Modeling of Flow-Induced Vibration Response of Heat Exchanger Tube with Fixed Supports in Cross Flow

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Abstract The shell and heat exchanger has either straight or U-tube with baffle support at an intermediate point and fixed at their ends. Cross flow-induced vibration has been found to be most destructive in many industrial applications. This work reveals modeling of flow-induced vibration response: vortex shedding, turbulent buffeting and fluid elastic instability of single straight tube with fixed support, which may cause the amplitude of vibration in terms of fretting wear at the tube support. Modal analysis has been performed using ANSYS to find out natural frequencies and mode shapes, and MATLAB program is developed to check flow-induced vibration response. It is found that results are in good agreement with available literature data. This approach can be used to provide conservative designs or troubleshoot an existing heat exchanger.

Keywords Flow-induced vibration · Vortex shedding, turbulent buffeting · Fluid elastic instability

1 Introduction

Flow-induced vibration has been a serious concern in the design and operation of shell and tube heat exchangers, such as steam generators, condensers and coolers. In shell and tube-type heat exchangers, one fluid flows inside the tubes, while another fluid flows through the shell and across the outside of the tubes. Considering performance and economics, larger-scale heat exchangers are being designed with higher mass flow rates and larger thermal gradients. It is thus desirable to use small diameter tubes and large distances between the tubes support to increase the heat transfer area and reduce shell-side pressure drops. These systems have, therefore, become more

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flexible and prone to flow-induced vibration damage in the form of fretting wear and fatigue failure at the tube supports as well as impacting at the tube middle span [1].

In heat exchangers, there are four types of flow configuration generally classified as: cross flow, internal axial flow, external axial flow and annular or leakage flow. Among these flow types, cross flow-induced vibration has been found to be the most destructive in many industrial applications [2].

The fluid flow causes excitation to the tubes. The performance of the tubes varies according to flow velocities and can be classified as

- Vortex shedding or flow periodicity.
- Turbulent buffeting.
- Acoustic resonance.
- Fluid elastic instability.

Vortex shedding and turbulent buffeting in general cause small amplitude vibrations which in turn may cause long-term fretting wear at the tube supports. And fluid elastic instability may cause tube failure due to its large vibration amplitude characteristics.

To render the problem amenable for most analytical studies and experimental investigations, the flow conditions are idealized as:

- The flow is uniform and steady.
- The incident of the flow is either axial or normal to the tubes.
- The tube motion is linearized, and it is assumed that the frequencies are well defined [3].

The baffle supports provide a simply supported condition.

The specific objective of the work is to check flow-induced vibration response of heat exchanger tube: vortex shedding, turbulent buffeting and fluid elastic instability using computational tools like ANSYS and MATLAB and compares results with available literature.

2 Vibration Evaluation Procedure

The vibration evaluation procedure involves an estimation of certain parameters for various flow-induced excitation mechanisms. The estimated parameters are compared with their respective limiting values to check whether or not vibrations from such excitation mechanisms can cause potential damage to the tubes and the shell. Need to individually examine various zones of interest, namely the nozzle inlet zone, U-bend region and baffle window region, since there is a likelihood of high turbulence and high cross flow velocity in these regions and the variations of span lengths between baffle supports compared to the central baffle region [2].

3 Steps of Vibration Evaluation

1. Calculate the effective mass per unit length.
2. Identify the zones of interest (inlet, baffle window, central baffle zones, U-bend, etc.) to calculate the natural frequency.
3. Calculate the natural frequency (f_n) for spans in various regions of interest. Perform modal analysis.
4. Calculate the damping parameter.
5. Calculate cross flow velocity U.

Formulation of flow excitation mechanisms.
Step 6 for liquid flow.

Vortex Shedding

a. Calculate vortex shedding frequency. Check for the acceptance criteria.

Turbulent Buffeting

Calculate tube response due to random excitation and check for the acceptance criterion.

Fluid Elastic Instability

Calculate the critical velocity and compare with cross flow velocity.
Keep the maximum cross flow velocity below the critical velocity.

Acoustic resonance

a. Calculate acoustic resonance frequency.
b. Calculate vortex shedding frequency. Check for various vortex shedding criteria.
c. Calculate turbulent buffeting frequency. Check for various turbulent buffeting criteria.

4 Problem Formulation and Methodology

4.1 Problem Formulation

Many researchers have developed a semiempirical model and mathematical model to predict interaction damping. Computer codes, such as VIBIC, PIPO-FE, ADINA, have been developed to calculate tube wear work-rates to aid in the prediction of tube fretting wear damage. Modal analysis has been performed to find out natural frequencies and mode shapes. MATLAB program has been developed to check flow-induced vibration response: vortex shedding, turbulent buffeting, fluid elastic instability of heat exchanger tube (fixed-fixed straight tube) which may cause long-term fretting wear at the tube supports.

4.2 Methodology

The flow-induced vibration procedure involves an estimation of certain parameters for various flow-induced excitation responses. The estimated parameters are compared with their respective limiting values to check whether or not vibrations from such excitation mechanism can cause potential damage to the tubes and the shell. Formulation of flow excitation response for liquid flow has been performed in this work. The following methodology is being performed to carry out flow-induced vibration response of shell and tube heat exchanger tube in cross flow.

Step 1 Calculate the effective mass per unit length.
Step 2 Identify the zone of interest.
Step 3 Calculate the damping parameter.
Step 4 Calculate cross flow velocity for the (TEMA) shell under consideration.
Step 5 Develop flowchart for flow-induced vibration response: vortex shedding, turbulent buffeting, fluid elastic instability of shell and heat exchanger single straight tube in cross flow.
Step 6 Perform modal analysis by using ANSYS and find out the natural frequencies and respective mode shapes.
Step 7 Formulate the response due to flow excitation in liquid flow.
Step 8 Check the response of tube for the following responses by utilizing MATLAB tool:

(a) Develop program in MATLAB for the vortex shedding response.
(b) Develop program in MATLAB for the tubular buffeting response.
(c) Develop program in MATLAB for the fluid elastic instability response.

Step 9 Compare results with the available literature data.

4.3 Steps for the Newton–Raphson Method

1. Choose initial values of velocity magnitude $|v|_0$.
2. Use the estimated $|v|_0$ and calculate mismatch δp.
3. Take iteration $n - 1$ for velocity $|v|_0$ and calculate mismatch ΔQ.
4. Use the estimated $|v|_0$ to formulate Jacobian matrix J(0).
5. Check if all the mismatches are below small no. Terminate process. If yes, otherwise go back to step 1 to start the next iteration with updates given below equation.

6. Calculate velocity.

$$|v_1| = |v_0| + \left(\frac{1 + |v_0|}{|v_0|}\right)$$

5 Modal Analysis

5.1 Modal Analysis

A modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component. The natural frequencies and mode shapes are an important parameter in the design of a structure for dynamic loading conditions. A modal analysis can be performed using ANSYS in which damping is ignored in modal analysis to determine the natural frequency. Any applied loads are ignored since natural frequency information is also helpful for avoiding resonance reduction noise and as an important meshing check.

5.2 Natural Frequency Analysis

The frequency with which any object will vibrate if disturbed and allowed to vibrate on its own without any external force is called as natural frequency. Inherent characteristics property we recognized each other by face, non-living objects are identified by properties like weight, natural frequency, etc. No external force is applied during the analysis as natural frequency being inherent characteristics property of any component/assembly. Word free in the name "free vibration" indicates free from external force. Constraints: Except free-free run, natural frequency analysis is carried out as per actual constraints. Damping is neglected for natural frequency calculations. Analysis demonstrates magnitude of frequency and corresponding mode shape Mode are normalized. And result plot helps us to understand magnitudes of frequency and corresponding mode shape (usually, a very high value) are not meaningful and should be neglected.

6 Mathematical Calculation of Problem

Case no-01: Let's consider the segmental baffle heat exchanger design, few parameter of heat exchanger are given below for study flow-induced vibration [4].

$$
\begin{aligned}
\text{Tube length} &= 3.58\,\text{m} \\
\text{Shell inside diameter} &= 591\,\text{mm} \\
\text{Baffle thickness} &= 9.5\,\text{mm} \\
\text{Baffle cut} &= 147\,\text{mm} \\
\text{Baffle hole diameter} &= 0.768\,\text{in.}/19.5\,\text{mm} \\
\text{Baffle spacing} &= 447\,\text{mm} \\
\text{Tube outside diameter} &= 19.1\,\text{mm} \\
\text{Tube inside diameter} &= 16.56\,\text{mm} \\
\text{Tube material density} &= 8.5\,\text{g/cm}^3 \\
\text{Tube modulus of elasticity} &= 110 \times 10^9 \\
\text{Tube pattern} &= 30^\circ \\
\text{Pitch diameter ratio} &= 1.25 \\
\text{Snell side fluid} &= \text{water} \\
\text{Fluid density} &= 1\,\text{gm/cm}^3 \\
\text{Fluid Kinematic, viscosity} &= 10^{-5}\,\text{m}^2/\text{s}.
\end{aligned}
$$

Calculation procedure for shell-side liquids to check flow-induced vibration response

Case no: 01
Step 1:
Calculating tube mass per unit length (m)

(a) Calculating hydrodynamic mass coefficient C_m by TEMA method
$C_\text{m} = 1.756$ (triangular pitch ratio diameter = 1.25)
(b) Calculate the tube mass per unit length

$$
\begin{aligned}
m &= \text{structural mass} + \text{added mass} + \text{contained fluid mass} \\
&= \frac{D^2 \pi C m \rho s}{4} + \frac{Di^2 \pi \rho i}{4} + \frac{\left(D^2 - Di^2\right)\pi m \rho s}{4}
\end{aligned}
$$

$$
\begin{aligned}
\text{Structural mass} &= \frac{D^2 \pi C m \rho s}{4} \\
&= \pi \times \left(1.9 \times 10^{-3}\right)^2 \times 1.756 \times 1000/4 \\
&= 0.503\,\text{m}
\end{aligned}
$$

$$
\begin{aligned}
\text{Added mass} &= \frac{Di^2 \pi \rho i}{4} \\
&= \pi \times \left(16.56 \times 10^{-3}\right)^2 \times 1.1455/4
\end{aligned}
$$

$$= 2.46 \times 10^{-4}\,\mathrm{m}$$

$$\begin{aligned}\text{Contained fluid mass} &= \frac{\left(D^2 - Di^2\right)\pi m \rho s}{4}\\ &= \pi \times \left(0.0191^2 - 0.0156^2\right)/4\\ &= 0.579\,\mathrm{m}\end{aligned}$$

$$\begin{aligned}\text{Mass per unit length} &= 0.503 + 2.64 \times 10^{-4} + 0.579\\ &= 1.08\,\mathrm{m}.\end{aligned}$$

Step 2:
The natural frequency for the straight tube

(a) Natural frequency of tube = 6.2706 Hz (From modal analysis)

Step 3:
Calculate damping parameter

$$\begin{aligned}\delta_1 &= \frac{3.41D}{mf_n}\\ &= 3.41 \times 0.0191/1.08 \times 6.2706\\ &= 9.617 \times 10^{-3}\\ \delta_2 &= \frac{0.012D}{m}\frac{\rho s \mu^{0.5}}{f_n}\\ &= 2.122 \times 10^{-4} \times 0.039\\ &= 8.47404 \times 10^{-6}\\ \delta &= \max(\delta_1 \delta_2)\\ &= \max\left(9.617 \times 10^{-3}, 8.47404 \times 10^{-6}\right)\\ &= 9.617 \times 10^{-3}\end{aligned}$$

Correlations of Pettigrew, Rogers and Axisa.

$$\begin{aligned}D/D^* &= (1.7P/D)^{-1}\\ &= (1.7 \times 1.25)^{-1}\\ &= 0.47\end{aligned}$$

$$\begin{aligned}\varepsilon = &\left(\pi/\sqrt{8}\right)\left(\frac{1 + (D/D^*)^3}{\left(1 - (D/D^*)^2\right)^2}\right)\left(\rho s D^2/m\right)\left(2\nu/\pi f_n D^2\right)^{0.5}\\ &+ \left(\frac{N-1}{N}\right)\left(\frac{22}{f_n}\right)\left(\frac{\rho s D^2}{m}\right)\left(\frac{t_b}{L_i}\right)^{0.6}\end{aligned}$$

$$
\begin{aligned}
&= \left(\pi/\sqrt{8}\right)\left(\frac{1+0.47^3}{\left(1-0.47^2\right)^2}\right)\left(1000 \times 0.0191^2/1.08\right) \\
&\quad \left(2 \times 10^{-5}/\pi \times 33.84 \times 0.019^2\right)^{0.5} \\
&\quad + \left(\frac{7}{8}\right)\left(\frac{22}{6.276}\right)\left(1000 \times 0.0191^2/1.08\right)\left(\frac{6}{0.447}\right)^{0.6} \\
&= 0.0359 \\
\delta &= 2\pi\varepsilon n \\
&= 2 \times \pi \times 0.035 \\
&= 0.2258
\end{aligned}
$$

Step 4:
Calculate the cross flow velocity U.

$$U = 0.15\,\mathrm{m/s}\ \text{(TEMA standard)}$$

Step 5:
Check for vortex shading S_u:

(a) Calculate Strouhal number:

$$S_u = 1/1.73 \times 1.25 = 0.469$$

(b) Calculate vortex shedding frequency:

$$
\begin{aligned}
f_s &= \frac{S_u U}{D} \\
&= \frac{0.469 \times 0.15}{0.0191} = 3.683
\end{aligned}
$$

(c) Check:

(1) $0.8\, f_s < f_n < 1.2\, f_s$
$2.9464 < 6.2706 < 1.2 \times 3.683$
$2.9464 < 6.2706 < 4.4196$
(Not resonance)

(2) $\dfrac{Df_n}{U} > 2S_u$
$\dfrac{6.2706 \times 0.091}{0.15} > 2 \times 0.462$
$0.798 > 0.924$ (Not acceptable)

(3) $\dfrac{Df_n}{U} < 0.2S_u$

$0.798 < 0.0924$ (Not acceptable)

Au-Yang et al. criteria
Calculating the reducing damping C_n

$$\begin{aligned} C_n &= 4\pi \in M_n/D^2 \rho s \\ &= 4\pi \times 0.0359 \times 1.08/\left(1000 \times 0.0191^2\right) \\ &= 1.33555 \\ C_n &< 64 \\ 8.484 &< 64 \end{aligned}$$

$$\text{(a) } C)\frac{U}{Df_n} < 3.3$$

$$\frac{0.15}{6.2706 \times 0.0191} < 3.3$$

$$1.54 < 3.3$$

(Check OK)

Maximum deflection due to vortex shedding at resonance.

$$\begin{aligned} C_L &= 0.091 \\ Y\text{max} &= \frac{\rho s U^2 D C_L}{4\varepsilon f_n^2 m^2 \pi^3} \\ &= \frac{0.091 \times 1000 \times 0.0191 \times 1}{4 \times 0.0359 \times 1.08^2 \times 6.2706^2 \times \pi^3} \\ &= 1.915 \times 10^{-4} \end{aligned}$$

Check: If $Y_{\max} < 0.02D$

$$1.915 \times 10^{-4} < 0.02 \times 0.0191$$

$$1.915 \times 10^{-4} < 3.82 \times 10^{-4}$$

(Accept OK)

Step 6:
Turbulence-induced excitation

a. Determine $C_R(f) = 2 \times 10^{-3}$
b. The mean square response for a pinned-pinned span is given by

$$
\begin{aligned}
y_{\max}^2 &= \frac{\left[\rho s U^2 D C_R(f)\right]^2}{128 \varepsilon f_n^2 m^2 \pi^3} \\
&= \frac{\left[12 \times 10^{-3} \times 1000 \times 1 \times 0.019\right]^2}{128 \times \pi^3 \times 0.022 \times 33.84^2 \times 1.03^2} \\
&= 1.802 \times 10^{-11}
\end{aligned}
$$

$$
\begin{aligned}
y^{2rms} &= \sqrt{\bar{y}_{\max}^2} \\
&= 4.2460 \times 10^{-6} \\
4.2460 \times 10^{-6} &< 0.254 \times 10^{-4} \quad \text{(O.K.)}
\end{aligned}
$$

Step 7:
Fluid elastic instability

(a) *Calculating reduced velocity parameter*

$$
\begin{aligned}
\chi &= \frac{m\delta}{\rho s D^2} \\
&= \frac{1.08 \times 0.2258}{1000 \times (0.0191)^2} \\
&= 0.6687
\end{aligned}
$$

(b) *Calculating reduced velocity by*

$$
\begin{aligned}
\frac{U_{\mathrm{cr}}}{f_n D} &= 8.36(P/D - 0.9)\chi^{0.34} \\
&= 8.36(1.25 - 0.9)0.6687^{0.34} \\
&= 2.729
\end{aligned}
$$

Check:

(c) $U < U_{\mathrm{cr}}$

$$
\begin{aligned}
&0.15 < 2.729 \ \text{(O.K.)} \\
&0.054 < 0.5 \quad \text{(O.K.)}
\end{aligned}
$$

(a) For the particular tube layout

$$
\begin{aligned}
\frac{U_{\mathrm{cr}}}{f_n D} &= K_{\mathrm{mean}}\left(\frac{m\delta}{\rho s D^2}\right)^{0.5} \\
\frac{U_{cr}}{6.2706 \times 0.019} &= 4.5\left(\frac{1.08 \times 0.2258}{1000 \times 0.0109^2}\right)^{0.5} \\
&= 0.4715
\end{aligned}
$$

$$0.15 < 0.4715 \text{ (O.K.)}$$
$$0.3592 < 0.5 \quad \text{(O.K.)}$$

(b) $\dfrac{U_{\text{cr}}}{f_n D} = 4.0\left(\dfrac{m\delta}{\rho s D^2}\right)^{0.5}$

$$\begin{aligned} U_{\text{cr}} &= 6.2706 \times 0.0191 \times 4 \times 0.408^{0.5} \\ &= 0.3916 \end{aligned}$$

Check:

$$U < U_{\text{cr}}$$
$$0.15 < 0.3916 \quad \text{(O.K.)}$$
$$\frac{U}{U_{\text{cr}}} < 0.5$$
$$0.383 < 0.5 \quad \text{(O.K.)}$$

(c)

$$\begin{aligned} \frac{U_{\text{cr}}}{f_n D} &= 2.1\left(\frac{m\delta}{\rho s D^2}\right)^{0.5} \\ U_{\text{cr}} &= 6.2706 \times 0.0191 \times 2.1 \times 0.408^{0.5} \\ &= 4.358 \end{aligned}$$

Turbulent buffeting:

$$\begin{aligned} f_{tb} &= \frac{U}{DX_1X_t}\left\{3.05\left(1 - \frac{1}{X_t}\right)^2 + 0.28\right\} \\ &= \frac{0.15}{0.0191 \times 1.25 \times 1.25}\left\{3.05\left(1 - \frac{1}{1.25}\right)^2 + 0.28\right\} \\ &= 2.5078\,\text{Hz} \end{aligned}$$

7 Output of Program

Case no: 01

Input parameter of program
Mode shape: 1
Mass per unit length (kg/m): 1.08
Tube outside diameter (m): 0.0191
Shell-side velocity: 0.15

Angle of inclination (deg): 30
Structural damping: 0.0359
Fluid density (kg/m^3): 1000
Kinematic viscosity of shell-side fluid (m^2/s): 10^{-5}
Distance between tube supports (m): 0.447
Modulus of elasticity: 110×10^9
Moment of inertia for bending (m^4): 2.89×10^{-9}
Value of lambda (frequency constant): 3.1416
Set natural frequency? [Y/N]: 'y'
Natural frequency: 6.2706.

(1) **Vortex shedding response**:

(i) Value of vortex shedding frequency is 3.683 Hz and velocity of shell-side fluid = 0.15 m/s.
(ii) Value of Strouhal number is 0.465 at pitch ratio = 0.125 with respective 30° layout pattern.

Damping Factor: 0.2256 Reduced damping: 1.3356
Natural Frequency: 6.2706 Hz

(2) **Turbulent buffeting response**:

(i) Fig. 2a shows graph which is same as Fig. 1a which shows the graph vortex shedding vs flow velocity.

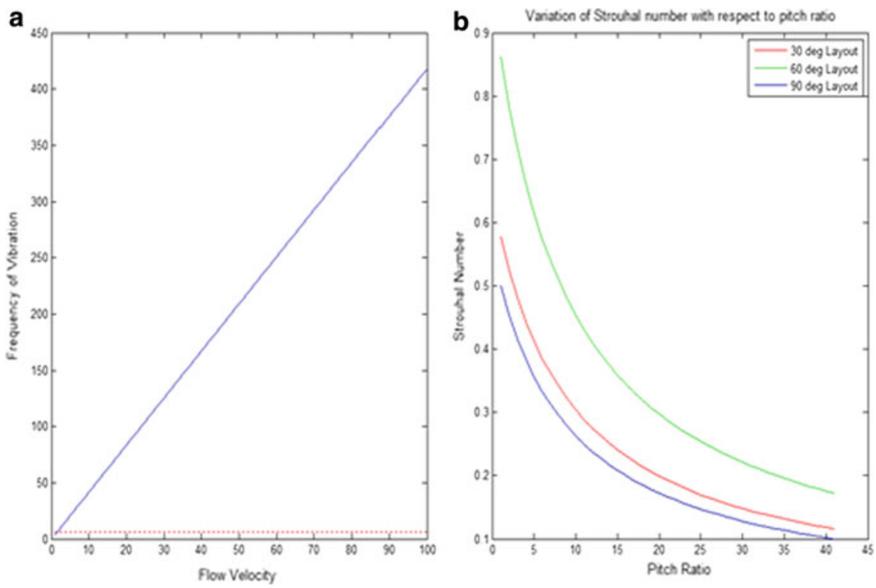

Fig. 1 **a** Frequency of vibration versus cross flow velocity. **b** Strouhal number versus pitch ratio

(ii) Fig. 2a shows the graph of turbulent buffeting frequency on the Y-axis and diameter on the X-axis. The value of buffeting frequency = 3.8237 Hz at diameter 0.0191 m.

$$\text{Damping Factor: } 0.2256 \quad \text{Buffeting Frequency } 3.8237\,\text{Hz}$$
$$\text{Natural Frequency: } 6.2706\,\text{Hz}$$

(3) **Fluid elastic instability response**:

(i) Fig. 3a shows the graph of the critical velocity vs fluid elastic instability (zeta). The value of critical is equal to 4.649 m/s at the fluid elastic instability parameter $(K) = 4.50$.

(ii) Fig. 3b shows the graph of the critical velocity vs fluid elastic instability (zeta). The value of reduced is equal to 3.569 m/s (near about) at the fluid elastic instability parameter $(K) = 4.50$.

$$\text{Damping Factor: } 0.2256 \quad \text{Through Critical Velocity: } 4.6496\,\text{m/s}$$
$$\text{Natural Frequency: } 6.2706\,\text{Hz}$$

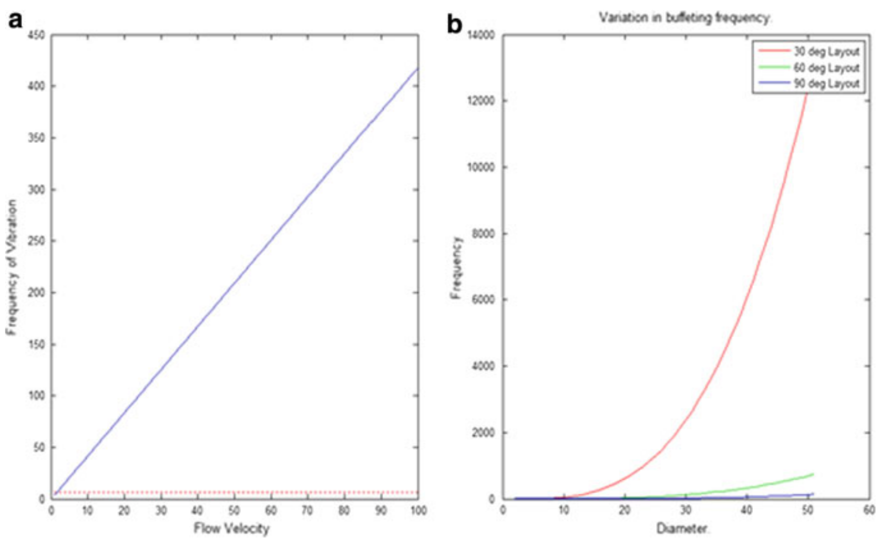

Fig. 2 **a** Frequency of vibration versus flow velocity. **b** Turbulent frequency versus diameter

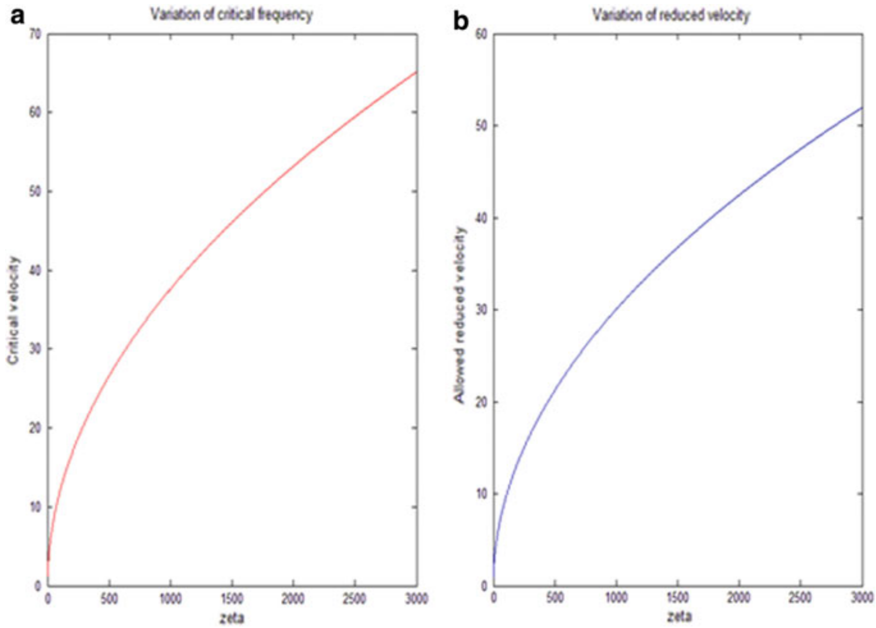

Fig. 3 **a** Critical velocity versus Zeta (K). **b** Reduced velocity versus Zeta (K)

8 Result and Interpretation

Table 1, it is found that the system is free from flow-induced vibration, and results are in good agreement with the available literature data (Table).

Table 1 Result of Case no: 01 (From the available literature)

Parameter	Result of MATLAB program	Result of the available literature
Damping factor	0.2256	0.2256
Reduced damping	1.3356	1.3356
Natural frequency (Hz)	6.2706 (ANSYS)	6.2706 (ANSYS)
Velocity of cross flow (m/s)	0.15	0.15
Through critical velocity (m/s)	4.5 (K $= 4.5$)	4.358 (K $= 4.5$)
Buffeting frequency (Hz)	3.8237	2.52078

9 Conclusion

This work outlines flow-induced vibration response such as vortex shedding, turbulent buffeting, and fluid elastic instability of shell and tube heat exchanger single straight tube. Compassion between MATLAB program and the available literature results is listed down:

1. Damping factor, reduced damping and values are same.
2. Critical velocity value for MATLAB program is 4.5 and 4.358 m/s for the available literature when $K = 4.5$; the results are near about same.
3. Buffeting frequency values for MATLAB program and the available literature are 3.8237 Hz, 2.52078 Hz, respectively. Both values of frequency are less than natural frequency value than it is acceptable value for working conditions.

To check flow-induced vibration response based on the vortex shedding, turbulent buffeting, fluid elastic instability by using MATLAB code and results are in accordance with the available literature data. Since the fretting wear of tubes can cause failure of the equipment leading to heavy loss of life and money, this program can be used as an effective design tool at the initial stages to model conservative design of heat exchangers.

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