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Effect of Fin Geometry on Flow-Induced Vibration Response of a Finned Tube in a Tube Bundle

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ABSTRACT

An experimental study is carried out on a parallel triangular finned tube array with P/D_{eff} ratio 1.62 to examine the effect of fin geometry on flow-induced vibration response. Fins on a tube increase the heat transfer rate but these also affect the fluid dynamics around the tube. The flow pattern across the finned tubes is complex as compared to bare tube arrays. There are numerous parameters that affect the finned tube vibration subjected to air cross-flow in a tube array. In the current study, some of these parameters i.e. fin thickness and fin density are focused and their effects on flow-induced vibration response are analyzed in different rows of fin tube array. The current experimentation is performed in a subsonic wind tunnel using a single flexible Aluminum finned tube in a rigid array. Seven tubes with similar specifications but distinct fin thickness and fin density are used for the testing purpose. Their amplitude response suggests that the flow-induced vibration behavior is greatly affected by changing the finned tube parameters. It has also been observed during spectral analysis that the Strouhal number is independent of fin geometry since it remained constant in different rows of the array for finned tubes under study. It suggests that the vortex shedding has also contributed towards the finned tube vibration predominantly in the first, second and the fourth row of tube array.

Keywords: Finned tube bundle; Flexible tube; Mass ratio; Fin geometry; Parallel triangular

1. INTRODUCTION

One of the major concerns of process engineers, designers, and operators while dealing with heat exchangers is the vibration due to shell-side fluid flow across the tube bundles. Tubes vibrate when subjected to cross-flow of fluid and these vibrations are proportional to the intensity of flow. These vibrations produce motion of higher amplitudes causing the premature tube failure in different forms like tube fretting, fatigue and wear. Many utility industries have lost millions of dollars due to the failure of handling this vibration problem. The shutdown of different plants as a result of these vibrations motivated the researchers to examine this phenomenon caused by the fluid flow (Lever and Weaver 1986, Goyder 2002, Lumsden and Weaver 2007, Khushnood, Qureshi *et al*. 2012).

Flow-induced vibration (FIV) can be characterized by 4 different excitation mechanisms including fluid elastic instability (FEI), vortex shedding, turbulent buffeting, and acoustic resonance. Turbulent buffeting and vortex shedding may cause a slight damage in tube bundles due to flow-induced vibrations over an extended amount of time or they generate high levels of noise due to acoustic resonance. These phenomena do not cause the tube failure, but it happens due to fluid elastic instability. FEI occurs when the free-stream velocity in a cross-flow tube bundle exceeds the critical velocity and ultimately with any further rise in flow speed, the tube's magnitude of vibration grows exceptionally (Pettigrew and Taylor 2003 (a), Pettigrew and Taylor 2003 (b), Desai and Pavitran 2013, Khushnood and Nizam 2017).

The major work that is performed on FIV comprises of the plain tube's vibration exploration encountered with a single and two-phase flow of water, Air-water and refrigerant (Pettigrew, Taylor *et al*. 2001, Lumsden and Weaver 2006, Mitra, Dhir *et al*. 2009, Ricciardi, Pettigrew *et al*. 2011). Nowadays, finned tubes are used as a part of heat exchangers in preference to plain tubes and their utilization is probably going to expand more in future. Heat exchangers started using finned tubes not only for enlarging the rate of productivity but also to reduce their size as well and to do so a need to explore the effect of fins on flow induced-vibration rises. For past couple of years, investigators started exploring finned tube's vibration response subjected to air cross flow. Moreover, they have also investigated and observed the effect of type of tubes on FEI threshold Lumsden and Weaver (2010). Roberts (1966) is the earlier scientist who investigated the FEI by studying the behavior of the array of the cylinder under FIV. Connors (1970) performed experiments on a cascade of cylinders and answered about the applied fluid loading because of the cylinder's movement from its original position and gave the relationship for this phenomenon as:

$$\frac{U_{crit}}{fD} = K\left(\frac{m\delta}{\rho D^2}\right)^n \quad (1)$$

Where, U_{crit} represents critical pitch flow velocity, f is natural frequency of tube, D is tube diameter, m is mass per unit length of tube, δ is logarithmic decrement of damping, ρ is the density of fluid, and K and n are constants.

Many researchers observed contradictory effects on finned tube arrays. Kouba (1986), Nemoto (1992) concluded that fins had no essential effects on noise production in finned tube arrays. Nemoto, Takakuwa *et al.* (1997) examined the finned and bare tubes vibration response and proved that acoustic resonance in case of finned tube array is greater when compared with bare tube clusters. This difference is not unexpected because the fins are considered to affect not only the shedding phenomenon but also sound attenuation due to viscous losses in fins. These effects are seen to be directly dependent on the height and pitch of fin and on the spacing ratios among the tubes. Katinas, Perednis *et al.* (1991) examined the vibration response of smooth and finned tubes in water cross-flow. He observed that the finned tubes excited mainly due to vortex shedding. The vortex shedding frequency is identical to the natural frequency of the finned tubes. Hamakawa, Fukano *et al.* (2001) observed that span wise size of the wake of vortices is majestically larger than the pitch of fins. FEI in finned tube arrays is found by very few researchers. Roberts (1966) and Wang and Weaver (2012) considered various tube array geometries and fins impact on FEI is determined. Hirota, Nakamura *et al*. (2002) performed experiment of investigating two staggered rows of helical finned tubes and observed that the behavior is just much the same as the bare tube results of similar array geometry. The characteristics length for fluid elastic instability is concluded by them using volumetrically based effective diameter ($D_{eff.vol}$). Although the $D_{eff.vol}$ is different from the effective diameter based on the projected area ($D_{eff.area}$) of finned tube. However, the difference between the two values is 4 to 6% and for the current finned tubes under study the $D_{eff.area}$ proposed by Mair and Palmer (1975) is used.

$$D_{eff} = \frac{1}{P}\left[t\left(D_f - D_b\right) + PD_b\right] \quad (2)$$

Here, D_{eff} is the effective diameter, P represents fin pitch, D_f is fin diameter, D_b is bare tube The facts that are illustrated above describes clearly that FIV occurs not only in the confined finned cylinder but also in clusters of finned tubes. diameter, and t is fin thickness.

Concrete guidelines are yet to be defined for the design of heat exchangers containing finned tube arrays. There are few uncertain issues yet to be explored, keeping in mind that the end goal is to amplify our basic knowledge of the effect of fin geometry on the FIV mechanism in finned tubes arrays. The vibration behavior of finned tubes is greatly affected by changing fin parameters like fin material, type of fin, fin height, tube array pitch, fin diameter, fin thickness etc. Fin thickness and fin density analysis and their effect on FIV response in different rows of fin tube array is a step towards the improvement of design guidelines of heat exchangers.

2. EXPERIMENTAL SETUP

Multiple experiments are carried out in a subsonic open wind tunnel manufactured by GUNT (model HM-170). A square shape test section with 300 mm length and 300 mm width is installed in the wind tunnel. The wind tunnel draws air from surrounding that passes through prechamber of the tunnel. To make the flow laminar and parallel to the measurement section, a flow straightener is present inside the prechamber of the wind tunnel. The air flows through the test section and then blows out to the surrounding. A valve is present for changing the inflow velocity of the fluid. The wind tunnel along with its components is shown in Fig. 1.

Two sets of flexible tubes are selected for the testing purpose. There are three tubes each in both sets. All the tubes are made of Aluminum with fins being spirally winded on their axis. The first set contained tubes of similar fin thickness but various fin density, and in the second set, there are tubes with similar fin density but different fin thickness. The flexible tube is present in the center which is the fourth row from the upstream as well as the downstream side of the tube bundle. To make this tube instrumental, an accelerometer is attached on the top of the tube and held fixed using a string tightened with two steel rods passing through the tube bundles from both ends. The flexible tube is held in air by guitar strings passing through a 2.75 mm diameter hole drilled at the top and bottom of the tube and then stretched to a base support structure using nut and washer. Aluminum fin tubes under study are manufactured in a Machine tool lab in UET Taxila, Pakistan. Finned tubes along with its line diagram are shown in Fig. 2. The tube bundle which is made up of screwed acrylic plates with P/D_{eff} ratio of 1.62 is designed using the wind tunnel's dimensions. The tube array is parallel triangular and contained a total of 45 tubes, out of which one tube is flexible and rest of them are fixed in the tube bundle. There must be some sort of similarity among the fin tubes for the correlation purpose and which is maintained by keeping the mass ratio constant.

Finned tube 2 has 4.5 fins per inch i.e. less number of fins as compared to the rest of the tubes, hence its mass ratio is minimum. The mass ratio of other tubes is designed with respect to this tube and kept almost constant by performing turning phenomenon

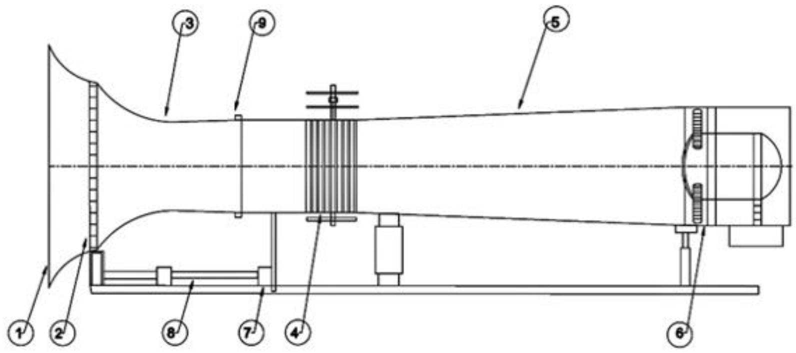


Fig. 1. A line diagram of the wind tunnel 1. Funnel, 2. Flow straightener, 3. Contraction portion, 4. Test model, 5. Divergent section, 6. Axial fan, 7. Tunnel support, 8. Length adjustable guide, and 9. Velocity measurement probes.

on the length of tube that is outside the tube bundle and is attached to the accelerometer. The natural frequency for all the tubes is also held constant at 9 Hz. Damping is a key parameter in vibrations and for the tube array under study, it is tried that only structural damping exists in the system. All the tubes in the array are fixed except the instrumental tube which is plucked and its amplitude decay is captured and analyzed using Fast Fourier Transform (FFT) of the signal to obtain the natural frequency of the tube that is held constant at 9 Hz for all the finned tubes with an error of ± 0.2. After installing the tube bundle in the wind tunnel, the inflow velocity of the wind is increased using a variable speed controller. For measuring the inflow wind velocity, a pitot-static probe is used. Acceleration signal for 39 different velocities ranging from 2.4 to 10 m/s is taken by increasing the wind speed with $0.2\ m/s$ increment.

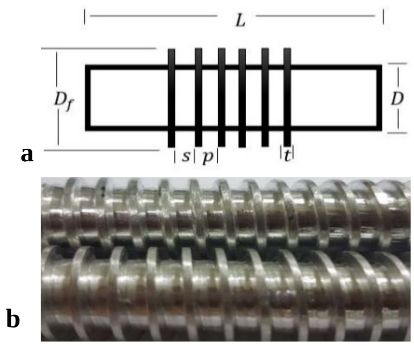


Fig. 2. (a) Line diagram of finned tube with nomenclature, (b) Finned tubes

Fig. 3. Data Acquisition System

After each reading taken, the tube is given enough time to reach the steady state and after that the signal is saved. Each signal consists of 5000 samples with sampling rate of 679 Hz. The root mean square (RMS) amplitude is determined using components along the flow as well as normal to flow. To obtain the experimental results, the signal from a calibrated accelerometer is observed on a software "Node commander" which is then analyzed for the RMS values of lift and drag acceleration using a signal analyzer software i.e. "Sigview". Figure 3 shows the data acquisition system.

3. Results And Discussions

Many researchers have performed experiments on bare tubes and observed that the vibration response especially instability is strongly dependent on tube location in a tube array in both fully flexible and a single flexible tube in a rigid array. The study of a fully flexible tube array is relatively complex because it requires many empirical coefficients. Therefore, it is desirable to reduce the complexity of the problem to have a good insight into the physics of the phenomenon under observation. The problem needs to be simplified using a single flexible tube in a rigid array since it experiences fluid elastic instability at the same threshold to that of multiple flexible tubes in an array. This idea is proposed by Weaver and Lever (1977) and used by many researchers including (Paiudoussis and Price 1988, Weaver and Fitzpatrick 1988, Austermann and Popp 1995, Price 1995, Mahon and Meskell 2009) since it greatly simplifies the study of vibration in tube array because it eliminates the stiffness mechanism that occurs between the tubes in case of flexible tube array.

3.1 Effect of Tube Location on FIV

In the first phase, the experiments are carried out on a single flexible tube in a rigid parallel triangular array and the tube location effect has been studied by placing the tube in different rows of the tube bundle. The complete specifications of the tubes are shown in Table 1. Finned tube 1 (FT 1) is installed in the first, second, third and fourth row of the tube array and observed its amplitude response. Similarly, data for other tubes in different rows of tube array is collected and observed. Figure 4 presents the FT 1 vibration behavior subjected to cross-flow in four different rows of tube array. It is observed that the amplitude increases with the flow velocity for all the locations. There is no instability observed up to 10 m/s free-stream velocity when FT 1 is in the first, second, and fourth row. The behavior of the same tube is completely different when placed in the third row. It is evident from (4c) that the tube has become unstable when the free-stream velocity is about 5.8 m/s. It is taken as the fluid elastic instability threshold. Khalifa, Weaver *et al*. (2012) reported almost the similar value for stability threshold for fully flexible tube array. When FT 1 is present in the first, second, and fourth row, the vibration amplitude increases gradually with the flow velocity but remain less than 0.2% D. The vortex shedding occurrence is not evident in the 3rd row despite the careful tuning of flow velocity.

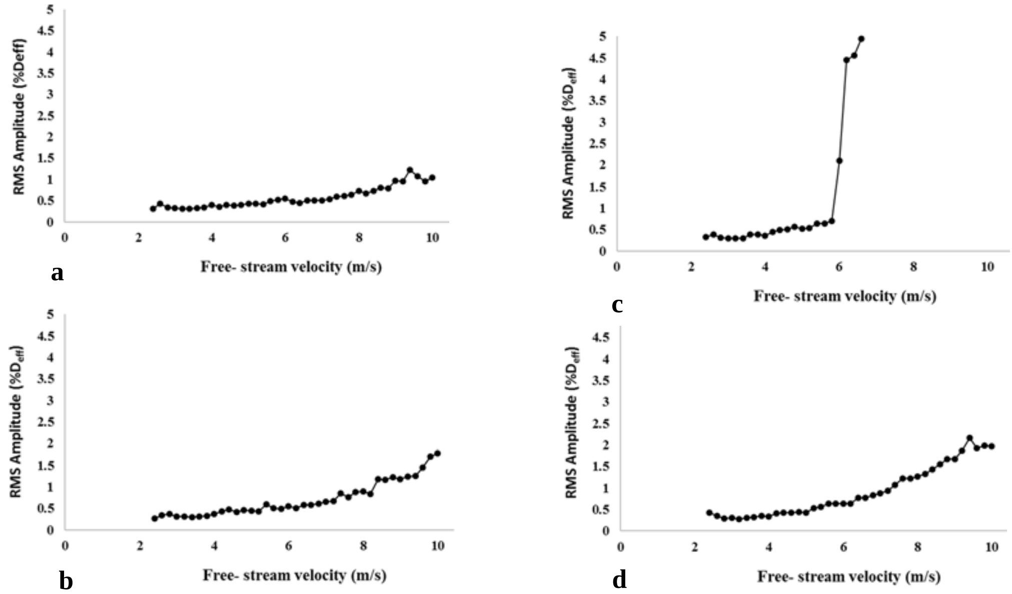


Fig. 4. The amplitude response for FT 1 in (a) first row, (b) second row, (c) third row, (d) fourth row.

Upto 5.8 m/s of flow velocity, tube experiences slight vibration mainly because of “turbulent buffeting”. After that, the tube’s vibration amplitude increases abruptly especially in the transverse direction which means that the tube has become unstable because of “fluid elastic instability”. It is concluded that tube become unstable in the 3rd row of the array in the presence of “damping mechanism” only, but it is not enough to produce instability in other rows. To ensure that this behavior is not tube specific, a similar experiment is performed with other finned tubes. The behavior is quite similar in most cases but for some tubes, the stability threshold has been delayed even in the third row. One such case is highlighted in FT 2. Figure 5 shows FT 2 amplitude response subjected to cross-flow in four different rows of tube array. The amplitude increases with the flow velocity for all the locations. The FT 2 vibration behavior resembles with FT 1 in the first, second, and fourth row and no instability is observed up to 10 m/s free-stream velocity. However, in the case of the third row, a delay in the fluid elastic instability has been observed for FT 2 and amplitude grows gradually with the flow velocity up to 9.4 m/s compared to FT 1 that become unstable at 5.8 m/s, which clearly indicate that apart from tube location the vibration behavior shows a strong dependency on the fin geometry. After 9.4 m/s, the tube’s vibration amplitude increases rapidly which suggests that the tube has become unstable because of fluid elastic instability. The effect of tube location on FEI for bare tubes is reported in the literature by many researchers, which is also a part of this study as well.

Table 1 Specifications of Finned tubes

Finned Tubes	Fin thickness (mm)	Outer Fin diameter (mm)	Inner diameter (mm)	Effective diameter (mm)	Fin density (FPI)	Fin depth (mm)	Weight (gm)
FT 1	1.30	15.85	12.85	13.70	6.0	1.5	175
FT 2	1.30	15.85	12.85	13.40	4.5	1.5	175
FT 3	1.30	15.85	12.85	13.77	6.5	1.5	175
FT 4	2.30	15.85	12.85	14.22	5.5	1.5	175
FT 5	2.30	15.85	12.85	14.34	6.0	1.5	175
FT 6	2.30	15.85	12.85	14.49	6.5	1.5	175
FT 7	1.70	15.85	12.85	14.00	6.0	1.5	175

Using the concept of the effective diameter, a comparison has been done between the bare tube and finned tube response when placed at a different location in a tube bundle array. There might be a slight discrepancy in the data due to the dissimilarity in the experimentation environment,

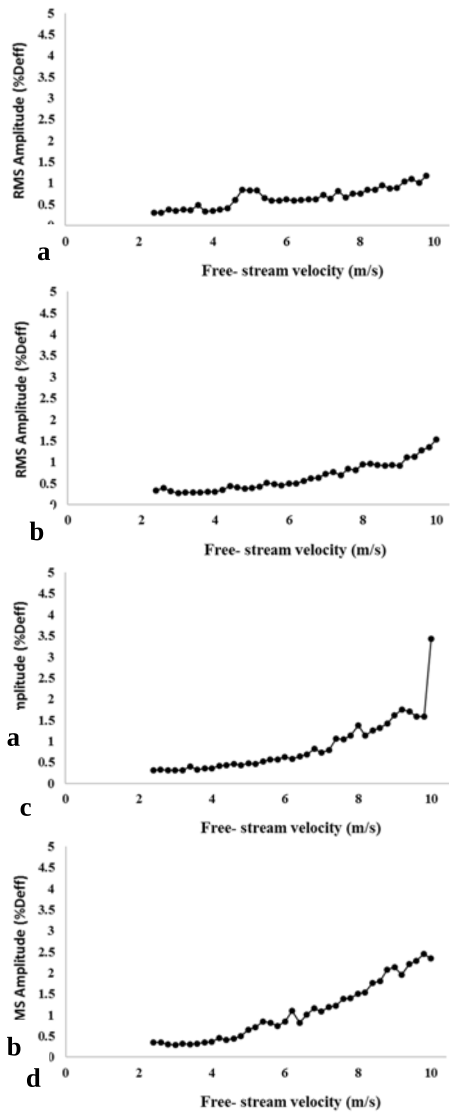


Fig. 5. Amplitude response for FT 2 in (a) first row, (b) second row, (c) third row, (d) fourth row.

the tube's material and many other factors that affect the experimentation. Table 2 presents the comparison of experimental conditions with the previous study performed by Khalifa, Weaver *et al.* (2012), in which he observed a single flexible tube's response in a rigid parallel triangular array by placing the bare tube in the second and third row. It is quite clear from the Fig. 6(a) that the amplitude response for the bare tube is less than the finned tube in the current range of flow velocity.

A similar behavior is seen in 6(b) when the tubes are in the third row. The vibration amplitude for the bare tube is less as compared to the finned tube when the velocity is low. There might be more shedding of vortices when the fluid would flow through the fins. Thus, the finned tube experiences more vibration than the bare tube at low velocity. Fluid elastic threshold starts early for bare tube and it became unstable when the free-stream velocity is about 5.4 m/s compared to the finned tube that becomes unstable at 6.0 m/s.

Table 2 Comparison of experimental conditions with the past literature

Parameters	Present study	Khalifa *et al.* (2003)
P/D_{eff}	1.62	1.54
Tube Arrangement	Parallel Triangular	Parallel Triangular
Test Section	300mm	305mm
Tube material	Aluminum	Aluminum
Tube type	Finned tube	Bare tube

Figure 6 presents a good comparison of current experimental results with the past experimentation data.

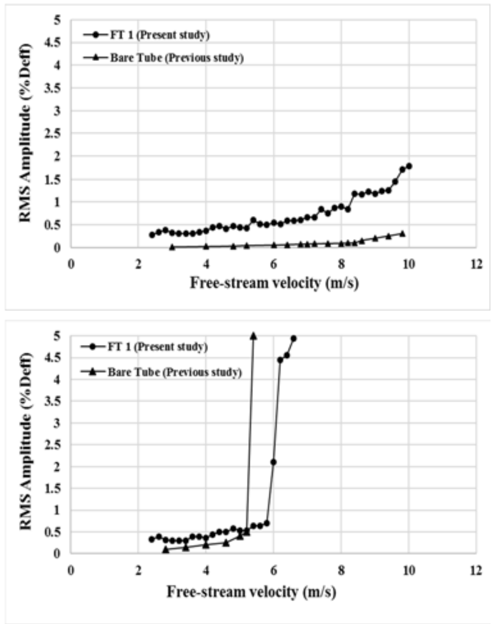


Fig. 6. Comparison of amplitude response when the tube is placed in (a) second row, (b) third row.

There is a clear relationship between the current experimentation and the previous study i.e. both the tubes become fluid elastically unstable when they are in the third with the slight difference in velocity. This difference is expected, since the tube's vibration behavior is strongly affected by the geometric parameters. To develop a better understanding of the effect of fin geometry on the tube's FIV response, a series of experiments are performed by varying fin thickness and fin density to analyze the acquired data. The analysis results are discussed in the subsequent section.

3.2 Effect of Fin Thickness on FIV

In the second phase, Experiments are carried out on tubes with similar fin density but different fin thickness (FT 1, FT 5, FT 7). Figures 7(a), (b), (c) and (d) presents the vibration behavior of FT 1, 5 and 7 subjected to cross-flow in the first, second, third and fourth row of the tube array. It is observed that the amplitude increases with the flow velocity but having distinct behavior for every finned tube. It is noticed in 7(a) that the vibration amplitude grows gradually up to 8 m/s and after that, it increases rapidly for FT 1 and 7, but FT 5 becomes unstable early when the free-stream velocity is about 7 m/s. Fluid elastic instability threshold limit reaches early for the finned tube with maximum fin thickness. Tube response indicates that the array has become unstable by a velocity of about 7 m/s for FT 5 and by 9 m/s for FT 1 and 7 respectively. The vibration amplitude of finned tube that has maximum fin thickness is almost twice than the other two tubes in the first row of tube array. Figures 7(b) and (d) shows that the vibration pattern and amplitude for the tube with maximum fin thickness in the second and fourth row is comparable to the first row. Vibration grows gradually up to 8 m/s and after that the amplitude

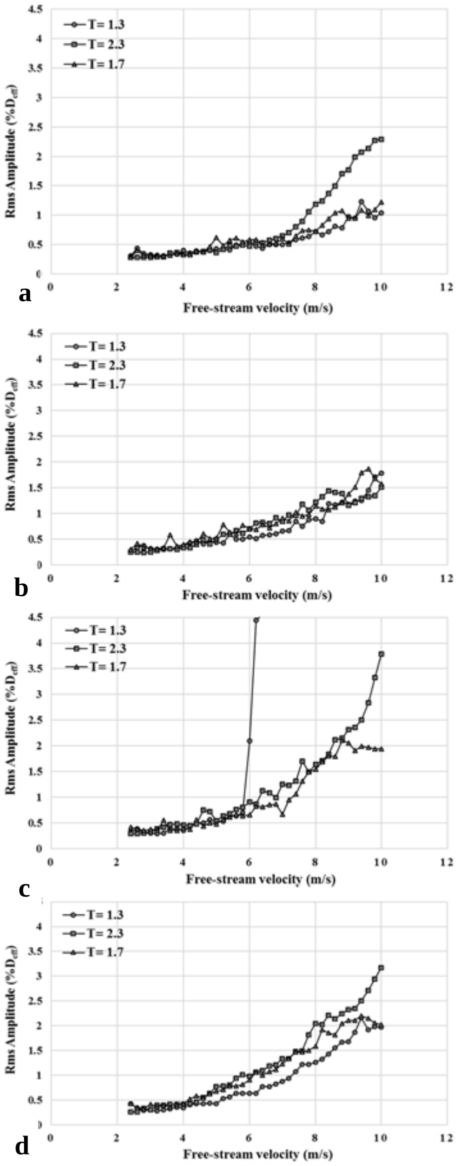


Fig. 7. Amplitude response for FT 1, 5 and 7 in (a) first row, (b) second row, (c) third row, (d) fourth row.

increases rapidly for all the tubes. Overall, the results are comparable for all the tubes in these rows. An inverse behavior is seen in Fig. 7(c), that the fluid elastic instability threshold limit reaches early for the finned tube with minimum fin thickness. Tube response indicates that the array has become unstable by a velocity of about 5.8 m/s and 9.0 m/s for FT 1 and FT 5, 7 respectively. A comparison of vibration response between FT 1 and FT 5 indicates that the amplitude in all the rows is almost twice for FT 5 except in the third row, in which FT 1 becomes unstable early and abruptly in contrast to FT 5. Since the tube experiences extreme vibration in the third row according to literature, which is also evident from this study and tube with maximum fin thickness responded better in this row, showing a gradual increase in the tube amplitude. It may be due to the reason that lowering the fin thickness leads to low structural rigidity and low fin efficiency as well. For FT 7, there is a delay in the occurrence of instability and amplitude grows gradually with the flow velocity in all the rows. The post instability tube's response is highly non linear and strongly depends upon the tube motion trajectory and time lag associated with FEI. Because of this non linear behavior, a comprehensive study is required to understand the post FEI response of the finned tubes in cross flow. In summary, the fluid elastic instability behavior of a single flexible tube in a rigid array is greatly affected by the tube fin thickness since low fin thickness gives higher fin density, leads to more surface area that is good for heat transfer but as a result tube life is compromised as it also enhances the vibrations.

3.3 Effect of Fin Density on FIV

In the third phase, experiments are carried out on tubes with similar fin thickness but different fin density (FT 4, FT5, FT 6). Since tubes in the first and second row experience less vibration as compared to third and fourth row. Therefore, the amplitude response of FT 4, 5 and 6 is almost comparable in the first and second row as shown in Figs. 8(a) and (b). Figures 8(c) and (d) presents the vibration behavior of FT 4,5 and 6 subjected to cross-flow in the third and fourth row of the tube array. In 8(c), the vibration amplitude grows gradually up to 9 m/s and after that, it increases rapidly for FT 5 and 6, but FT 4 becomes unstable early when the free-stream velocity is about 7 m/s. Fluid elastic instability threshold limit reaches early for the tube with low fin density. Therefore, 9 m/s is taken as critical velocity for these tubes.

Tube response indicates that the array has become unstable by a velocity of about 7 m/s and 9 m/s for FT 4 and FT 5 / 6 respectively. The finned tube that has minimum fin density, has the vibration amplitude almost twice than the other two tubes. In 8(d), the vibration pattern in this row is similar to the third row but its amplitude is low. Vibration grows gradually up to 8 m/s and after that, the amplitude increases rapidly for all the tubes. Overall results are comparable for all the tubes in the fourth row, not even twice the vibration for FT 4 which is almost three times in the third row as compared to FT 6. Thus to conclude, the FEI behavior of a single flexible tube in a tube bundle is greatly affected by the fin density since the instability threshold is delayed with the increase in

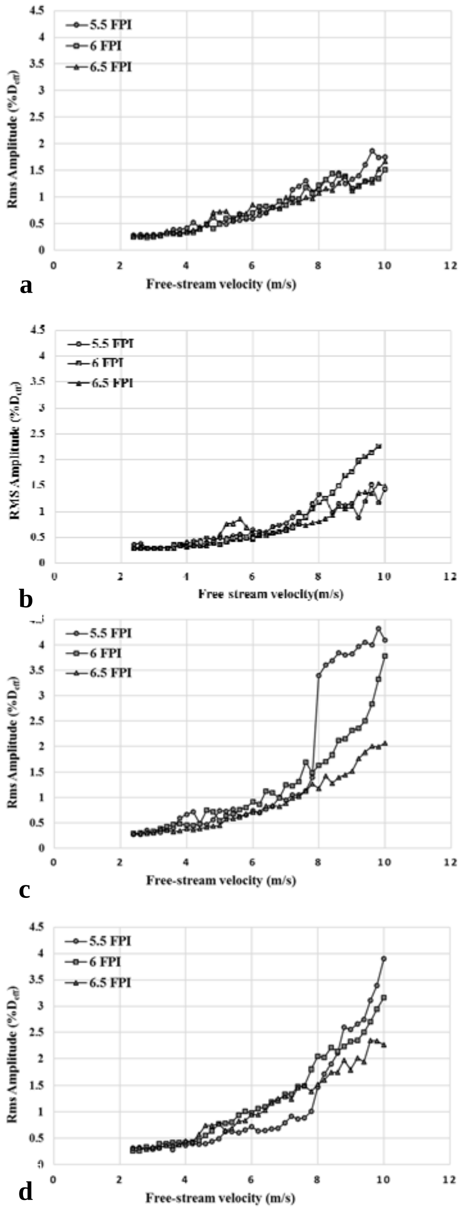


Fig. 8. Amplitude response for FT 4, 5, 6 in (a) first row, (b) second row, (c) third row, (d) fourth row.

fin density for a finned tube array. The reason might be the fin density has a direct impact on tube damping which increases linearly with the increment in fin density. This damping may oppose the pressure forces which results in energy dissipation and amplitude reduction of the monitored tube.

3.4 Spectral Analysis

The frequency spectra of FT 1 vibration in the first and third row of the tube array have been presented in Fig. 9. Figure 9(a) presents the frequency spectra of tube vibration when the tube has been placed in the first row. It is illustrated by the spectra that when the free-stream velocity (FSV) is low, the tube vibrates with small amplitude. It is also evident that frequencies of spectral peaks are proportional to free-stream velocity. The vortex

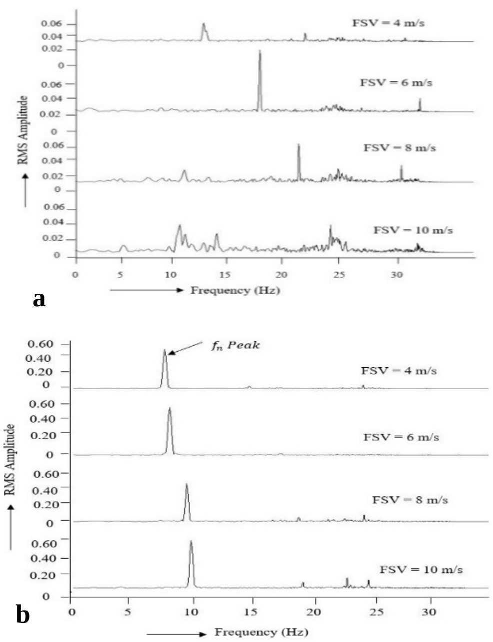


Fig. 9. Frequency spectra for FT 1 for the (a) first row, (b) third row.

shedding frequency increases with the fluid velocity. This behavior might be induced by VIV since it is directly related to vortex shedding frequency and that is proportional to FSV. There are multiple small peaks in the spectra, such as the case when V= 10 m/s. The small multiple peaks occurrence in the Fig. 9(a) is associated with the turbulence phenomenon. As the flow enters the tube array, the turbulence in the flow increases and continues to increase as the flow goes deep into the bundle. Also, the fins on the tube's surface increases the flow turbulence. This turbulence produces small amplitude vibrations and associated band of frequencies apart from the vortex shedding frequency which is proportional to the flow velocity. To adjust all the tubes to the same frequency during experimentation is not easy, that could also be the reason behind the multiples spectral peaks occurrence in the velocity range. Figure 9(b) presents the frequency spectra of FT 1 vibration when the tube is placed in the third row. The amplitude is larger in this row and increases with the velocity. Only one major peak is observed in the corresponding spectra, near to the tube's natural frequency. There seems to be instability characterized by sudden rise in amplitude that might be caused by the lock-in phenomenon which occur when the vibration frequency becomes equal to the tube's natural frequency. The tube location confirms that the vortex shedding is a dominant phenomenon when the tube is placed in the first row, since there is a peak shift in the frequency spectra with the increase in flow velocity.

When the same tube is placed in the third row, neither the vortex shedding vibration nor the vortex shedding frequency that is proportional to the flow velocity occurred. In addition, the peak amplitude is smaller in the first row as compared to the third row. Figure 10 shows the shedding frequency of spectral peaks in first, second, third, and fourth row of the tube array for FT 1,4,5,6, and 7 respectively. In 10(a), the shedding frequency is proportional to flow velocity in first, second, and fourth row. Likewise, the Strouhal number at corresponding shedding frequencies found to be similar, which indicates the vortex shedding presence in the array. However, in the third row, tube vibrates with larger amplitude and becomes unstable, so only one vibration peak is observed near to the tube's natural frequency. To ensure that this behavior is not tube specific, a similar analysis is performed with other finned tubes. The behavior of tubes is quite similar in all cases. In 10(b) and 10(e), similar behavior is observed in all the rows of tube array. In 10(c) and 10(d), the shedding frequency is proportional to flow velocity in all the rows. There is no FEI, so the tubes remain stable in the third row and the vibrations are due to VIV. It is concluded that the critical velocity of fluid elastic vibration is strongly influenced by fin parameters and orientation of the tube array. To calculate the shedding frequency, dimensionless number i.e. Strouhal number (Sr) is used.

$$Sr = \frac{f_s D_{eff}}{u_\infty} \quad (3)$$

Here, f_s represents shedding frequency, u_∞ is the free-stream velocity, and D_{eff} is the effective diameter that can be calculated using Eq. (2).

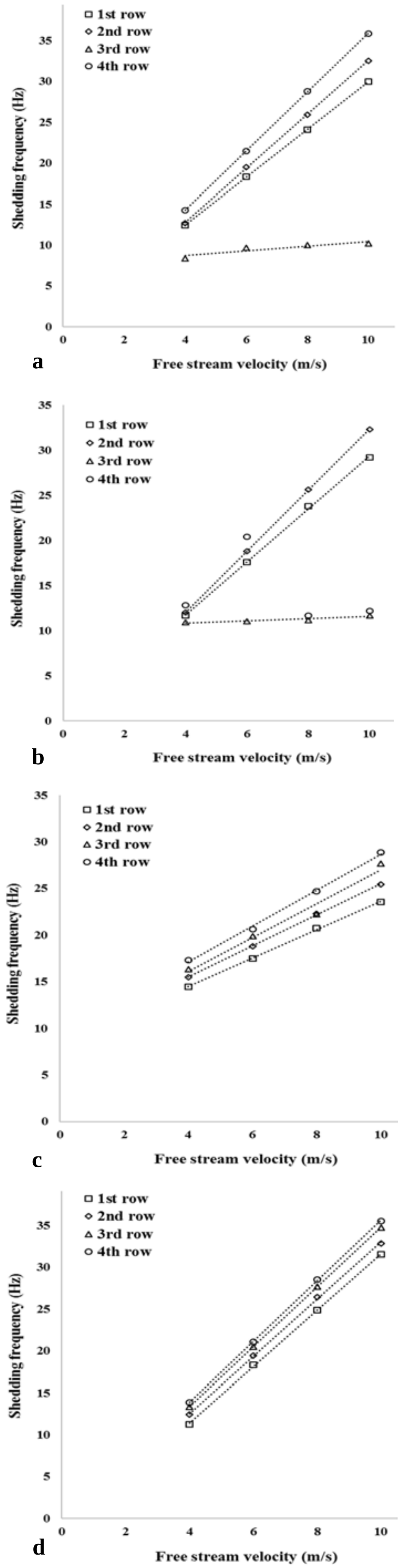
The Strouhal number is very important in frequency analysis of tubes because it gives the idea that inside the tube bundle during the flow, either vortex shedding mechanism is happening or not. The Strouhal number for different finned tubes is shown in Table 3.

Table 3 Strouhal number for finned tubes under study for different tube locations

Finned Tubes	First row	Second row	Third row	Fourth row
FT 2	0.04	0.04	0.04	0.05
FT 4	0.04	0.04	0.02	0.03
FT 5	0.04	0.04	0.04	0.05
FT 6	0.04	0.04	0.04	0.05
FT 7	0.04	0.04	0.02	0.05

In most cases, Strouhal number found to be similar in all the rows which indicate that it is independent of fin geometry but further analysis is required because in this study limited geometries are under observation. Shedding frequency and the Strouhal number is less for finned tubes as compared to the bare tubes. Thus to conclude, in finned tubes vortex shedding is effective as found in our analysis but at the same diameter, it is less when compared to bare tubes.

Turbulence spectral data showed a constant Strouhal number for almost all the arrays under observation for current Reynolds number range. These Strouhal numbers are plotted along-with the data from the literature for comparison and establishing trends. Figure 11 compares the present Strouhal number with the data from the literature. The Strouhal number obtained in the present study is for finned tubes and no Strouhal data is available



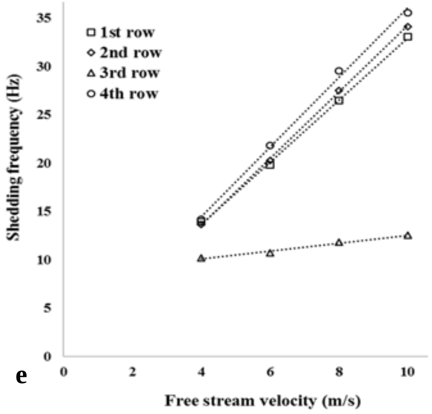


Fig. 10. Shedding frequencies of spectral peaks in the first, second, third and fourth row for (a) FT 1 (b) FT 4 (c) FT 5 (d) FT 6 (e) FT 7.

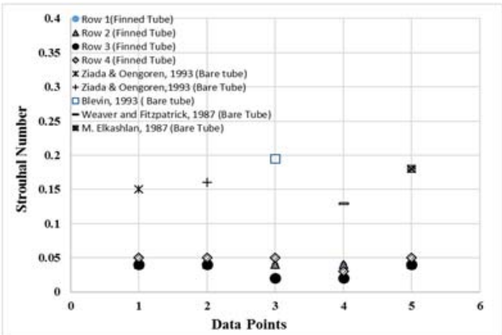


Fig. 11. Comparison of Strouhal number (current study Vs past experiments)

for them in literature. Therefore, the data has been compared with the bare tube's data reported by researchers.

It is evident that there is a clear profile difference between the vortex shedding in finned tubes and bare tube. This difference is expected, since the tube’s vibration behavior in an array is strongly affected by the geometric parameters. Results suggest that vortex shedding might be responsible for the constant Strouhal number. However, further research is required to determine the nature of the phenomenon and to establish conditions under which it is likely to create a tube vibration of vortex shedding in practice.

4. CONCLUSIONS

The Experimental analysis of fluid elastic vibration for parallel triangular finned tube array with P/D_{eff} ratio 1.62, subjected to single-phase cross flow of air is conducted. The effect of different fin parameters and tube location on the instability threshold is assessed for the finned tube arrays. Fin thickness and fin densities used are 1.3, 1.7, 2.3 mm and 4.5, 5, 5.5, 6, 6.5 respectively. Derived conclusions are as follows:

There is no instability observed for most of the finned tubes for the current range of free-stream velocity when those are placed in the first, second and fourth row. The amplitude for the finned tubes increases abruptly in the third row. Thus, tube experiences a maximum vibration in the third row.

Considering the third row, the instability threshold is delayed for the tubes with greater fin thickness. But, for all other rows, tubes with less fin thickness showed the similar behavior. Hence, it is derived that the array of tubes with less fin thickness are unstable in the third row at comparatively higher flow velocities.

With the increase in fin density, instability threshold is delayed and more fins have a stabilizing effect on it, hence the critical velocity is higher for fine finned tube array.

It is noted that the vortex shedding contributed towards the vibration of the finned tubes predominantly in the first, second, and fourth row. However, in the third-row, the tubes vibrated with relatively higher amplitude because of the early instability and no vortex-shedding is observed in this row.

A comparison between the bare tube and the finned tube vibration response indicates that the vibration amplitude for the bare tube is less as compared to the finned tube, so the fluid elastic threshold occurs early for the bare tube. In addition, there is a clear profile difference between the vortex shedding in the finned tube as compared to the bare tubes.

The Strouhal number is independent of fin geometry since it remained constant for various finned tubes when placed in different rows of a tube array.

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