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Development of an Experimentation Facility for the Analysis of Flow Induced Vibration in the Parallel Triangular Tube Array

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ABSTRACT: The failure of heat exchanger tube bundles is a common problem occurring in the function of shell tube heat exchangers and it is mainly observed due to the existence of flow induced vibrations. Consideration of this kind of vibration at the early design stage of heat exchanger helps to ensure better performance and reliable life of the heat exchanger. The current paper aims to Development of experimentation facility for the analysis of flow induced vibration in parallel triangular tube array subjected to air cross flow. Critical velocity at instability was calculated theoretically by Connor's equation. Connors equation indicates that critical velocity depends on natural frequency and logarithmic decrement. Natural frequency and logarithmic decrement of the test tube with different fin geometry was calculated by modal analysis and harmonic analysis using ANSYS 19.0. Modeling of the tube was done using CATIA software, also the significance of an effective diameter was taken under consideration. For the study, plain tube and finned tubes were used. Tubes with spiral fins were considered also for analysis different fin densities and different fin heights were considered. Also, their respective array patterns are considered triangular array geometry. The research outcomes help to study the effect of pitch ratio, tube array pattern, and fin density on the instability threshold which also helps in the future to compare the results between the theoretical and experimental procedure.

KEYWORDS: Critical Velocity, Effective Diameter, Natural Frequency, Logarithmic Decrement, Connor's Equation, Finned Tube.

I. INTRODUCTION

Shell and tube type of heat exchangers is the more often used automated process devices utilized in many nuclear power plants, petrochemical, process industries, and food industries. One of the major concerns of process engineers, designers, and operators while dealing with heat exchangers is the oscillations of the tube due to outside fluid flow across the tube bundles. Tubes vibrate when subjected to cross-flow of fluid and these vibrations are proportional to the velocity. These vibrations produce motion of higher amplitudes causing 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.

Though heat exchangers with very high cross-flow velocity give maximum efficiency, it results in flow-induced vibrations. There are different mechanisms responsible for flow-induced vibration viz. Vortex shedding, acoustic resonance, turbulent buffeting, and fluid elastic instability. Many researchers have studied in detail various mechanisms responsible for flow-induced vibration on the plain tube with different tube array geometry, for different pitch ratios and fluid flows. (Cooner's, 1980) (mitra, dhir, & catton, 2009) (mitra et al., 2009) the experimental and numerical investigation has been carried out for different pitch ratios ranges from 1.2 to 1.7 for normal triangular, parallel triangular. To identify critical flow velocity which is responsible for flow-induced vibration, different types of fluid flows i.e. single-phase flow and two-phase flow consisting of water, air, air-water, water-steam also different refrigerants are used. From the last two decades, flow-induced vibration on finned tube arrays has been studied. Compare to a plain tube, finned tube arrays are more entanglement for analysis. In the current paper effect of different

finned tube arrays with fin geometry and tube geometry on the critical flow, velocity has been reviewed. Tube material, tube pitch to diameter ratio, fin types, fin thickness, fin pitch, fin height, and fluid type are considered as parameters and their effects on stability threshold are discussed. In this paper, theoretical calculations of critical velocity at instability were done based on Connor’s equation. The finned tube of 19.05mm base diameter with different fin geometry was considered. Input parameters required for theoretical design were calculated using available literature. Natural frequency and logarithmic decrement were calculated by numerical analysis. Based on the results of the theoretical design, an experimental setup was designed. Detail design of each component was discussed here. Due care was taken so that testing of different array patterns possible.

II. DESIGN OF EXPERIMENTAL SETUP

Fluid elastic instability of finned tube array subjected to air cross flow, an experimental setup was developed. It consists test section of the wind tunnel. For getting results with a minimum flow rate, a small convergent-divergent section is introduced which minimizes the required flow rate as well as the number of tubes required.

Flow loop

The delineative representation of the flow loop is as shown in fig 1. The flow loop consists of a wind tunnel consisting of the honeycomb structure, convergent-divergent section, and test section. The size of the test section is 2m×0.6m×0.6m. Wind tunnel having a capacity of 50m/s. The pneumatic flow meter is used to measure the flow rate of air without any frictional loss. It consists $\pm 0.5\%$ of accuracy for a flow rate above 160 lpm. and $\pm 5\%$ for the flow rate between the range 10-160 lpm. this flow meter has a flow velocity range of 0-12m/s and pressure range of 0 – 10 bar which is within the limit of measurement. The flow rate of air is controlled by a control valve operated manually. The main function of this valve is to regulate the flow rate of air coming across the test section and gradually increase the flow rate as needed for testing.

Test Section

To get optimum results, the test section is further equipped with three sub-assemblies: (1) entrance section, (2) rectangular test section, and (3) exit section. A schematic of each of the sections is shown in Figure 2. The test section assembly is constructed from 12 mm wooden sheets and weighs approximately 35-40 kg. The entire assembly is mounted on a booming support structure. Vibration isolation rubber pads are used at the joints to isolate vibrations of the frame that escort the water flow from those of the test section and tube array.

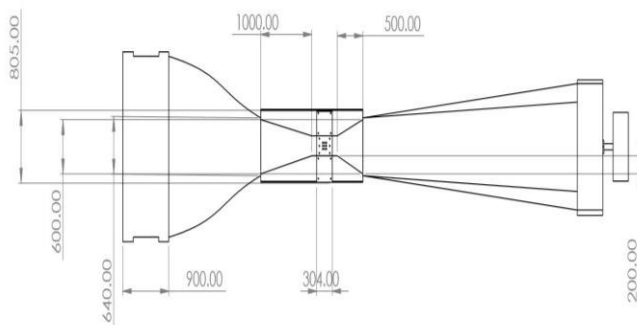


Fig.1. Schematic Diagram of Experimental Setup.

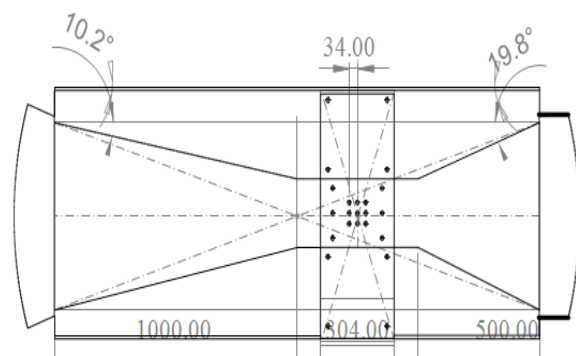


Fig.2. Schematics Representation of Test Section Assembly.



Fig.3. Experimental Test Setup.

According to (Connors, 1978) in fluid elastic and self-excited vibrations critical velocity of fluid flow can be calculated by equation 2.1.

$$\frac{V_c}{f_n D_e} = K \left(\frac{m \delta}{\rho D_e^2} \right)^{\frac{1}{2}} \dots\dots\dots (2.1)$$

where ‘K’ is threshold instability constant also known as Connors constant depends on tube pattern and tube spacing.

In the above equation, $\frac{V_c}{f_n D_e}$ the reduced velocity factor and $\left(\frac{m \delta}{\rho D_e^2} \right)^{\frac{1}{2}}$ is a mass damping parameter. From the above equation, critical velocity ‘V_c’ is directly proportional to the natural frequency of tube ‘f_n’, the mass of tube ‘m’ and logarithmic decrement of finned tube ‘δ’. The effective diameter ‘D_e’ is the mean diameter of the tube considering the effect of the fin.

Tubes selected for experimentation:

The test section was designed to be flexible for the contrasting tube types and array geometries. For the study, plain tube and finned tubes were used. Tubes with spiral fins and plane tubes were considered for the design of the experimental setup as shown in fig4.



Fig.4. Plain and Spiral Fines Tube With Different Fin Geometry.

To maintain the required stiffness and natural frequency, testing tubes are connected with a small threaded bolt. The tube array pattern was kept fixed at one end and the other end is free to vibrate for all tubes. The detailed specification of the tube array pattern considered is given in Table 2.1.

Table 2.1: Tube array specification:

Sr.No	Title	Specification	Sr.No	Title	Specification
1	Tube material	Stainless steel	7	Fin thickness, mm	0.5
2	Fin material	Stainless steel	8	No. of tubes	9
3	End condition	Cantilever	9	Fin height, mm	3 and 6
4	Tube O.D, mm	19.05	10	Fin pitch, mm	8.467 and 2.822
5	Tube I.D, mm	15.05	11	Finning length, l, mm	590
6	Fin type	Spiral	12	Tube pitch in mm	34

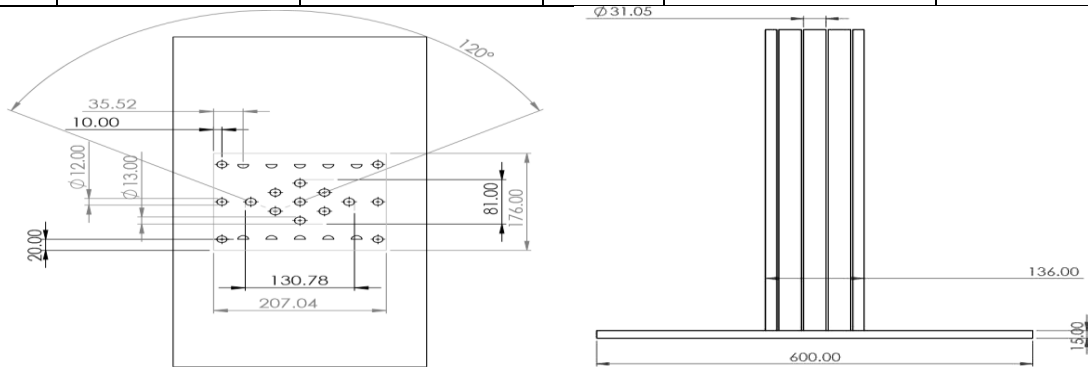


Fig.5. Schematic Diagram of Tube Array Pattern.

The main constraint of the design of the flexible tube pattern was to maintain pitch ratio as little as possible so that the experimentation could be done for the design of a heat exchanger which applicable for nuclear reactors. For the current experimental design tube with a 19.05mm base diameter was selected. Array with nine finned tubes with 34mm pitch was considered. Fin pitch for Coarse fins with 8.467mm (3fpi) and fine fins with 2.82mm (9fpi) was considered. Pitch of Here, fin thickness t_f was kept constant as 0.5mm and fin height h_f varied as 3mm and 6 mm. Tube preparations consist of a tube with the threaded rod at one end and another end is free. The rod length is the parameters used to tune the tube to the desired natural frequency. Also, the rod will reduce the effective stiffness of the tested tube to better simulate the natural frequencies of the much longer tubes used in service.

From the above input parameters, the initially effective diameter of the fin was calculated. Two different methods exist for the calculation of the effective diameter of the finned tube. The first one is based on the projected front area of fin and tube which is called a volumetric diameter of the tube given by the equation. (Hirota, Nakamura, Kikuchi, Isozaki, & Kawahara, 2018)

$$D_e = \sqrt{(D_f^2 - D_b^2) \frac{t_f}{p_f} + D_b^2} \dots \dots \dots (2.2)$$

The second method is also based on the volume of fin proposed by (Mair, Jones, & Palmer, 1975) effective diameter of the tube can be given by

$$D_e = \frac{1}{p_f} [t_f(D_f - D_b) + p_f D_b] \dots \dots \dots (2.3)$$

Where D_f finned tube diameter, D_b outer diameter of the tube, t_f fin thickness, and p_f fin pitch intensity. Here both methodologies used are based on plain fins with no serrations or twists. Furthermore, the use of these “effective diameters” essentially assumes that the effects of fins can be putative simply by augmenting the diameter of a base tube. (Arshad.H., Khushnood.S., Ahmad Nizam.L., Ameer Ahsan.M., & Ghufraan Bhatti.O.2018) (Wang & Weaver, 2010) studied fluid elastic instability from finned tube arrays and found good results. The use of Mair et al.’s effective diameter proved to be a useful predictor for vortex shedding from these finned tubes. In the current work, tube Mair et al.’s effective diameter was considered. The detailed selections of geometric parameters are given in Table 2.2.



Table 2. 2: Geometric Parameters of Finned Tubes:

Sr. No	Base tube Diameter (D)	Fin Tube Diameter (D _f) mm	Fin thickness (t _f) in mm	Fin Pitch (p _f) in mm	Effective Diameter (D _e) in mm.	Pitch Ratio (P/D _e)
1	19.05	25.05	0.5	8.467	19.41	1.75
2	19.05	25.05	0.5	2.822	20.13	1.69
3	19.05	31.05	0.5	8.467	19.77	1.72
4	19.05	31.05	0.5	2.822	21.17	1.61

III. EXPERIMENTAL TEST PROCEDURE

To start with the actual test, the outer plate with plain tubes was prepared first with the plain or finned tubes array pattern at a pitch ratio of 1.79. Then plain or finned tube array is added inside the test section and actual testing with plain or finned tube bundles will be carried out as follows;

- 1 Experimentation start with the flow rate is set to 2m³ / sec and the constant flow rate is achieved with the help of the control valve. The same process is repeated for all incremental flow rates in the step of 2 m³ / sec up to 35 m³ / sec
- 2 As soon as the steady flow rate is obtained the vibration results of the tubes are documented in the form of data centers and the data will be listed for at least 4-5 minutes to get uniform and uninterrupted constant readings.
- 3 The accelerometer (DYNAMIX 6503) will be placed on top of one of the last tubes to measure the magnitude of the vibration as air flows over the tubes.
- 4 Data is obtained through the data logger after the signal conditioner passes and DEWESoft 7.1.1 will be used to further use the data obtained from the accelerometer.
- 5 Vibration amplitudes are a percentage set against the reduced velocity that will be obtained to identify critical velocity in the assumed tube array instability.

Measurement Methodology:

The parameters obtained by free vibration analysis earlier will be used to evaluate the FEI in a normal square tube array pattern. The other initial parameters are required for results analysis and this analysis is done for 1.79 pitch ratio.

Effective diameter

The effective diameter is calculated by equation (3.1);

$$D_{eff} = [t/p (D_f - D) + pD] \dots \dots \dots (3.1)$$

Where **p** is the pitch of fin in mm, **t** is the thickness of fin, **D_f** is the diameter of fin, and **D** is the bare tube diameter.

For 3 mm fin height and fin pitch p = 8.467 mm

$$D_{eff} = [0.5/2.82(25.05 - 19.05) + 2.822 \times 19.05]$$

$$D_{eff} = 19.404 \text{ mm}$$

For 6 mm fin height and fin pitch p = 8.467 mm

$$D_{eff} = [0.5/2.82(31.05 - 19.05) + 2.822 \times 19.05]$$

$$D_{eff} = 19.759 \text{ mm}$$

For 3 mm fin height and fin pitch p = 2.822 mm

$$D_{eff} = [0.5/8.47(25.05 - 19.05) + 8.467 \times 19.05]$$

$$D_{eff} = 20.113 \text{ mm}$$

For 3 m fin height and fin pitch p = 2.822 mm

$$D_{eff} = [0.5/8.47(31.05 - 19.05) + 8.467 \times 19.05]$$

$$D_{eff} = 21.176 \text{ mm}$$

Hydrodynamic Mass:

When the body is accelerated in the fluid due to excitation, a large amount of the fluid moves around it. Thus, more forces are required to accelerate the body through the fluid than in a vacuum. Since force equals mass times acceleration, the additional force in terms of an imaginary added mass of the object acts in the fluid. This imaginary added mass is called as hydrodynamic mass. [8];

In this current analysis methodology, the tubes are fixed at one end in cantilever condition and fluid is moving around the tubes. Therefore, the additional effect of the hydrodynamic mass has to be considered. Theoretical added mass due to fluid flowing around the tube is evaluated from the following equation (3.2) presented in the work of Pettigrew et al. [9];



$$m_h = \left(\frac{\rho_f \pi D_{eff}^2}{4} \right) \frac{\frac{D_e^2}{D} + 1}{\frac{D_e^2}{D} - 1} \dots\dots\dots (3.2)$$

Where, **mh** is the hydrodynamic mass, **ρ** is the fluid density, **Deff** is the effective diameter of tubes, and **De** is the equivalent diameter to model the confinement due to surrounding tubes. Value of it is based on tube array pattern used. For rotated and normal square tube array pattern, the confinement factor is derived by [10&11] and is given by equation (3.3);

$$\frac{D_e}{D} = \left(0.96 + 0.50 \frac{P}{D_{eff}} \right) \frac{P}{D_{eff}} \dots\dots\dots (3.3)$$

Where, P/Deff is the effective pitch ratio.

Pitch ratio = P/D = 34/19.05 P/D=1.79

a) The effective fin pitch ratio for 3mm Height and pitch 3fpi

$$\frac{p}{D_{eff}} = \frac{34}{19.404} = 1.752$$

$$\frac{D_e}{D} = \left(0.96 + 0.55 \frac{P}{D_{eff}} \right) \frac{P}{D_{eff}}$$

$$\frac{D_e}{D} = (1.07 + 0.56 * 1.39) 1.39$$

$$\frac{D_e}{D} = 3.217$$

b) By similar calculations the effective fin pitch ratio for 6mm Height is 1.65 Therefore, hydrodynamic mass for 3mm- 3fpi is

$$m_b = \frac{1000 \times \pi \times (19.404 \times 10^{-3})}{4} \times \frac{3.217^2 + 1}{3.217^2 - 1}$$

$$m_b = 0.0194 \text{ kg/m}^2$$

Similar, calculations are made for the remaining tubes for getting the hydrodynamic mass, listed in Table 3.1 in calculation mass of tube consist summation of the hydrodynamic mass of tube and structural mass of tube.

$$m = m_t + m_h$$

The actual masses of the plain tube, 3mm, and 6mm height tubes are 0.781kg/m, 1.698 kg/m, and 2.679 kg/m respectively. The mass due to fluid in contact, actual structural and total masses are summarized in Table 3.1.

Table 3.1: Total Mass for Plain and Finned Tubes

Tube dimentions	mt kg/m	mh kg/m	m kg/m
Plain tube	0.8429	0.0189	0.8618
Finned tube 3mm-3fpi	0.9269	0.0194	0.9463
Finned tube 3mm-9fpi	1.0126	0.0200	1.0326
Finned tube 6mm-3fpi	1.0996	0.0206	1.1202
Finned tube 6mm-9fpi	1.3695	0.0224	1.3919

Reduced Velocity:

Reduced velocity is a dimensionless number and is expressed as the ratio of critical gap velocity to the product of natural frequency to diameter. Therefore,

$$\text{Reduced Velocity} = Vc/fnD_{eff}^2$$

IV. CONCLUSION

The current paper discussed a detailed procedure for the devlllopment of the experimentation facility for the analysis of flow-induced vibration. The design procedure is dependent on Connor’s equation. For analysis perposeee five differrent tube combinations were considered. This makes developed experimental procedures adaptable for all four types of tube array patterns and also suitable for plain tube as well as finned tube array. In this paper based on Connors equation, detailed calculation of all parameters was done following the standard equation. This makes the setup more reliable and expected results will be more accurate. Auther also explained the proposed experimentation

procedure which will help for the utilization of the research facility. Furthermore, actual testing of different tube arrays and positive results will be obtained in the future.

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