, respectively. This reveals the fact that the frequencies at which peaks are occurring in Fig. 4 are the values reflecting the first few modes of vibration of the pipe and the their harmonics at times. We would like to point out that the measurements reported in Fig. 4 show a local peak at 24 Hz but it is not that dominant compared to the other peaks. This correlation between the structural vibration modes and the pressure oscillation frequencies establishes an interaction between the structural vibration and the induced flow perturbations and this interaction was further illustrated by studying the flow perturbations in a pipe undergoing forced oscillation induced by the electromagnetic shaker. It should be noted that the study reported in this paper is primarily experimental with the focus on the flow behavior under vibratory agitation, and hence, details of the structural response study is not elaborated.

3.2. Effect of forced oscillations

Now one can have a general look at the pressure field when the pipe was oscillated at regulated frequencies and amplitudes. The pipe was oscillated sinusoidally at frequencies ranging from 4 to 32 Hz, including both the values, in steps

Table 1
Quantification of pipe displacements for various excitation amplitudes (voltage).

Actuation voltage (V)	Actual vertical traverse (twice of amplitude of motion) (mm)
4	1.40
6	1.55
8	1.70
10	1.85
12	2.00

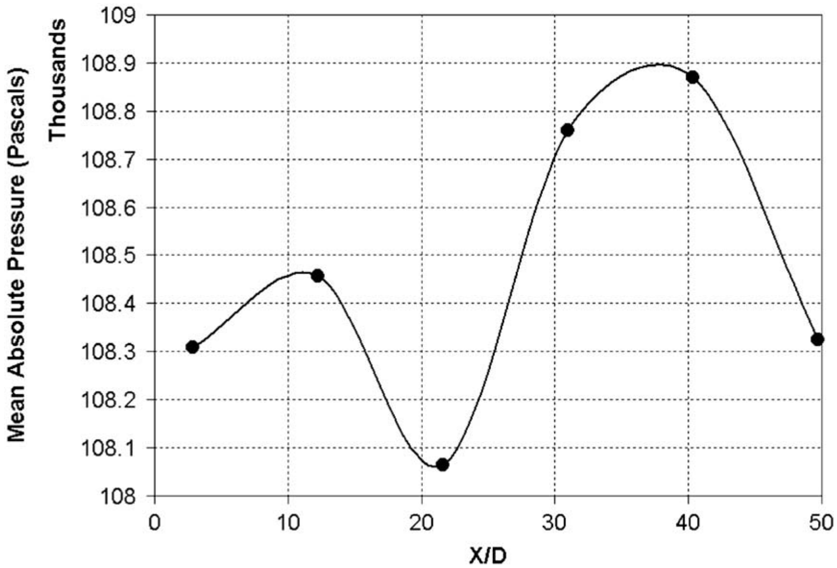

Fig. 6. Spatial variation of mean pressure for a forced sinusoidal motion of 4 Hz with 4 V excitation.

of 4 Hz. These variations in frequencies were carried out for fixed amplitude of DC voltage input to the shaker of magnitudes starting from values of 4 to 12 V in steps of 2 V, which corresponds to variation in the magnitude of oscillations. A general quantification of the displacement is given in Table 1.

3.2.1. Dynamics of forced oscillations at a given frequency and amplitude

First let us consider distribution of pressure at various ports at a frequency of 4 Hz. Fig. 6 shows the axial variation of average absolute pressure when the pipe is oscillated with a sinusoidal signal having a frequency of 4 Hz and excitation amplitude of 4 V (as given in Table 1). It can be observed in Fig. 6 that the pressure has a sinusoidal spatial distribution with pressure nodes at the inlet and exit of the pipe as well as at the center of the pipe. The pressure maximum occurs at the 5th port which can be attributed to the axial momentum convection due to the axial flow velocity, whereas the sinusoidal behavior of the pressure distribution is induced by the pipe oscillation through the transverse flow momentum.

The data shown in Fig. 7 shows the probability density function (PDF) of the pressure transients and depicts the general trends in the pressure oscillation patterns at all the port locations. It can be seen that the PDF of the pressure oscillations have a minimum value at port 6 with the maximum standard deviation (176. 2 Pa). This implies that the pressure at port six has maximum oscillations over a wider range of pressure. The higher oscillations can be attributed to the maximum displacement of the pipe at that location, as seen in the mode shapes shown in Fig. 5, which results in higher transverse acceleration of the flow. Therefore, the transverse acceleration can be considered to be the driving factor behind the pressure oscillations in vibrating pipe flow. Due to this feature of the pressure transients at port 6, most of the transient measurements of pressure reported henceforth are for this port.

Fig. 8 represents the temporal oscillations in wall pressure at port 6, when the pipe is excited with 4 Hz and 4 V excitation and its correlation with the measured displacement of the pipe at the clamping location. The data in Fig. 8 shows that the pressure oscillations and the displacements are typically out of phase and the frequency of pressure oscillations are more than that of the forced disturbance, which is exactly equal to the excitation frequency.

The phenomena occurring inside the flow field shows the effects of the inertia and compressibility of the fluid medium and the turbulence existing in the flow due to high velocity of the flowing fluid. Depending on the boundary condition at any given moment, the effect of one supersedes the other and thus the behavior shown in the plots are manifested. For

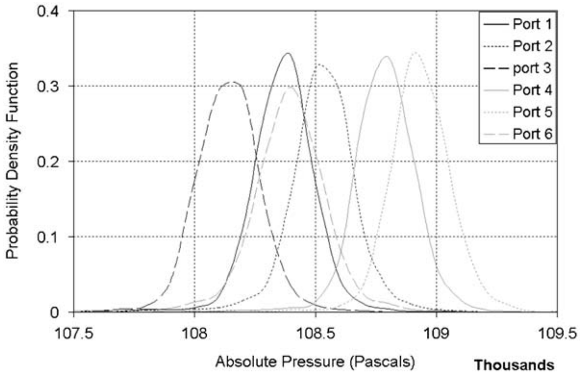

Fig. 7. Probability density function of absolute wall pressure at various port locations for a 4 Hz, 4 V excitation.

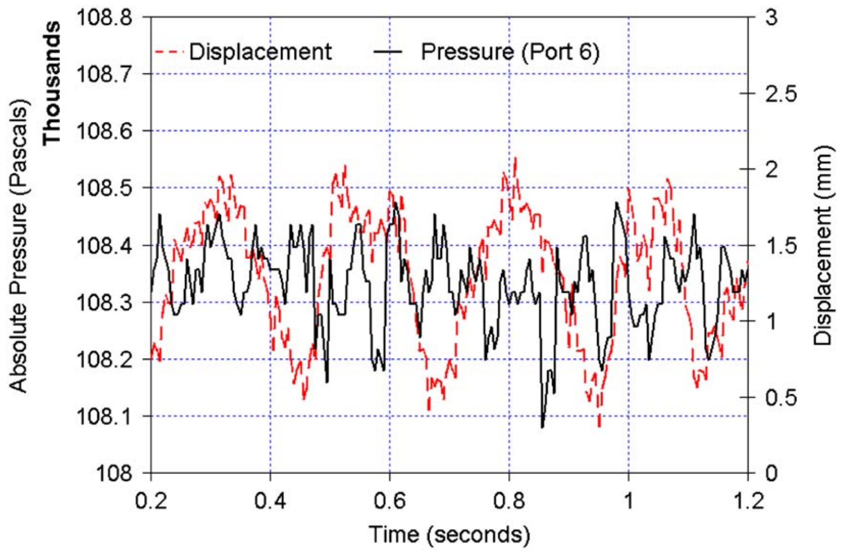

Fig. 8. Temporal variation in absolute pressure at port 6 and its correlation with the pipe displacement for 4 Hz, 4 V excitation.

example, consider the moment when the pipe starts moving upwards. The fluid layer immediately in contact with the lower half of the pipe wall will have the maximum tendency to move and there would be created a vacuum like situation at the top/diametrically opposite side of the wall (the same effects will be played cyclically continuously at both end). The tendency to move will impart both an axial and transverse (along the line of motion of the pipe) momentum to the fluid inside. Now the transverse momentum has more obstacles against it because there is significantly less room available in the diametrical direction for the movement. So, it leads to the fluid getting compressed and thus, arise the effects of compressibility. One might argue that due to low flow speed (of the order of 4.8 m/s), the flow Mach number is quite low and the flow can be considered to be incompressible. However, there is a distinct difference between a compressible flow and compressible fluid. Air, due to its low bulk modulus, is a compressible fluid even at low Mach numbers and hence can be easily compressed through work interaction with the surroundings. The compressibility also has the effect of damping out the pressure oscillations. Whereas, the axial momentum has more room for it to maintain itself with varying degrees of difficulty depending on the position along the axis and hence the nearby boundary condition (whether this is the direction from where flow is coming in or this is the direction in which the fluid is escaping the pipe into another pipe of zero gauge pressure). Depending on which effect has more predominant role to play at a given moment for a fluid particle, the compressibility effect will raise the pressure in general and the inertia effect will try to lower the pressure in general (under many cases, the inertia will add to the compressibility as well). Added to that effect is the structural effect of how much displacement is actually taking place at any axial location of the pipe. Through fast Fourier transformation data plots at various amplitudes and frequencies, it can be seen later that the harmonics of the first and thirds modes of vibration of the pipe are more prominent in the pressure drops. The mode shapes indicate that the ends of the pipes will have the maximum displacement from the midpoint of the pipe compared to the other locations of the pipe. But the pipe inlet is more rigidly fixed to the surrounding structures in the experimental set up, so it has lesser of a chance to move and hence the outflow end will have the maximum displacement. Therefore, the cyclical effects of compressibility and momentum transport will have more impact on it and it will have more fluctuations in its values. On the other hand, the fluid particles towards the inlet will have to move against the incoming flow which will try to damp out most of the flow fluctuations taking place in there. So, the overall nature of the pressure distributions captured is basically result of the interplay of these three effects, namely, inertia, compressibility and turbulence present in the flow.

3.2.2. Effect of forcing frequency and amplitude

When one tries to intuitively predict the dominant frequency present in the pressure fluctuations, it seems logical to think that the dominant frequency must not be above a few harmonics of the, say, third mode of vibration of the pipe because frequencies beyond that will be damped out by turbulence present in the flow. The PDF of the wall pressure magnitude at the sixth pressure measurement port and the FFT variations of the pressure oscillations in the pipe at that port for different forcing frequencies are shown in Figs. 9 and 10, respectively, for displacement amplitude of 2 mm (12 V, as in Table 1). The data in Fig. 9 shows that the pressure magnitude is almost constant for the range of forcing frequencies with a very gradual downward drift in the most probable pressure with the increase in frequency. However, Fig. 10 shows that the pressure oscillations are quite sensitive to pipe oscillations and the pressure oscillations in general take place at a much faster rate compared to the oscillation frequency. However, one should expect the pressure oscillations to have same frequency as the domain motion if the interaction is linear. This factor could be attributed to the low density of the fluid, i.e. air. It can be expected that if the density of the fluid is more, then the damping effect of the inertia will be much more predominant and hence there would be lesser frequencies of oscillation of the fluid pressure. Furthermore, the fluid properties (density and viscosity) typically have a nonlinear relationship with the flow properties (velocity), which in turn has a nonlinear relation with the pressure. Therefore, the nonlinear variation of pressure oscillation with domain motion is perhaps not all that out of line. Secondly, the sensitivity of the system to the frequency of pipe oscillation could also be noticed in the shift of frequency with maximum presence. In the 4 Hz case, the most dominant frequency is about 75.25. As we consider higher values of frequencies of oscillation, there is a distinct left shift of the most dominant frequency. Thirdly, a closer look at the values of the most dominant frequencies and also the other frequencies with sharper peaks reveal that they are always some harmonics of the natural frequencies of the pipe (namely, 8.74 Hz for the first mode, 23.95 Hz for the second mode and 46.54 Hz for the third mode). Also, with increase in oscillation frequency, the value of the most dominant frequency settles around 61 Hz, which is a harmonic of the first mode of natural frequency of vibration of the pipe. This is perhaps because of the reason that the pressure oscillations are "on their own" at low frequencies, but at high frequencies of oscillations, they cannot adjust to the very quick changes in the configuration of the flow domain and hence settle for some values which is maintainable at high frequencies of oscillations and thereby would be dictated by the properties of the pipe itself. All of the above reflect a coupling of the structural behavior of the system with fluid dynamics of the flow.

Figs. 11 and 12 show the dependence of pressure PDF and frequency content of pressure transients, respectively, on the amplitude of oscillations for a 4 Hz excitation frequency. The PDF's shown in Fig. 11 show a monotonic decrease in the most probable pressure magnitude with an increase in excitation amplitude as well as a slight drop in the maximum probability suggesting a higher magnitude of pressure oscillations due to wider pressure range. The drop in mean pressure can be attributed to the increased level of transverse acceleration of the pipe. In general, there is an increase in the pressure oscillation magnitude with the increase in excitation amplitude, however, the frequency content of oscillations do not show a monotonic behavior, as shown in Fig. 12. Interestingly, it can be seen in Fig. 12(d) that the pressure oscillations corresponding to 10 V are much more subdued, with a much broader spectrum, compared to any other excitation magnitude. This may be caused by the damping of the oscillations due to an unfavorable phase relationship between the

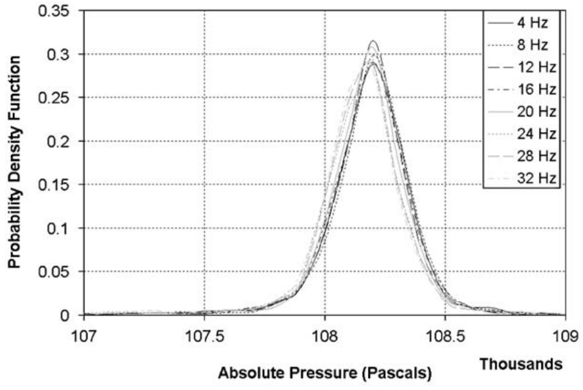

Fig. 9. PDF of the absolute wall pressure at port 6 for a 12 V excitation at different forcing frequencies.

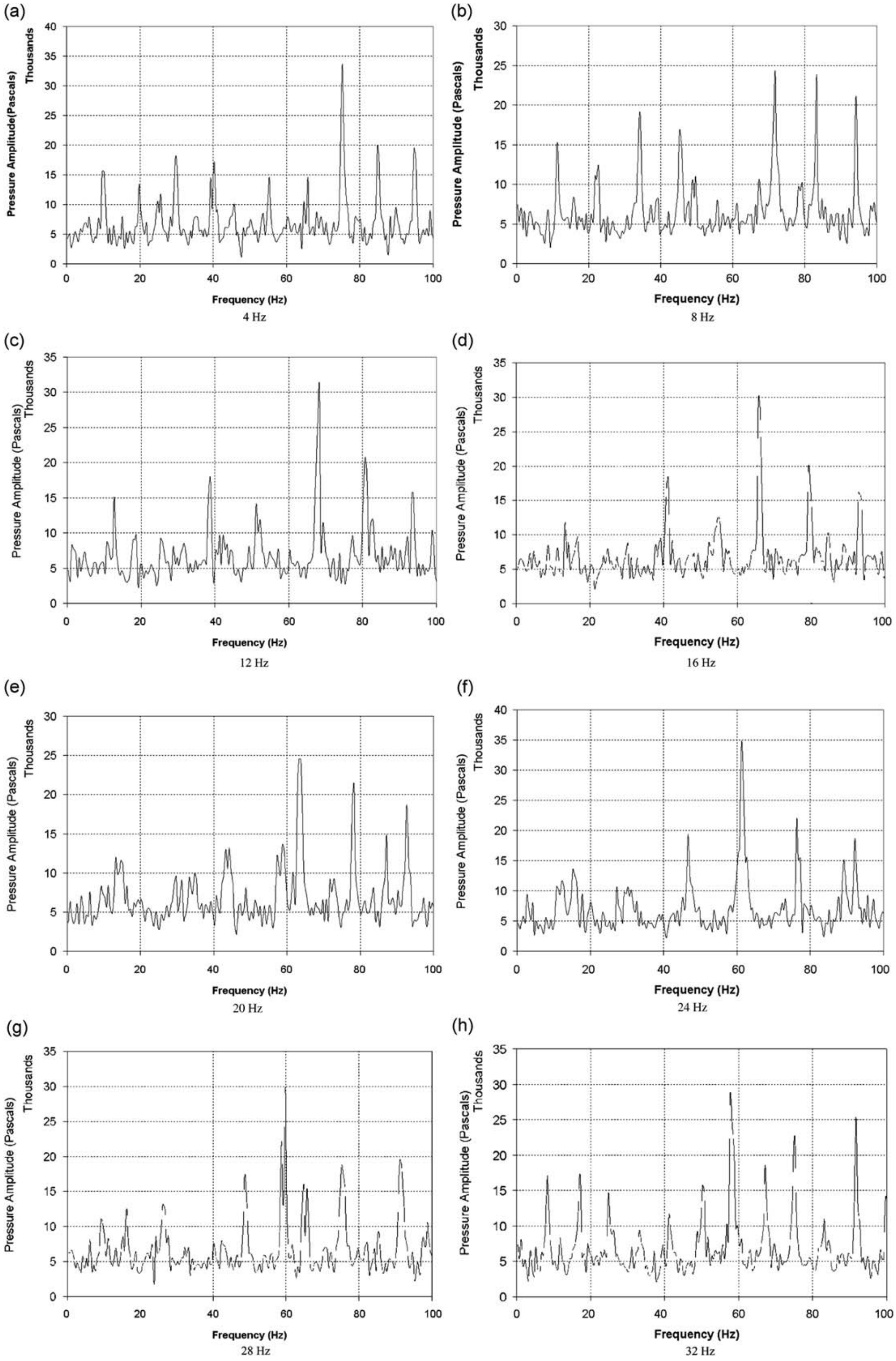

Fig. 10. Frequency spectrum of pressure transients at different forcing frequencies at port 6 for 12 V excitation.

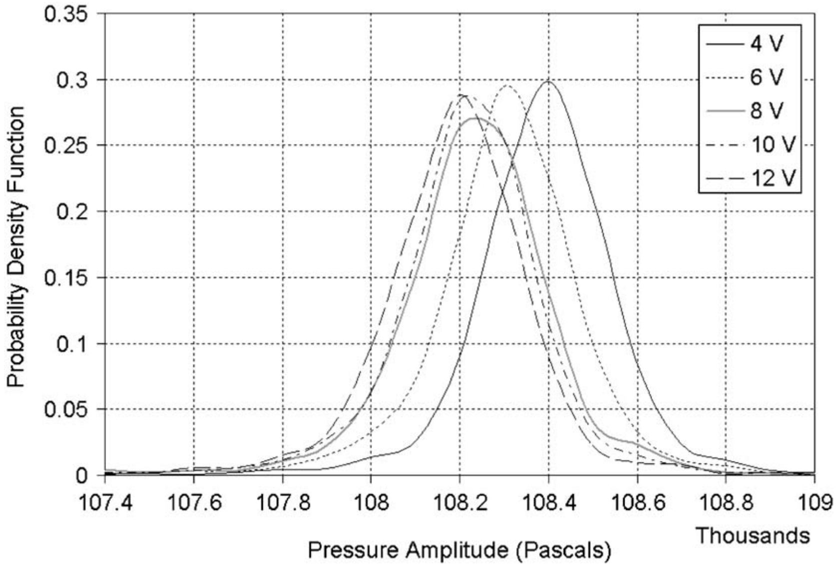

Fig. 11. PDF's of wall pressure amplitudes at port 6 for 4 Hz excitation at different excitation amplitudes.

driving mechanism (i.e., the transverse acceleration) and the damping mechanism (i.e., flow inertia, compressibility and turbulence).

Now let us consider the other effects that the changes in frequency of oscillations have on the flow behavior. The most important thing that is noticeable is that there is a steady drop in the mean values of pressure from the zero readings (i.e. no oscillations case). Now, since more often than not the occurrence of minimum pressure takes place at port 3, the effect of variation of mean pressure at port 3 on the pipe oscillation frequency is shown in Fig. 13. The data shown in Fig. 13 shows two distinct trends. There can be observed a clear drop in the mean values of pressure as the amplitude of oscillation goes up. Even though these plots are for only port 3 where minimum pressure is occurring most of the times, the above observation has been found to be true for all the other ports. When the oscillations are taking place at a frequency of 4 Hz, 4 V, the pressure values move much faster than the domain and the effect of amplitude, i.e., movement of the domain is sensed significantly by the fluid particles. As the amplitude goes up, with frequency remaining same, fluid particles have lesser and lesser time to adjust to the changes in the position of the domain and thus they adjust by following the natural tendency of the system, i.e. with a drop in mean pressure. The natural tendency of the system is to produce a drop in mean values of pressure. It seems logical to conclude that the changes in pressure in the system are basically the result of interplay of inertia (or momentum transfer), compressibility and turbulence (which always has a dissipative effect on the system). Normally, the inertia (or momentum transfer in the axial direction) reacts much faster to any changes in system dynamics and therefore the evacuating effect it has on the system is always more predominant than any other effect and that would obviously result in a fall of pressure in the fluid medium involved. The fall in pressure becomes less and less precipitous with increase in frequency as the amplitude is increased. This could be explained again in the light of the statements made in the above paragraph that with increase in amplitude, the evacuating effect of the inertia is so prominent that the progressive decrease in available time with increasing frequency is having lesser and lesser effect on the fluid particles. The significance of the evacuating effect of the axial momentum transfer is also evident from the fact that as the amplitude is increased; the average value of the pressure also goes down.

One feature of the measurements presented in this paper needs further elaboration at this point. It can be estimated easily that the flow velocity in the pipe for a Reynolds number of 3020 is equal to 4.8 m/s. Therefore, the corresponding dynamic head of the flow is equal to 14 Pa. It can be expected that under normal conditions the pressure fluctuations will scale as the dynamic head. However, the measurements reported in this paper shows pressure fluctuations of the order of hundreds of Pascals. The higher values of pressure fluctuations can be attributed to the periodic compression and expansion of the gases due to the domain movement. The movement of the pipe does dynamic work on the air inside the pipe (much like in a compressor or piston–cylinder assembly) and therefore, the measured pressure fluctuations are higher than the dynamic head of the flow.

3.2.3. Effect of pipe oscillations on flow rate

The oscillations of the pipe causes changes in the pressure field inside the pipe, which affects the velocity field throughout the system and therefore forces the incoming flowrate to oscillate as well. Fig. 14 shows the time history of the flow rate through the pipe for an oscillation frequency of 32 Hz and three actuation voltages of 4, 8 and 12 V. The incoming

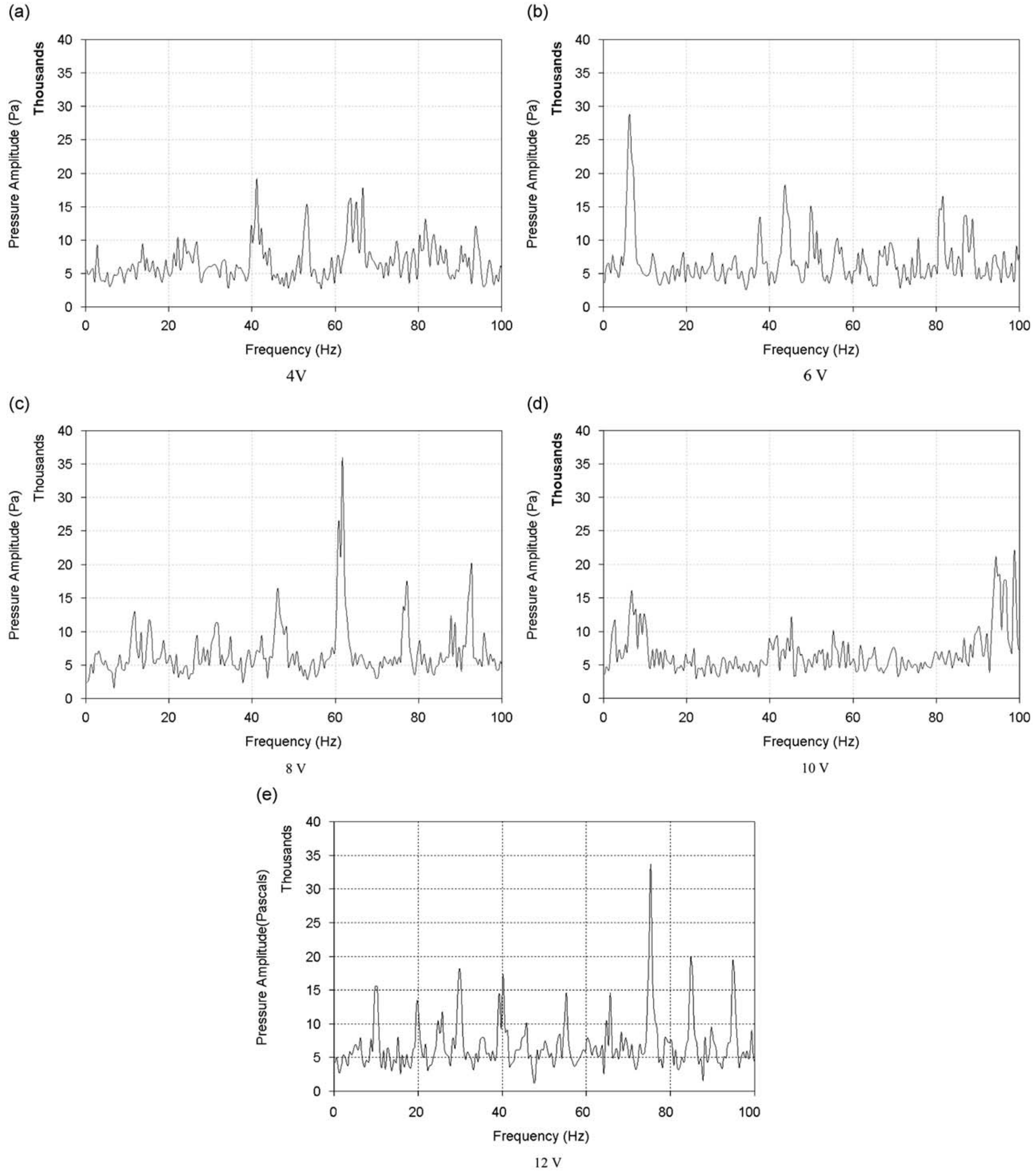

Fig. 12. Frequency spectrums of pressure transients at port 6 for a 4 Hz excitation at different excitation amplitudes.

flow oscillation have oscillation frequencies much less than the pipe displacement frequencies, as seen by the large period of oscillations in Fig. 14, because the incoming flow is energy assisted at a location not far away from the inlet to the pipe and therefore the incoming flow can damp out on its own the velocity fluctuations that are transmitted to the inlet regions. The corresponding PDF's for the flowrate fluctuations and their statistical details are shown in Fig. 15 and Table 2, respectively. The data in Table 2 presents a drop in the mean flow rate when the actuation amplitude is increased. However, the data in Fig. 13 shows a drop in the mean pressure as the actuation amplitude is increased, which should lead to an increase in the volume flow rate. Therefore, it can be inferred that the flow rate is primarily governed by the pressure

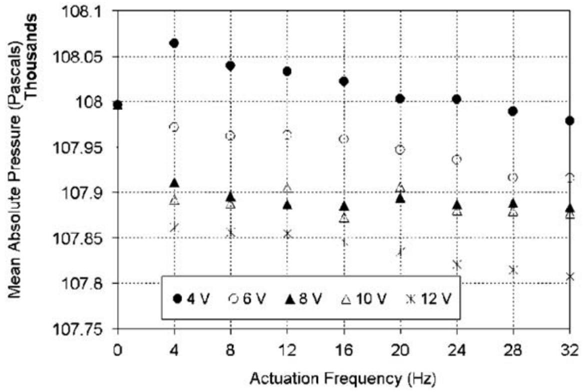

Fig. 13. Dependence of mean absolute pressure at port 3 (minimum pressure in the pipe) on excitation frequency and amplitude.

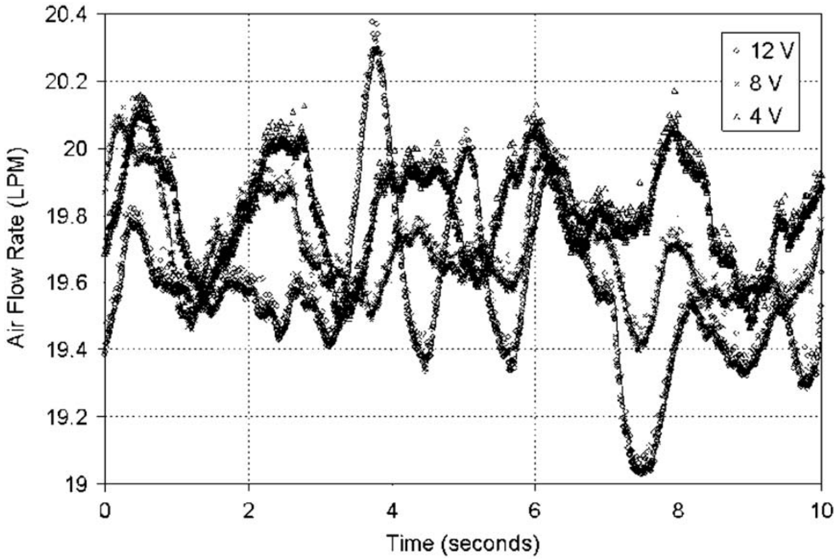

Fig. 14. Temporal variations of fluid flow rate through the pipe for a 32 Hz excitation at different excitation amplitudes.

transients, which indeed increase with an increase in the oscillation amplitude, as seen in Fig. 12. The sample variance and the standard deviation are also seen to increase (Table 2) with the amplitude, showing that the transverse acceleration causes more agitation to the flow.

The effect of actuation frequency, for a 12 V excitation, on the flow rate of the fluid is shown in Figs. 16 and 17 and Table 3. The data shows a drop in mean flow rate with the increase in actuation frequency and a corresponding increase in the sample variance. The transverse acceleration transmitted to the flow by the moving pipe increases with an increase in the actuation frequency. Thus, the fact that the flow rate reduces with an increase in the frequency once again reaffirm the notation that the flow rate is primarily governed by the pressure transients, which increase with an increase in acceleration (i.e., both the amplitude as well as frequency). It can be seen in Figs. 14 and 16 that the maximum–minimum variation in the flow rate is equal to 1.4 LPM, which corresponds to a 7 percent variation about the mean value.

The study presented in this paper elucidates the effect of system dynamics on determining the flow behavior through a rigid pipe. This is a very complex problem involving the interaction of fluid properties (compressibility, viscosity), flow properties (turbulence) and structural dynamics. The domain motion tends to drive the flow perturbations and the fluid and flow properties tries to damp out the perturbation through a nonlinear dissipative mechanism. The adverse effect of flow oscillations, induced by pipe motion, can lead to departure of the flow from the intended design conditions and can render the fluid supply system inadequate.

It should be pointed out that the study reported in this paper cannot provide complete closure to the problem of flow perturbations in oscillating pipes. Further study on this field should include the effect of fluidic damping of oscillations due

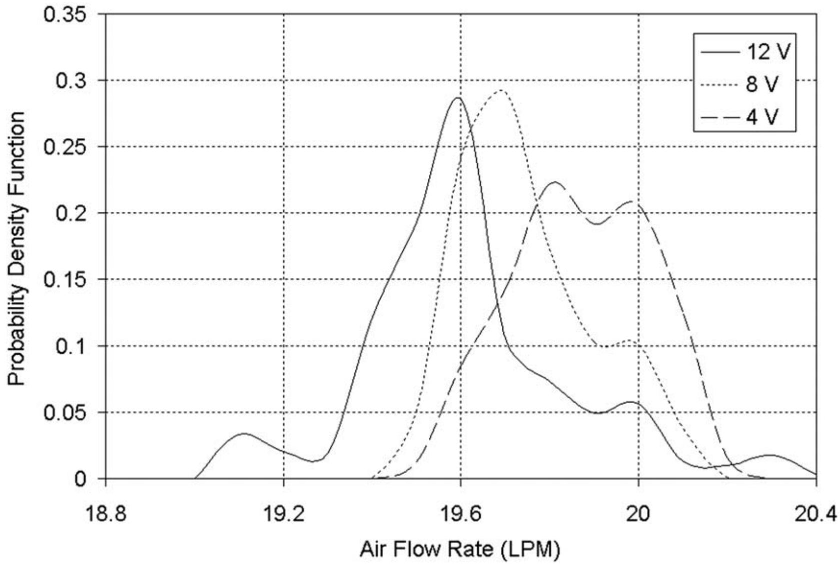

Fig. 15. PDF's of the air flow rates through the pipe when oscillated at 32 Hz and different excitation amplitudes.

Table 2
Statistical details of the flow rate oscillations for 32 Hz excitation.

Actuation voltage (V)	Mean flow rate (LPM)	Median flow rate (LPM)	Sample variance	Standard deviation
4	19.82	19.81	0.023	0.153
8	19.70	19.67	0.023	0.151
12	19.57	19.54	0.054	0.233

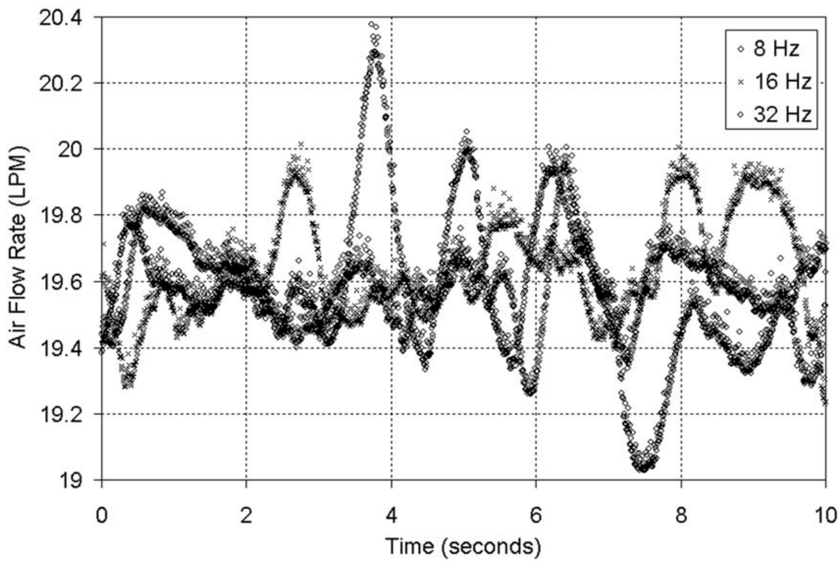

Fig. 16. Temporal variations in the flow rate for different excitation frequencies for an excitation amplitude of 12 V.

to compressibility and viscosity as well as the effect of turbulence in order to understand the nonlinear interaction between the domain motion and the fluid mechanics. A detailed study on this line can be expected to provide definitive and not suggestive explanations to the phenomena reported in this paper and lead to a proper theoretical formulation of this problem, which the present study lacks.

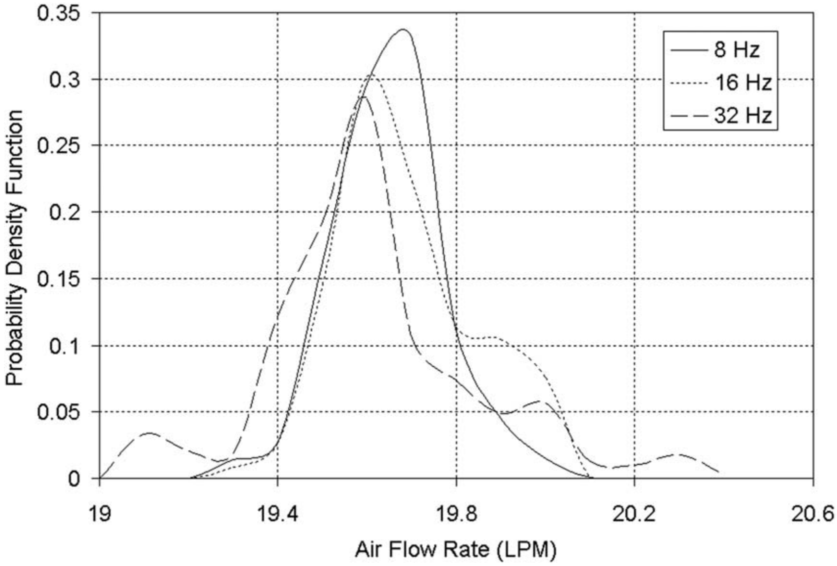

Fig. 17. PDF's of the flow rates for different excitation frequencies for an excitation amplitude of 12 V.

Table 3
Statistical details of the flow rate oscillations for 12 V excitation.

Actuation frequency (Hz)	Mean flow rate (LPM)	Median flow rate (LPM)	Sample variance	Standard deviation
8	19.61	19.61	0.015	0.122
16	19.60	19.60	0.024	0.154
32	19.57	19.54	0.054	0.233

4. Conclusions

This paper discusses an experimental study of induced flow perturbations in an oscillating pipe. The results presented in this paper show that the wall pressure undergoes both a temporal as well as a spatial oscillation if the pipe is forced to oscillate periodically. The maximum pressure drop occurs at the center of the pipe. The frequencies of pressure oscillations in a non-oscillating pipe are identical to the natural structural modes of the pipe. Under forced oscillations, the pressure oscillations are harmonics of the structural modes. The mean pressures are more at low amplitude oscillations than at high amplitude oscillations suggesting a damping effect due to fluid inertia. A drop in mean pressure with an increase in oscillation frequency further supplements this observation. The flow rate is also seen to oscillate but at a much lower frequency. It is speculated that the reported phenomenon will be more significant in case of incompressible liquid due to the absence of the damping influence of compressibility.

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