Experimental Study of the Influence of the Surrounding Tube Location and Mass Ratio on the Fluidelastic Instability of Flexibly Mounted Tubes in a Parallel Triangular Tube Bundle

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Experiments have been designed to investigate the fluidelastic instability in a flexibly mounted tube in a parallel triangular tube bundle with a tube diameter of 12.7 mm and pitch-to-diameter (P/D) ratio of 1.45. The monitored tube, placed in the centre of the tube bundle, was equipped with a wireless accelerometer to acquire the vibration response under air cross-flow in a wind tunnel. The vibration response of the monitored tube was captured by changing the surrounding flexible tube location and the mass ratio. This experiment is the first of its kind, as it has not yet been reported in the literature. It is observed that the fluidelastic instability occurs for all the surrounding tube locations and mass ratios and is strongly dependent on the location and mass ratio. A strong tube-to-tube coupling exists, as indicated by the amplitude analysis of the monitored tube. The stability analysis suggests that increasing the mass ratio of the surrounding tubes seems to have an effect on the stability of the monitored tube.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$C_{mf}, C_v$</td>
<td>Damping coefficient of tube in moving fluid and static fluid per unit length (Ns/m²)</td>
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<tr>
<td>$C_s$</td>
<td>Total tube damping coefficient (Ns/m²)</td>
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<td>$F_t$</td>
<td>Fluid force per unit length (N/m)</td>
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<tr>
<td>$K_s, K_T, K_F$</td>
<td>Tubes, structure, and fluid stiffness matrices (N/m²)</td>
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<td>$M_s, M_T, M_F$</td>
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<td>$U_{fs}$</td>
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<td>$\rho$</td>
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1. INTRODUCTION

The design of shell and tube heat exchangers is mainly dependent on the knowledge and prevention of flow induced vibrations in heat exchangers’ tube bundles. The four mechanisms that can cause vibrations in a heat exchanger tube bundle are fluidelastic instability, turbulence, vortex-induced instability, and acoustic resonance.¹ Flow-induced-vibrations in a heat exchanger tube bundle can cause failure in the tube bundle. Each tube in a tube bundle is supported by tube sheets
and baffles in a multi-span tube bundle. The natural frequency is one of the main concerns for designer and engineers, since its matching with any of the excitation frequencies can cause resonance and, ultimately, failure.

Much research has been carried out in the field of flow-induced vibrations for many decades, driven by the importance of the field and its applications in industry.

Fluidelastic instability is considered to be the most important excitation phenomenon as it is the main cause of tube failure in heat exchangers. According to Price, turbulence in the flow induces small amplitude vibrations in the tube. These vibrations alter the flow pattern and fluid forces on the tube. This alteration induces more vibration in the tubes. If the vibration continues to increase, then a stage comes when the tube vibration grows exponentially; the tube is then said to be fluidelastically unstable. Chen presented a review of different instability models and compared them with published experimental data. Roberts concluded that a tube displacement mechanism can account for the dynamic instability of the tube row in cross-flow. He found that the motion of the alternating tubes produces changes in the wake pairing downstream of the tube row, which results in the net work done by the fluid on the moving tubes in every cycle. So, it was established that the lack of space between tubes in the tube bundle results in “jet switch phenomena” in tightly packed tube bundles.

Numerical simulation has been performed to analyse the fluidelastic instability of multiple cylinders subjected to cross-flow. The tubes are arranged in a rotated (parallel) triangular bundle. The results show that vibrations with high amplitude occur in multiple cylinders as the free stream velocity goes beyond a critical value. Also, beyond the critical value, the cylinders start vibrating in elliptical orbits, indicating dynamic instability in tubes. Fluidelastic instability results in large-amplitude vibration or even failure of the tubes in the tube bundle. In tube bundles, the interactions of the tubes with the fluid and the coupling among the tubes through the fluid are very complex. However, it is likely that the unsteady fluid dynamic forces on a tube are mainly induced by vibrations of the tube itself and its neighbouring tubes. Under this assumption, the vibration behaviour of a tube in the tube bundle should be most affected by its surrounding tubes. For instance, a tube surrounded by fewer tubes in the tube bundle may have a different vibrational response from one surrounded by more tubes under conditions that are otherwise the same. A comprehensive review of the different excitation mechanisms in the tube bundle has been presented by Khushnood et al.

In practical operation of shell and tube heat exchangers, all the vibrating tubes may have different masses. When a tube in a tube bundle becomes damaged and starts to leak, it is usually clogged from the inlet side in order to avoid mixing between shell and tube side fluid. Moreover, there is a possibility that tubes will not be equally filled at all times. This means that in a complex tube-to-tube interaction within the tube bundle.

The objective of this paper is to study the flow-induced vibrations of a flexibly mounted tube in a parallel triangular array subjected to cross-flow with surrounding flexible tubes having different mass ratios. The emphasis is on studying the effects of the location and mass ratio of the surrounding flexible tubes on the vibration behaviour of the monitored tube. The summary of studies carried out previously on the vibrational behaviour of tube bundles (Table 1) presents, at a glance, a comparative view of different experimental parameters selected by different researchers for their experimentation.

### 2. MATHEMATICAL MODELLING

The study has been carried out on a flexibly mounted tube in an array subjected to the coupling of fluid. The effects of fluid elasticity have been analysed and expressed frequently in terms of fluidelastic parameters. The geometric parameters that characterize a general tube array are the tube outer and inner diameters $d_o$ and $d_i$ respectively, pitch $q$, and length of the tube $L$. The total tube mass $m_s$, total tube damping $C_s$, and total tube stiffness $k_s$ are the mechanical properties of the flexible tube. Because of the relatively small length of the tube and the fact that it is suspended by means of piano wire, the first mode of vibration is dominant and the equation of the tube without the fluid is presented in Eq. (1) and Fig. 1

$$m_s \ddot{u}_s + C_s \dot{u}_s + k_s u_s = 0.$$  \hfill (1)

Alternatively, it can be expressed in a more elaborated form as in Eq. (2)

$$\ddot{u}_s + 2\xi \omega_n \dot{u}_s + \omega_n^2 u_s = 0;$$  \hfill (2)

where $\xi$ and $\omega_n$ are the damping ratio and the natural frequency of the flexible tube,

$$k_s = m_s \omega_n^2;$$  \hfill (3)

and

$$C_s = 2m_s \Omega_n \xi.$$  \hfill (4)

The flow regimes are defined in terms of reduced velocity ($U_r$) is presented in Eq. (5) as:

$$U_r = \frac{2\pi U_p}{\omega_n a_o}.$$  \hfill (5)

The pitch or gap velocity ($U_p$) can be expressed in the form of Eq. (6) as:

$$U_p = \frac{U_f s q}{q - d_o}.$$  \hfill (6)

![Figure 1. Elementary cell $\Omega_F$ in tube bundle configuration.](Image)
Now, consider that the flexible tube has been placed in the fluid at rest. As there are additional parameters including fluid viscosity ($C_v$) and added mass ($m_a$), the modified equation of motion becomes:

$$(m_s + m_a)\ddot{u}_s + (C_s + C_v)\dot{u}_s + k_s u_s = 0; \quad (7)$$

or

$$\ddot{u}_s + 2\xi_s \omega_n^0 \dot{u}_s + \omega_n^2 u_s = 0; \quad (8)$$

where $\omega_n^0$ and $\xi_s$ designate the pulsation and damping coefficients of the tube in fluid at rest. Thus, the following identification can be used:

$$k_s = m_s \omega_n^2 = (m_s + m_a + m_i) \omega_n^2. \quad (9)$$

This identification yields:

$$m_a = m_s \left(\frac{\omega_n^2}{\omega_n^2} - 1\right) - m_i; \quad (10)$$

and

$$C_v = 2(m_s + m_a + m_s) \omega_n^2 \xi - C_s. \quad (11)$$

Ignoring the internal tube mass (i.e. $m_i = 0$) for no internal tube flow condition, Eqs. (10) and (11) reduce to Eqs. (12) and (13):

$$m_a = m_s \left(\frac{\omega_n^2}{\omega_n^2} - 1\right); \quad (12)$$

$$C_v = 2(m_s + m_a) \omega_n^2 \xi - C_s. \quad (13)$$

For the moving fluid, the equation of motion after introducing damping and added mass terms becomes:

$$(m_s + m_a)\ddot{u}_s + (C_s + C_{mf} + C_v)\dot{u}_s + (k_s + k_{mf}) u_s = F_t. \quad (14)$$

Equation (14) can be rewritten as:

$$\ddot{u}_s + 2\xi_{mf} \omega_n^0 \dot{u}_s + \omega_n^{mf2} u_s = F_t; \quad (15)$$

where $\xi_{mf}$ and $\omega_n^{mf}$ are the damping ratio and natural frequency of the flexible tube in the moving fluid. The stiffness and the damping terms are written as:

$$k_{mf} = (m_s + m_a) \omega_n^2 - m_s \omega_n^2; \quad (16)$$

$$C_{mf} + C_v = 2\omega_n^2 \xi_{mf} (m_s + m_a) - 2m_s \omega_n \xi. \quad (17)$$

Finally, the stiffness and damping terms due to the moving fluid are given by Eqs. (18) and (19):

$$(k_{mf} = m_s \omega_n^2 \left(\frac{\omega_n^{mf2}}{\omega_n^2} - 1\right); \quad (18)$$

$$C_{mf} + C_v = 2m_s \left(\omega_n^2 \xi_{mf} \frac{\omega_n^2}{\omega_n^2} - \omega_n \xi\right). \quad (19)$$

In practice, dynamic instability occurs when the critical flow velocity is reached, which is the critical mass ratio. The fluidelastic parameters $C_{mf}$, $C_v$, $k_{mf}$, and $m_a$ can be determined experimentally by measuring the frequency and damping ratio of the flexible tube in stationary as well as moving fluid.

Equations (20), (21), and (22) present a mathematical for-
The monitored tube is indicated by a black filled circle and the vibration response. The whole assembly was supported by four mounted on top of the monitored tube in order to capture its vibration in static as well as moving fluid. This model captures the vibration behaviour of the flexible tube in static as well as moving fluid.

![Image of tube structure](image-url)

### 3. EXPERIMENTAL SETUP

Experiments were conducted in a GUNT low-speed open-loop wind tunnel (model HM 170). The test section is made of acrylic plates and has internal dimensions of $300 \text{ mm} \times 300 \text{ mm}$. The model studied is mounted in the test section, and the flow medium (air) is set in motion so as to achieve the desired flow in the tunnel. The carefully designed nozzle contour and a flow straightener in the wind tunnel ensure a uniform velocity distribution with a turbulence level less than 1% of the test section. The built-in axial fan with downstream guide vanes and a variable-speed drive is characterized by energy-efficient operation at high efficiency. Air velocities of up to 28 m/s can be achieved in this open wind tunnel without the test model being placed within. Within the test section is placed 150 mm upstream of the tube bundle to measure the flow velocity and is connected to a digital differential pressure transducer to display the flow velocity. The measurement of the velocity has an uncertainty of about $\pm 0.025 \text{ m/s}$. By using the HM 170 system for data acquisition, the measured values of velocity can be transferred to a computer and saved in an Excel file. A line diagram of the wind tunnel is presented in Fig. 3.

A total of 77 aluminium tubes, with the central one instrumented with a tri-axial accelerometer, were used to form a parallel triangular tube bundle (Fig. 3). The experiments were performed in four phases, each with a different mass ratio of surrounding tubes. First, surrounding tubes with lowest mass ratio was used. Then, the mass ratio was increased by adding a suitable dead mass inside them to achieve the required mass ratio of the surrounding tube. In each phase, the surrounding tubes—numbered 1 through 6—were placed at different locations in the tube bundle, and the vibration responses were recorded at different velocities. A total of four mass ratios and six locations of surrounding tubes were used. The complete specifications of the tube bundle is presented in Table 2.

The monitored tube was suspended using a piano wire (0.2 mm thick) and a tensioning mechanism provided to adjust the tension of the wire (Fig. 4). The natural frequency of the monitored tube and the surrounding flexible tube was tuned to 15 Hz with an uncertainty of 0.5%. The accelerometer was mounted on top of the monitored tube in order to capture its vibration response. The whole assembly was supported by four acrylic plates: two used to hold the rigid tubes and two to hold the tensioning mechanism (Fig. 4).

Figure 5 presents a cross-sectional view of the tube bundle. The monitored tube is indicated by a black filled circle and the

### Table 2. Specifications of the tube bundle.

| Tube material | aluminium |
| Type of tube array | Parallel triangular |
| Mass per unit length of the tube | 0.255 kg/m |
| Mass ratio of surrounding tubes with respect to monitored tube | 0.25, 0.5, 0.75, 1.0 |
| Outer / inner diameter of tubes | 12.7 mm / 11.5 mm |
| Total number of tubes | 77 |
| Tube length | 292 mm |
| P/D ratio | 1.45 |
| Modulus of elasticity of tube | 690000 MPa |
| Density of tube | 2800 kg/m$^3$ |
| Density of air | 1.225 kg/m$^3$ |
surrounding flexible tubes by numbers marked on them.

Figure 6 presents the accelerometer signal of the free vibration of the tube in still air and the subsequent Fast Fourier transform of the signal. The peak in Fig. 6(b) denotes the natural frequency of the tube.

The data acquisition system for acquiring the vibration response of the monitored tube consists of a G-link wireless triaxial accelerometer sensing node along with the WSDA Wireless USB Base Station connected with a PC through Node Commander Software developed by Microstrain Corporation. The data were acquired at a sampling frequency of 679 Hz. The signal collected from each sensor was averaged to achieve good repeatability of the results, and the data obtained were saved in an Excel file for further analysis. The time domain signals of the vibration response were analysed using Sigview software developed by Signal Lab. The acceleration signals captured from the wireless accelerometer were calibrated, and the RMS acceleration amplitude associated with the tube natural frequency was divided by the frequency squared to obtain the RMS vibration amplitudes.\(^{19}\)

4. EXPERIMENTAL OBSERVATIONS AND DISCUSSION

Experiments were conducted on a flexible monitored tube surrounded by other flexible tube to determine the fluidelastic instability of the monitored tube. The effect of the flexible surrounding tubes on the monitored tube was investigated with different mass ratios. Four sets of experiments were conducted with four different mass ratios and six different tube locations.

4.1. Effects of Surrounding Flexible Tube Location

Figure 7 presents the vibration response of the monitored tube for all four mass ratios at different locations of the surrounding flexible tubes. The amplitude analysis shows an increasing trend of vibration amplitude with increases in velocity for all four mass ratios. The mass ratios of 0.75 and 0.25 lead to an exceptional rise in the vibration amplitude of the monitored tube when the reduced velocity reaches about 14, whereas the mass ratios of 0.5 and 1.0 lead to a reduced velocity of about 20.
This exceptional rise in amplitude is associated with the phenomenon of fluidelastic instability. The experimental results show that the critical reduced velocity associated with fluidelastic instability of the monitored tube shows a strong dependency on the mass ratio of the surrounding tubes. Under instability, an amplitude of about 10% of the tube diameter was observed in the case of tube 1 with a mass ratio of 0.25 for the monitored tube.

In the case of tube 2, at mass ratios of 0.5, 0.75, and 1.0, a trend of increasing vibration amplitude is observed with the increase in velocity but with different vibration amplitudes. The maximum amplitude of almost 6.5% of the tube diameter was observed at the mass ratio of 0.25 and the minimum monitored tube vibration of almost 1.5% of the tube diameter was observed at a mass ratio of 0.5.

For tube location 3, at mass ratios of 0.5 and 1.0, a rapid increase in the vibration amplitude of the monitored tube was observed at a reduced velocity of about 20. Conversely, at the mass ratio of 0.25, there was no significant effect on the amplitude response of the monitored tube (Fig. 3(c)). At a mass ratio of 0.75, the instability occurs early at a reduced velocity of about 13, characterized by a rapid increase in vibration am-
Figure 7. Amplitude responses of the monitored tube for all four mass ratios: (a) Tube 1 made flexible; (b) Tube 2 made flexible; (c) Tube 3 made flexible; (d) Tube 4 made flexible; (e) Tube 5 made flexible; and (f) Tube 6 made flexible.
amplitude. Similarly, with regard to the location of tube 4 at the mass ratios of 0.5 and 1.0, a sudden rise in amplitude level was observed at a reduced velocity of about 12. At the mass ratios of 0.25 and 0.75 no significant rise in amplitude was observed, and the amplitude level was relatively low: about 4% of the tube diameter.

It was also observed that the location of tube 5 shows unique behaviour at all four mass ratios compared to other tube locations. The vibration amplitude increases very slowly with the increase in velocity, with a maximum amplitude of about 2% of the tube diameter except at the mass ratio of 0.5. Thus, the location of tube 5 seems to have a stabilizing effect on the monitored tube (Fig. 7(e)). In the case of the location of tube 6 at mass ratios of 0.25 and 0.5, a gradual increase in the vibration amplitude of the monitored tube was observed with increasing velocity and a maximum amplitude of 9% of the tube diameter was observed at the mass ratio of 0.5. Table 3 presents the critical reduced velocities for the monitored tube under different tube locations and mass ratios.

In summary, the vibration response of the monitored tube depends not only on the tube location but also on the mass ratio of the surrounding tubes. The tube pairs 2 & 6 and 3 & 5, while in different rows, occupy symmetric locations with respect to the monitored tube. While it was expected that tubes 2 and 6 should influence the vibration response of the monitored tube in a similar manner, same as tube 3 and tube 5, this is not the case. This may be due to the irregular coalescence jets that form downstream of the first row of tubes. This irregularity in the jet coalescence phenomenon increases as the flow goes deep into the bundle. When the flow reaches the flexible tubes in the tube bundle (as in this case), the jet coalescence pattern is altered by the vibration of the tubes and fluid coupling between tubes. This makes the phenomenon highly irregular and complex, leading to different behaviours for every tube location, as seen in the experiment results. Also, the vibration pattern of the monitored tube is different for each surrounding tube location, and mass ratio and tends to be elliptical after instability, which makes the interaction between the monitored and surrounding tubes more complex.

4.2. Effect of Mass Ratio

This section presents the comparative analysis of the vibration amplitude responses of the monitored tube performed under all four mass ratios with different tube locations. At a mass ratio of 1.0 of the surrounding tubes, the highest vibration amplitude of the monitored tube was observed to be about 9% of the tube diameter for surrounding tube location 4, whereas for the tube location 5, the minimum vibration amplitude was observed and was less than 1% of the tube diameter even with the higher values of the reduced velocity. Also, for tube location 6, the monitored tube vibration amplitude jumped significantly at the reduced velocity of about 8, showing early signs of instability (Fig. 8(a)).

At the mass ratios of 0.25 and 1.0, the monitored tube shows the similar behaviour but has distinct vibration amplitudes for different tube locations. The maximum vibration amplitude response—almost 10% of the tube diameter—in the monitored tube was observed for tube location 1 at the mass ratio of 0.25. The comparison of all tube locations at this mass ratio suggests that, for tube location 1, the monitored tube shows early instability at the highest amplitude levels as compared to the other tube locations.

Figures 8(b) and 8(c) present the amplitude analysis of the monitored tube for all tube locations at the mass ratios of 0.5 and 0.75, respectively. At the mass ratio of 0.75, the monitored tube shows early signs of instability for tube location 3 with the amplitude levels approaching about 8% of the tube diameter. At the mass ratio of 0.5, it was observed that for tube locations of 1, 3, and 5, there is an early rise in the amplitude of the monitored tube, indicating instability at a reduced velocity of about 20. Therefore, it can be concluded that the vibration amplitude and the occurrence of instability in the monitored tube for a specific mass ratio has a strong dependency on the surrounding tube location, which is significant at higher values of reduced velocity. However, a detailed analysis is required to analyse whether a single surrounding tube has the same impact as multiple flexible surrounding tubes on the vibration of the monitored tube.

4.3. Amplitude Changes With the Surrounding Flexible Tube Held Rigid

Figure 9 shows a plot of the experimental results of the changes in vibration amplitude of the monitored tube when the vibrating surrounding flexible tube is kept rigid. The surrounding flexible tube was placed in a certain location, as indicated by tubes 1 to 6 (Fig. 5), and allowed to vibrate. The response of the monitored tube was captured. Then, the surrounding tube was held rigid and the response of the monitored tube was again captured. The vibration amplitude of the monitored tube when the surrounding tube vibrates was subtracted from the amplitude of the monitored tube when the surrounding tube was held rigid. This amplitude difference has been plotted as the percentage reduction in amplitude to show the extent of coupling between the monitored tube and the surrounding tubes (Fig. 9).

The analysis suggests that strong coupling exists between the monitored tube and the surrounding tubes, dependent upon the location and mass ratio of the surrounding flexible tube. At all four mass ratios, the effect of the surrounding flexible tube locations on the vibration amplitude of the monitored tube is not significant at low values of reduced velocity. However, at higher velocities, this effect becomes significant. In fact, for the mass ratio of 0.5 at the reduced velocity of about 27, the peak appears on the negative side of the x-axis (Fig. 9(b)). This peak shows that when the surrounding flexible tube is held rigid, the vibration amplitude of the monitored tube increases by about 800%, which otherwise (when the surrounding tube is vibrating) would be much smaller. This indicates that vibration of the surrounding flexible tube has a profound stabilizing ef-
fect on the monitored tube at higher reduced velocities. Large peaks can also be seen for the mass ratios of 0.25 and 0.75. The mass ratio of 1.0 shows smaller peaks compared to other mass ratios even at higher velocities (Fig. 9(d)).

The current analysis suggests that the location of the surrounding flexible tube is a considerable factor for vibration amplitude analysis of the monitored tube but in some cases, such as at the mass ratio of 0.5, tube location 5 seems to be critical for reduction of the vibration amplitude of the monitored tube even at low values of reduced velocity.

As discussed in Section 4.1., due to the irregular gap flows and fluid coupling between vibrating tubes, complex physics exists inside the tube bundle and a detailed visualization study is required to fully understand the fluid-to-tube and tube-to-tube coupling of different tubes at different locations inside the tube bundle.

4.4. Stability Analysis

A stability map is one of the most important tools to understand instability in a single cylinder as well as a group of cylinders (i.e., tube bundles). The map is mathematically formulated in terms of two non-dimensional parameters, that is, the mass damping parameter (MDP) and the reduced velocity, and given by Eq. (28):

$$\frac{U_c}{f_n d_o} = K \left( \frac{m o}{\rho d_o^2} \right)^b.$$  \hspace{1em} (28)

Roberts is considered to be the first to have studied fluidelastic instability in cross-flow.\(^\text{5}\) He performed experiments on a row of cylinders in a wind tunnel and calculated the values for the constants based on his experimental data, that is, \(K = 9.8\) and \(b = 0.5\). Connors carefully studied the fluidelastic instability in the tube bundle and considered that the value of \(K\) should be 9.9, which is also in agreement with Roberts’ findings.\(^\text{34}\)
Later, researchers focused on understanding the fluidelastic instability in the tube bundle in detail under different geometries and P/D ratios. The main goal is to develop concrete guidelines for designing the tube bundle in order to avoid failure of the heat exchanger. Different models have been proposed to predict the stability boundary based on the geometric parameters of the heat exchanger. Pettigrew et al. recommended that $K = 3.3$ can be used as a design guideline for all tube bundles, while others are of the opinion that the value of $K$ is dependent on the geometry of the tube bundle.\(^{35}\) A standard value of $K = 3.0$ is generally suggested for the design of all types of tube bundles.\(^{36}\)

For the current study, a comparison of the experimental data has been presented with the theoretical stability boundaries proposed by Price,\(^2\) Weaver and Fitzpatrick,\(^1\) and Pettigrew and Taylor\(^{37}\) (Fig. 10). The comparison suggests that the stability boundaries are not universal and are unable to reflect the complete dynamics of the tube bundle. At the mass ratio of 0.25, most of the data points lie in the unstable region, indicating that the proposed models underestimate the stability boundary (Fig. 10(a)). This suggests that at lower mass ratios (in our case, 0.25), the surrounding tubes tend to destabilize the monitored tube and contribute towards early triggering of instability of the monitored tube. But as the mass ratio increases
Figure 10. Stability maps of monitored tube for all six locations of the flexible tube: (a) mass ratio of 0.25; (b) mass ratio of 0.5; (c) mass ratio of 0.75; and (d) mass ratio of 1.0.
from 0.25 to 1.0, the experimental data tend to follow the proposed stability boundaries (Fig. 10(b), (c), and (d)). So, it is recommended that the mass ratio of the surrounding tubes be incorporated when developing the design guidelines for tube bundles.

5. CONCLUSIONS

From the current study, the following conclusions can be drawn.

- For the reduced velocity range of \(0 \leq U_r \leq 10\) for all six surrounding tube locations and all four mass ratios, the vibration amplitude levels of the monitored tube are generally less than 2% of the tube diameter, indicating that the monitored tube is stable in this range and independent of the surrounding tube location and mass ratio. However, in the reduced velocity range of \(U_r > 10\), the vibration amplitude of the monitored tube shows dependency on the location of the surrounding flexible tube and mass ratio.

- The surrounding flexible tube tends to limit the amplitude of the monitored tube at higher values of reduced velocity. Tube location 5 is considered to be critical for the stability of the monitored tube.

- The stability analysis suggests that as the mass ratio of the surrounding tube increases, the data points tend to follow the stability boundaries. Thus, it is recommended that the variation in mass ratio of the surrounding tubes be incorporated in the development of design guidelines for tube bundles.

This study was limited to an analysis of the vibration of only the central/monitored tube. Therefore, it could be extended by acquiring the responses of both the surrounding tubes and the central one and analysing the correlation between two vibrating flexible tubes. Also, the other surrounding tube locations could be studied to understand the full picture of the actual heat exchanger.

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