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Analysis of vibration amplitude as a function of excitation frequencies in an existing beam and shell heat exchanger

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ABSTRACT/RESUME

Abstract: *This work aims to contribute to understanding the parameters influencing the amplitude of vibrations in the variation of excitation frequencies in a core and shell heat exchanger. Vibrations produced by fluid-structure interaction are typical and originate in the functioning of systems, where they manifest a malfunction but can be a potential source of damage. Shell and tube bundle exchangers in the petroleum industry are constantly exposed to this kind of vibration problem due to Karman vortex force frequencies. For this case study, the natural frequencies of the vacuum tube of 36.993 Hz, or at the extreme operation of 36.927 Hz, can generate vibrations compared to the interaction fluid-elastic instability, which is very low in our case of 0.0608, under flow with a critical operating speed of 734.412 cm/sec.The good vibration program analyzed vibration amplitudes induced by variation of excitation frequencies under license available to the oil company SONATRACH in Arzew. To draw the amplitude profiles for a core and shell heat exchanger existing in this oil refinery, we have integrated into the program all the data specific to this exchanger, and we have imposed different excitation frequency values, with values lower and higher than those specific to the tube.The simulation results showed that the vibration amplitude is low when the excitation frequency values are lower than the natural tube frequency of the case studied; these low amplitudes cause cuts in the beams by the baffles****.*** *The amplitude greater than half the distance between the pipes leads to the collision problem. A good distribution of amplitudes and harmonic profile can be observed for frequencies greater than or double the natural frequency of the tube. These vibration amplitudes* *may be beneficial in creating flow turbulence, which directly influences the exchanger's performance in DTLM, thus reducing the impact of fouling and extending the life of the exchanger. For the frequencies studied, the value exceeding 120 Hz can have a dangerousness impact of 117.6%. We could that the Good Vibrations program, which is very effective in pronouncing the influence of external and mechanical forces giving vortex frequencies in heat exchangers and analysis of their direct impact on the proper functioning of this essential device in the oil industry. Acoustic vibrations and sound frequencies are not considered in our work.*

1. Introduction

The vibration of the elastic bundle heat exchanger induced by the internal fluid can improve the heat transfer between the shell and tube side media at different temperatures and open a new research direction [1].

A great deal of literature has been dedicated to the study of the elastic tube bundle heat exchangers variation, which found that the vibration responses of the elastic bundle can improve the heat transfer coefficient, which is three times that of the rigid bundle[2]. Also, the performance of plane-bending elastic tubes under vibration conditions is about 2.6 times higher than that under non-vibration conditions[3].Others have held that the vibration frequency increases monotonically as the void fraction of the two-phase flow increases driving, a rotating triangular tube array. Tan et al.[4] concluded that a low water inlet velocity causes the tube bundle to have a high vibration amplitude based on water tunnel experiments; the analysis of turbulence on the flow-induced vibration response of the parallel triangular tube bundle and turbulence strongly affecting the frequency response of the tube bundle[5]. Suppose curved elastic tube horizontally to improve heat transfer [6].

Another study shows that the hull side velocity significantly influences the vibration frequency of the stainless steel connection, and the low-velocity fluid-induced elastic beam vibration has a harmonic frequency. The elastic tube bundle was made of copper, and stainless steel was used to connect the tube bundle. The stainless steel connector is not easy to install, and easy to corrode and rust for a long time [7].

2. Phenomenon of vibration in the bundle and shell exchangers

Shell-and-tube heat exchangers are widely used in oil refineries and other significant chemical processes. In this model, two separated fluids at different temperatures flow through the heat exchanger, one through the tubes (tube side) and the other through the shell around the tubes (shell side). Several design parameters and operating conditions influence the optimal performance of a shell and tube heat exchanger. This can serve as a foundation for more advanced applications that involve analyzing parameters and considering additional factors such as corrosion, thermal stress, and vibration [7, 8].

The interaction between fluid force and elastic displacement is called fluid elastic excitation. It is usually produced when there are other mechanisms

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to induce the movement of the pipe. For example, when the fluid amplitude is not too large, it will increase until the fluid amplitude collides.

The vibration will persist even when the fluid velocity drops below its original speed. Research shows that when the fluid velocity is low, the vibration may be caused by vortex shedding or turbulent buffeting, while in the high-velocity region, the induced vibration mechanism is mainly fluid excitation [9.10].

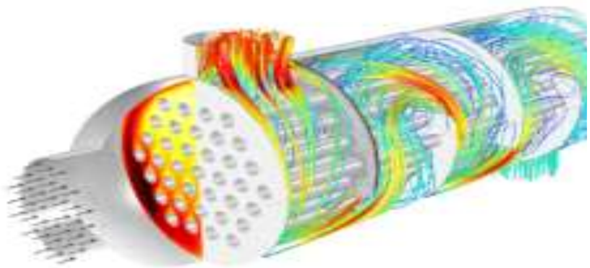

Figure 1. *COMSOL design of heat exchanger*.

Turbulence induces vibrations. The higher flow velocity in the tube bundle promotes heat transfer in the fluid, but the heat exchange tube responds randomly to vibrations caused by turbulence ***[****10,7,11****].***

In addition, flow turbulence will also promote and enhance the formation of other vibration mechanisms, such as vortex separation at the floor. Turbulent flow is more random [12, 13].

The heat exchanger vibration problem we study refers to the vibration of the tube bundle.

Tube bundle vibration is caused by interference force or exciting force. These exciting forces can be divided into two categories: mechanical exciting forces and fluid-induced elastic exciting forces. The excitation frequency can usually be accurately predicted and corresponding measures can be taken to prevent it[14, 15].

2.1.Excitation mechanisms

2.1.1. Fluid-elastic instability

This is by far the most dangerous excitation mechanism in heat exchanger tube bundles and the most common cause of tube failures. Fluid-elastic instability forces are a result of the body's motion. Price, [16] has presented a comprehensive review of fluid-elastic instability of cylinder arrays in cross-flow. According to Price, the nature of fluid-elastic instability can be illustrated as a feedback mechanism between structural motion and the resulting fluid forces[17]. A small structural displacement due to turbulence alters the flow pattern, inducing a change in fluid forces. This in turn, leads to further displacement, and so on. If the displacement increases (positive feedback), then fluid-elastic instability occurs. Three mechanisms [16] that enable the cylinder to extract energy from the flow: require a phase difference between cylinder displacement and fluid force generated relies at least two-degrees of freedom with a phase difference between them because of non-linearities, the fluid force is hysteretic and its magnitude depends on the direction of cylinder motion.

A considerable theoretical and experimental research has been undertaken in the past three decades to arrive at a safe and reliable design criteria against fluid-elastic instability. The topic has been reviewed on regular basis from time to time by various researchers including [18, 19, 20, 21, 22, and 23].

2.2 Vibration characteristics of a bundle of the tube

Each component of one heat exchanger vibrates at a frequency natural that can be measured experimentally. The main tube configurations encountered are straight tubes and U-shaped bent tubes. These tubes are considered to be embedded at each of their ends at the level of the plates; they can rely on baffles.

Characteristics of vibration of a U-tube are more difficult to predict than those of a straight tube. One large number of contributions are generally not available. Consequently, a simplified approach is often used. This approach considered straight parts and elbows separately [15].

2.2.1. Natural frequency of tubes

The natural frequency of the heat exchanger tube is an important factor and primary consideration in the design of fluid flow-induced vibration. To calculate this frequency, various models have been used; resonance conditions must be removed ,ensuring the separation of frequencies natural tubing and excitation frequencies [20].

In the case of a straight tube supported or embedded at the ends, the resonant frequencies are well known:

$$F_n = \frac{X_n}{2\pi L^2}\sqrt{\frac{EI}{M}} \quad (1)$$

With: E(N/m ^{2}): Young's modulus of the tube material;

F_n(Hz): The ^{nth} frequency;

I(m ^{4}): Inertia of the tube;

L (m): length between two bindings;

M (kg/m): Linear mass of the tube;

X_n : Numbers depending on the type of connection at the ends.

In the case of a U-shaped tube, the lowest frequency can be associated with a mode of vibration out of the plane of the U. The calculation method is no longer so simple. However, the **TEMA standard** gives the minimum resonance frequency:

$$F_1 = 0.0403\frac{C_u}{R^2}\sqrt{\frac{EI}{M}} \quad (2)$$

With: C_u: Coefficient depending on the distance from the supports to the bent part.

Studies have been made for the calculation of the linear mass of the tube (M) and its stiffness (EI) [21]**,** it is necessary to take into account the characteristics of the tube, whether it is a tube with a circular section insulated in a fluid or not.

2.2.2. Vibration amplitude

Frequencies naturally are characteristics of tubes themselves and are independent of how they are excited. On the other hand, the amplitude depends on the natural frequency tubes, the fashion shape, the frequency and the strength from mechanism excitement, and the system depreciation internal and external [22].

Has weak flow velocities, fluid flow around a given heat exchanger tube acts like adamper, limiting the movement's amplitude [23].

The pipes in many heat exchangers'tubes and shells vibrate, but the amplitude is so small that the vibration poses no problem. For the amplitude to be noticeable, the system's damping needs to be low.

2.2.3. Vibration response of a system

The notions of mass and stiffness, makes it possible to determine the resonance frequencies that describe a conservative system. For sustained excitation at these frequencies, the forces and displacements become infinite. In practice, the system is dissipative given various phenomena [24]:

- Internal material damping;
- Friction and shocks on the supports;
- Viscous damping due to the fluid.

The resulting overall damping cannot be predicted theoretically. However, in specific setups, it is possible to evaluate it. For instance, a bundle of tubes in an exchanger is a highly intricate illustration of this concept. The following number is called critical damping:

$$\varepsilon = \frac{C}{2}\sqrt{EIM} \quad (3)$$

With: C: Coefficient of viscous friction;

EI et$M$: Characteristics of the resonant system.

The movement of the structure depends on the value of ε : $\varepsilon < 1$Damped sinusoidal motion;

➢ $\varepsilon > 1$Exponentially decreasing movement without oscillations. But this parameter is difficult to

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interpret, and we often use the logarithmic decrement δsuch as:

$$ln\delta = \frac{A_n}{A_{n+1}}$$
(4)

$A_n$etA_{n+1} is the amplitude of two successive oscillations.

Another widely used parameter is the boost factor, representing the amplification due to resonance seen by the tubes:

$$Q = \frac{1}{2\varepsilon} \quad (5)$$

for $\varepsilon \ll 1$. These values are conservative and should prevent any damage.

2.2.4. Damping system

The damping system has a strong influence on the vibration amplitude. Once the energy excites the vibration ceases, the vibration amplitude, decreases over time [25]. The speed at which vibration dampens outside is often exponential. The logarithm of the difference of amplitude peaks successive is called the logarithmic decrement δ_0, indicates depreciation. Most tubing in heat exchangers are very slightly amortized structures with low logarithmic decrement values.

Whether energy input can't be dissipated through depreciation, vibration amplitude will increase over time leading to a leakage condition [26]. The parameters influence the damping, such as the tube's movement type, number of supports, frequency, amplitude of vibration, etc. [24].

3. Excitation mechanisms in exchanger tube bundles

The determination of the forces exerted by a flow on the tubes enables to quantify the response of the latter [27]. Fig. 2 provides an example of the vibes induced on a tube bundle depending on the flow velocity (single-phase flow) [28].It highlights the three mechanisms of vibratory excitations. Turbulent forces are present as soon as the flow velocity is non-zero; the forces due to alternating vortices have significant vibratory effects when there is resonance, while the fluid-elastic forces are preponderant at very high flow velocities.

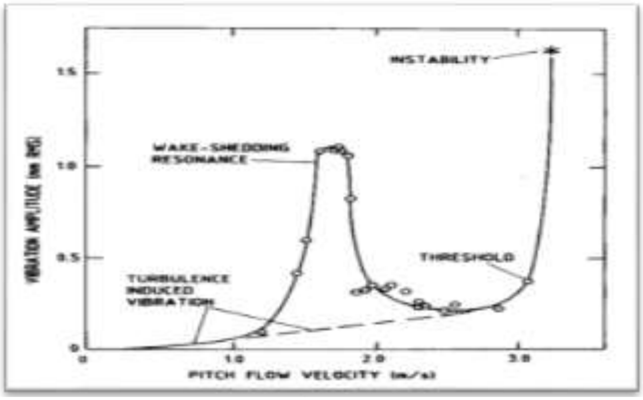

Figure 2. *Vibration response of a tube bundle as a function of flow velocity [28].*

Fig. 3 represents the deviation from the mean of the amplitude of the vibrations under flow of a single cylinder as a function of the dimensionless velocity of the mean flow [29].

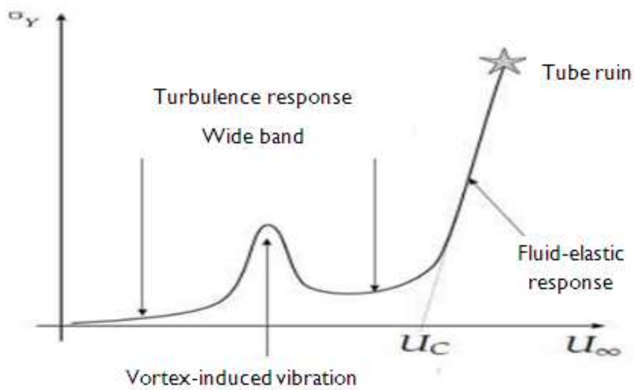

Figure 3. *Evolution of the standard deviation σ_y of the amplitude of the vibrational response of a tube in transverse flow as a function of the mean velocity of the fluid [29].*

The appearance of the different excitation mechanisms depends on the technical parameters. Each of these phenomena is an essential step for the vibration analysis of the systems studied to dimension them to the mechanisms they will be confronted with.

Vibrations are necessary for heat transfer between internal and external circuits, and are a natural part of flow and structure interaction. However, excessive vibrations can harm the structure. To prevent this, anti-vibration bars (AVB) are used in the steam generators of nuclear power plants of the CANDU (CANada Deuterium Uranium) type [30].

3.1 Turbulent excitation

The turbulence of the flow in the tubes is the cause of this form of excitation.

3.1.1. Karman Vortex

As the fluid flows laterally through a single cylindrical object, when it flows through the tube bundle, there is also a Karman vortex behind the tube, as shown in Fig. 4 when the shedding frequency of Karman vortex is equal to the natural vibration frequency of the pipe, the pipe will vibrate violently [30].

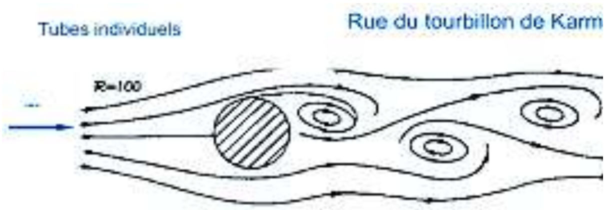

Figure 4. *Karman vortex for a bundle and shell heat exchanger for a single tube [31].*

Von Karman [32] and alternating vortices have been observed [33]. The fluctuation frequency of the drag due to the alternating vortices is twice as large. Their vibratory effect depends on an adimensional parameter, Strouhal number S_t:

$$S_t = \frac{f_w d}{V} \quad (6)$$

With: S_t Strouhal number
f_w Vortex frequency
d Hydraulic diameter
V Fluid velocity

For an insulated tube, the Strouhal number is around 0.2. In contrast, more spread out Strouhal numbers exist for tube bundles.

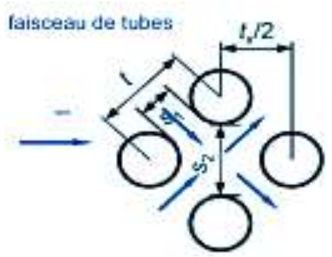

Figure 5. *Turbulence excitation for a bundle of tubes [30].*

3.2. Fluid-elastic instability

Fluid-elastic instability is the most dangerous excitation mechanism in the heat exchanger tube bundle and the most common cause of tube failure [32]. A method of prediction of this phenomenon which uses dimensionless groupings, is developed as follows [33], critical speed parameter:

$$V_r = \frac{U_c}{F_n D_h} \quad (7)$$

Damping parameter:

$$A_r = \frac{M\delta}{\rho D_h^2} \quad (8)$$

With: U_c Critical fluid velocity
F_n Natural frequency of the tube
D_h Hydraulic diameter
M Linear mass of the tube
δ Logarithmic decrement
ρ Fluid density

The unsteady model, which is based on the experimental measurement of irregular fluid forces, can be used in two-phase flows. Authors [34, 35] have already used this model to calculate critical velocity in an aligned tube bundle.

However, this model requires a lot of experimental data. Moreover, [36] reported a weak correlation between forces and displacements of adjacent tubes, thus making it difficult to apply it to a model with several flexible tubes. The quasi-stationary model makes it possible to overcome this difficulty and offers the advantage of costing less in experimental data.

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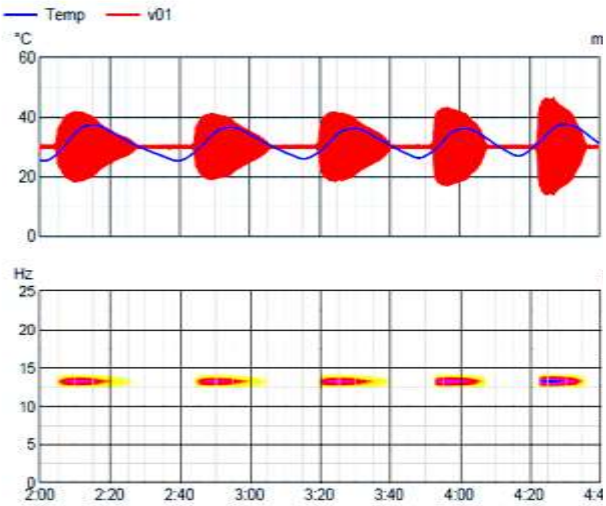

Figure 6. *Profile of vibration amplitudes as a function of time at temperature below 40°C and frequency below 15 Hz [37].*

4. Profile of vibrations

4. 1. Elastic excitation

4.1.1. Formation mechanism: If a tube in a tube bundle suddenly moves from its original position, it can alter the flow field and disrupt the force balance on nearby pipes due to fluid elastic force. This causes those tubes to vibrate at their natural frequency.

When the transverse flow velocity of the fluid reaches a specific critical value, the work done by the total elastic force of the flow on the tube bundle will be greater than the work consumed by the damping effect of the tube bundle, so that the tube begins to vibrate with large amplitude. This vibration is called fluid elastic excitation. The fluid cross-flow velocity that causes the pipe to vibrate with large amplitude is called the critical cross-flow velocity.

5. Simulation of vibration analysis:

This study relied on GV program, which is used for the design and testing of tube bundle heat exchangers with the four standard tube configurations (30°, 45°, 60°, and 90°) regarding the avoidance of short-term damage due to flow-induced vibrations in the beam, which are caused by fluid-elastic instability or vortex excitation in cross-flow. In addition, the peak amplitudes associated to vortex and turbulence excitation are also calculated to estimate the lifetime due to long-term damage.

Pipe structural data can also be calculated with variable mass assignments and material data.

The main routine now includes an algorithm that can calculate straight pipes' natural frequencies and deflection shapes. Additionally, critical velocities for gas and liquid flows in ideal bundles with homogeneous distribution are now calculated using thermal atlas stability diagrams [37, 38].

The program's great advantage lies in considering the distribution of the actual velocities and the variable material and structural data along the axis of the heat exchanger. This is possible thanks to the implementation of analytical models for the approximate calculation of the distribution of velocities according to the position and the flow conditions of the pipes. The critical velocities are also calculated section by section for each row of pipes considering possible oblique flows and the degree of risk for the pipes from the shell/beam edge clearance.

The result is given for

• The row of tubes endangered by the fluid-elastic instability and the associated form of deviation; the following two rows of pipes or Eigen modes at risk are also listed.

• The row of pipes threatened by the generation of eddies or turbulence.

5.1. General product description

• Calculation of velocity distributions in all flow sections of cross-flow devices

• Calculation of all rows of tubes in the bundle of cross-flow devices and determination of the row of tubes most at risk

• Calculation of forms and frequencies of deviation by a rapid analytical solution procedure

• Consideration of higher modes and corresponding Eigen frequencies, deflection shapes, energy ratios, and Strouhal numbers

• Consideration of the influence of the edge distance (duct/bundle) of the respective pipe row

• Determination of the risk of vibration due to turbulence and vortex excitation for all rows of tubes in cross-flow devices and determination of the row of tubes most at risk, taking into account the higher modes and specifying the maximum amplitudes, which are a direct measure of the degree of risk.

The axial velocity distribution is considered, and the vibration mode weights the excitation forces.

In the case of a vortex excitation, the resonance distance, which varies along the axis of the tube, is also accounted for via the frequency ratio and the integral value of an amplitude amplification function.

The size and position of the exit nozzle affect the velocity distribution in the last three rows of tubes in the exit area. The rows of pipes to be calculated according to the risk of vibration are highlighted in color. By default in the program, all rows of heat exchanger tubes are calculated. Individual pipe rows can be calculated. In added, the number of tube rows, the total number of tubes and possibly (in the case of a tube row to be computed) the number of tube rows in the relevant window and the vertical distance of the tube row to be computed from the center of the set are displayed.

The risk of vibration depends on rows of fully-filled tubes that reach the shell, representing the maximum load on the bundle. Therefore, as long as the chord lengths of the pipe rows are considered a match, an exact match between the calculated tube table and the actual pipe length table is not crucial.

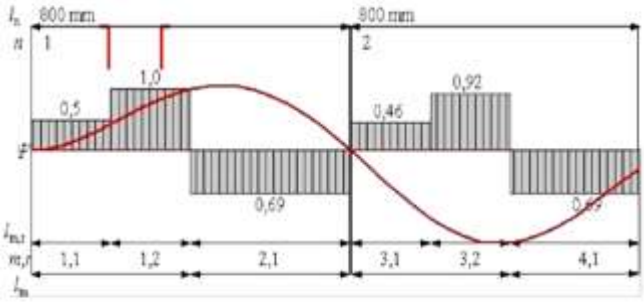

Figure 7. *GV vibration amplitude profile.*

5.2. Speed models

The following have been specified for all rows of tubes and independent of vibration mode:

• Occupation of the total mass or the hydrodynamic mass coefficient for the variable mass reservations;

• Minimum number of tubes supporting axial loads;

• Stiffness of the plate in standard values and for each flow section;

• The leakage flow factor records the cross flow component (also >1 possible): to take into account variations in mass flow rates;

• Dimensionless critical speeds are automatically calculated in the program based on layout, tube row and mode, considering feed conditions and widths. However, the oblique flow must be taken into account in the default.

To perform the standard calculation in the program, it suffices to specify;

• Influx factor that specifies the influx velocity distribution; standard values;

• Edge clearance correction factor, which influences the edge clearance on the grille;

• total damping which is constant for all the modes;

• Partial flow factor (< 1) for the incoming volume flow;

• Correlation factor: for turbulent excitation at a given mode;

• Buoyancy coefficient for vortex excitation;

• Critical speed in the real beam.

5.3. Circulation Fluid Data

The table below resumes the different fluids features.

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Table 1. *Fluid characteristics*

Heat exchangers	tube side	Grille side
circulating fluid	Total Ess	Light solvent
Inlet temperature (°C)	48	115
Outlet temperature (°C)	60	95
Mass flow (kg/h) m and M	m	1.1 10^{5}
Density ρ(kg/m ^{3}) at 15°C	711.8	719.3
Specific heat Cp (kcal/kg.°C)	0.650	0.7
Thermal conductivity λ/λ' kcal/hm°C	0.1065	0.1399
Dynamic viscosity μ / μ'(kg/mh)	0.2124	0.0216
Quantity of heat Q (kcal/h) *10^{6}	1.4868	1.4868
d_{in} tube interior (m)	0.01478	
d_{ex} tube outside (m)	0.01905	
Tube wall thickness e (m)	0.00213	
Number of passes on the grille side/tubes nc/nt	1/2	
Number of tubes N_t	265	
Inner diameter of the grille (m) Din	0.536	
Shell outer diameter (m) Dex	0.556	
Space between baffles (m) B	0.173	
Baffle thickness (m)	0.005	
Baffle diameter (m)	0.533	
Pitch between tubes (m) Pb	0.0254	
Tube length (m) L	3.991	
tube side inlet mass flow m (kg/h)	1.9.10^{5}	
MTLD (°C)	50.9	
Caloric ratioR	1.6	
Heat exchanger efficiency E	0.17	
Correction factor F	0.95	
DTLM corr (°C)	48.35	
Section per pass S / Sc (m ^{2})	0.0227 / 0.0232	
Mass velocity G / G' (Kg/hm ^{2})	8370044 /4741379	
Equivalent diameter D_{eq}(m)	0.043	
Reynolds number *Re/Re'*	582435 / 9438856	
Number of Prandtl *Pr / Pr'*	1.29 / 0.108	
Nusselt Criterion *Nu/Nu'*	1050 / 1289	
local coefficients h_1/ h_2(Kcal/ hm $^{2°}$ C)	5870.07 / 4193	
Coefficient. overall transfer rate U (Kcal/ hm $^{2°}$ C)	2157.86	
exchange surface available As (m2)	29.88	
Transfer resistance shell Rd	2.10^{-4} hm^2 °C/kcal	
Thermal conductivity	39 kcal/ m °k (design data)	

Table 2.*Metallic grades, their mechanical characteristics and chemical composition*

Grades/ exchangers	Nuances HE	
tubing	A179	Seamless,low carbon steel
Grille	A285-C	Carbon steel, under pressure
baffle	Steel	General Purpose Steel
Flanges	A105-1	Forged carbon steel flanges

Table 3. *Material characteristic*

ASTM/ CM	Temp °C	HB hard-ness	Resist-ance N/m^2	Densi-ty
A105-1	-20/ +450	100-130	350-450	7.85

Table 4. *Parameters Nomenclature HE range*

Outer diameter of tubes D0	11.43cm
Tube thickness T	0.66cm
Tube inner diameter Di	10.11cm
Tube length L	399.1cm
Longitudinal pitch of Pl tubes	7.45cm
Transverse pitch of the tube Pt	25.81cm
Arrangement Not for P tubes	14.9cm
Added mass factor Cm	1.53

Table 5. *Span Geometry 2 One end fixed and the other simply supported*

Hull side pressure Ps	53.9kg/cm2
Tube material density ρtube	7.86E-03kg/cm3
Density of the tube inside the fluid ρin	7.62E-07kg/cm3
Density of the tube without fluid ρin	7.80E-04kg/cm3
Tube weight per unit length We	0.1754kg/cm
Interior fluid weight per unit length Wt	0.001kg/cm
Hydrodynamic mass Hm	0.1225kg/cm
Effective pipe weight per unit length W	0.298kg/cm
Tube moment of inertia I	325kg cm^2
Young's modulus of tubes at design temperature	1.8E+06 kg/cm^2

Gravitational constant g	981 cm/sec2
Number of Strouhal St	0.8
Xl Pl / D0	0.65
Xt Pt / D0	2.26
Dynamic viscosity µ	0.01 centipoise
Reynolds number Re	8.92E+06
Fluid velocity outside the tube (at full load) V	100cm/s
Shell-side specific heat ratio ϒ	1,603
Distance between reflective wall B	304.8cm
Logarithmic decrement of δr tubes	0.02
Fluid elastic parameter of operating condition X	0.0608
Natural frequency of vacuum tube fn1	36.993Hz
Natural frequency of the tube in operating condition fn2	36.927Hz
Acoustic frequency fa	25567.18Hz
Critical flow velocity in operating condition Vcr	734.412 cm/sec

6. Results and discussions

After integrating the existing core and shell heat exchanger data in the oil refinery shown in the previous tables; mechanical characteristics, and those of fluids as well as the dimensions of the tubes and the thermal and hydraulic parameters of the flow, GOOD VIBRATION provides the profile of the amplitudes of vibration from the natural frequency of the tube to the flow illustrated in the Fig. 8.

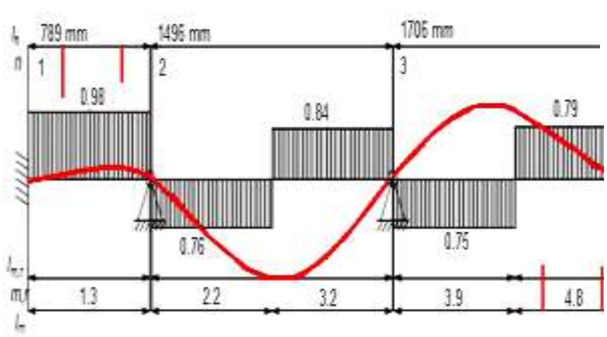

Figure 8. *Tube natural frequency amplitude* ***GV***

In Fig. 8, the amplitudes are distributed among three sections of tubes, each 399.1cm long and spaced out at 789mm, 1496mm, and 1706mm. The resulting wavelengths are 1.3, 5.4, and 8.7.

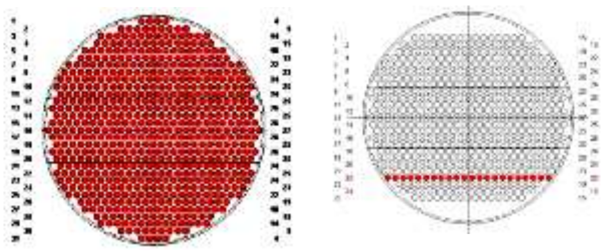

Figure 9. *Tube row diagram GV*

In Fig. 9, the first row of the tube simulation is shown. It has a length of 399.1 cm and is represented in the profile figure. For the values of vortex excitation frequencies imposed for the simulation, according to the results of the program; it is noted that the shape of the vibrations is wavy, starting with excitation frequency values lower than the natural frequency of the tube in the operating condition. The latter is not too far from that in no-load; respectively 36.927 Hz and 36.993 Hz.

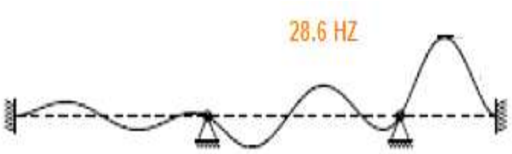

Figure 10. *28.6Hz frequency vibration amplitude GV*

The amplitudes at 28.6 Hz and 32.6 Hz frequencies are low. When compared to the frequency of the tube

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in use, which is 36.993 Hz, the amplitude profile at 28.6 Hz is non-harmonic. Additionally, the spacing between the amplitudes is too small and incoherent, which results in low vibration.

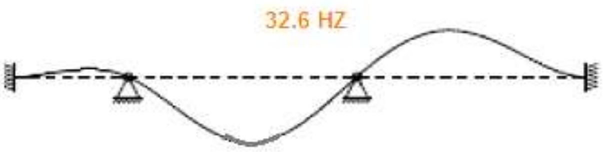

Figure 11. *32.2Hz frequency vibration amplitude* GV

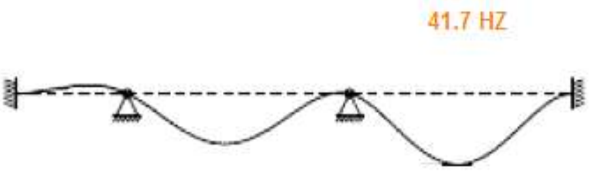

Figure 12. *41.7Hz frequency vibration amplitude GV*

When the excitation frequency is close to the natural tube frequency for the frequency of 32.6 Hz (fig. 11) and that of 41.7 Hz (fig. 12), the amplitude profile is almost similar to that developed by the natural tube frequency (fig. 7).

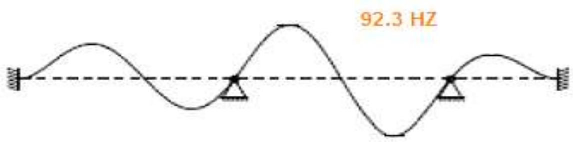

Figure 13. *92.3Hz frequency vibration amplitude GV*

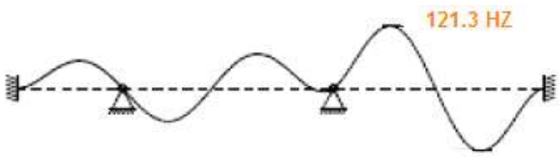

Figure 14. *121.3Hz frequency vibration amplitude* GV

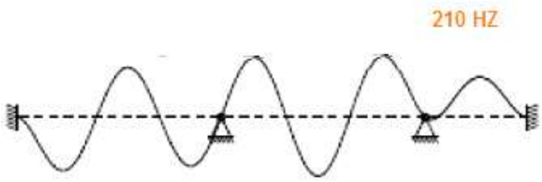

Figure 15. *210Hz frequency vibration amplitude GV*

Fig. 10 shows that when the excitation frequency is 28.6 Hz, which is lower than the natural frequency of the tube, the amplitude is lower at the entrance and wider at the tube exit.

Regarding the amplitude of excitation frequencies, they exceed the proper tube frequency. The values are 92.3, 121.3, and 210 Hz, as illustrated in figures 13, 14, and 15. One can notice that they are more and more harmonic as the frequency increases, with a good distribution along the tube, which can be positive.

The beam is exposed to strong vibrations due to fluid-elastic instability, with a level of danger estimated at 117.3% at the frequency 210 Hz, according to the result of the program.

Conclusion

The simulation showed that the excitation frequency close to the natural tube frequency remains without negative effect, whereas when they are lower than the natural tube frequency, they can generate weak vibrations. These weak amplitudes cause cuttings of the bundles by the baffles. For frequencies greater than or double the natural frequency of the tube, a good distribution of amplitudes and harmonic profile is observed. These vibrations amplitudes could be beneficial and positive for the improvement of the exchanger's efficiency by participating in a good creation of the turbulence of the flow, which directly influences the exchange coefficient and the Reynolds number. Moreover, it can avoid the risk of fouling problems, more prevalent in core and shell heat exchangers in the petroleum industry.

Therefore ensure a long life of the exchanger, except that it must not exceed an optimal frequency value which can be critical with a risk of danger for our equipment, this risk which is determined by Good vibration program which was evaluated at 117.6% for frequencies exceeding 120 Hz in the case studied.

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