ACOUSTIC RESONANCE EXCITATION IN A HEAT EXCHANGER: A CASE STUDY

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Abstract. *Flow-induced vibrations of heat exchanger tube bundles often cause serious damages resulting in lost efficiency and high repair costs. The objective of this paper is to describe the mechanisms that cause flow-induced vibrations in heat exchanger tube bundles with cross-flows. These phenomena are investigated by means of acceleration and sound pressure measurements, as well as with the aid of extensive acoustic models visualization of cavity and vibrations models of tubes and plates . The tube layout pattern, the spacing ratio, Strouhal number and Reynolds number determine the dominant instability mechanism. The excitation mechanisms are generally classified as tube ressonance by vorticity shedding, acoustic resonance, turbulent buffeting and fluid-elastic instability. Typically, this problem is solved by inserting plates inside the tubule banks, in order to inhibit the acoustical instabilities by modifying the acoustic field. A practical case is analysed and the main mechanism of excitation was detected. Measurements indicated the efficiency of the proposed solution.*

Keywords*: Flow-induced vibration, vorticity shedding, acoustic resonance, Strouhal number, heat exchange cross-flow.*

1. INTRODUCTION

The increased demand for sugar and alcohol from sugar cane, either for internal consumption and for export, has greatly increased the orders for new mills. The excess of energy generated is transferred to the matricial electrical energy system. The need for higher efficiency has led manufactures to improve the design of heat exchangers, operating with higher velocities and different tubes arrangements compared to previous designs.

Flow-induced vibrations and acoustic resonances in heat exchangers have caused serious damages in the system integrity resulting in high repair costs. Reviews of fluid-induced vibrations have been published by Païdoussis (1982). Gorman (1976) identified four principal sources of cross-flow excitation in tube banks, classified as (a) vortex shedding (b) acoustic resonance (c) turbulent buffeting and (d) fluid-elastic instability. This paper describes the identification of a vortex-induced acoustic resonance problem in a heat exchanger and the solution proposed.

2. EXCITATION MECHANISM

2.1. Vortex shedding

Periodical pressure fluctuations limited to a narrow frequency range are known as vortex shedding. This phenomenon may excite vibration in liquid flow or acoustic resonance in gas flow. When the frequency of vortex shedding coincides with some acoustic resonance of the ducts, resonant vibration and rapid tube damage may occur, especially in liquid flows. The velocity range over which the tubes exhibit large amplitude vibration is referred to as the "Lock-in" range, and is show in Fig. 1. Although this excitation has been recognized since 1950 only recently studies have shows clearly that it results from vortex shedding from tubes (Weaver, 1993; Ziada et al., 1989a & 1992)

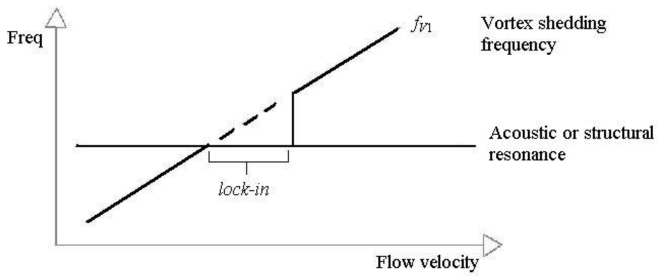

Figure 1. Lock-in condition when the frequency of excitation matches with an acoustic resonance

The vortex shedding occurs in the beginning of the section tubes, mainly in heat exchangers with little space between tubes. The frequency of vortex shedding is represented by f_{V1} and is function of the Strouhal number, ST_V defined as:

$$ST_V = \frac{f_{V1}.D}{U} \qquad (1)$$

where D is the external diameter of the tubes; U , is the air velocity in the heat exchanger.

2.2. Acoustic resonance

Acoustic resonance may take place in heat exchanger tube bundles when vortex or periodic wake shedding frequencies coincide with a natural frequency of the acoustic standing waves within a heat exchanger. Such resonances normally cause intense acoustic noise and often serious tube and baffle damage. Acoustic resonance is possible in gas heat exchangers with both finned and unfinned tubes.

Acoustic resonance requires two conditions (Pettigrew, 2003): (i) coincidence of vortex shedding and acoustic frequency, and (ii) sufficiently high acoustic energy or sufficiently low acoustic damping to allow sustained acoustic standing wave resonance. In heat exchanger tube bundles, acoustic standing waves are generally normal to both the tube axes and the flow direction.

2.3. Turbulent buffeting

Flow turbulence is a significant excitation mechanism in heat exchanger with cross flow. This mechanism is characterized by irregular fluctuations in space and time. Interior tubes, well within a heat exchanger tube bundle, are excited by turbulence generated within the bundle. This excitations is governed by tube bundle geometry. On the other hand, inlet tubes are excited by turbulence generated by upstream components such as inlet nozzles, entrance ports and upstream piping elements.

The dominant frequency of excitation can be determined by Owen's equation:

$$ST_{tb} = \frac{f_{tb}D}{U} S_t S_l = 3.05\left(1 - \frac{D}{T}\right)^2 + 0.28 \qquad (2)$$

where U is the flow velocity between tubes; D , the external diameter of tube; L , space between tubes in the column direction; T , space between tubes in the row direction; $S_t = \frac{T}{D}$ and $S_l = \frac{L}{D}$.

For the geometry analysed in this paper, $ST_{tb} = 1.14$. Turbulent buffeting is more critical to high capacities of excitation.

2.4. Fluid elastic instability

Fluid elastic instability is based on self-excited forces, which are caused by the interaction between tube motion and fluid flow. At the onset of fluid elastic instability, fluid forces occur which are proportional to tube displacement and influence the stiffness of the system. This so-called soft or stiffness-controlled excitation leads to a coordinated motion of all tubes and is dominant in low-density fluids and staggered tube configurations. It is superimposed by a second kind of fluid forces, proportional to tube vibration velocities, which reduce the damping of the system. This mechanism leads to a sudden rise of the amplitudes and is known as "galloping" (Gelbe, 1995). Figure 2 shows a typical amplitude response and galloping.

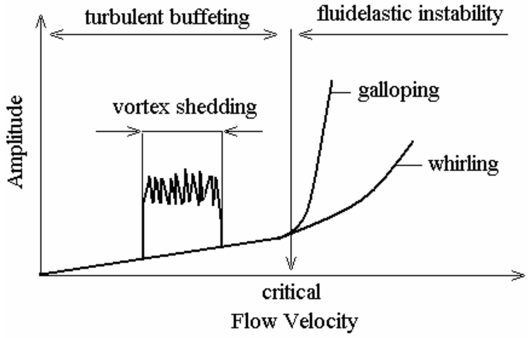

Figure 2. Typical amplitude response to vortex shedding excitation.

The stability equation for dimensionless critical gap velocity is obtained by (Schoröder, 1999):

$$\frac{u_{crit}}{f_n D} = K\sqrt{\frac{m\delta}{\rho D^2}} \tag{3}$$

where f_n is resonance frequency of tube; D, the external diameter of tube; K, a constant; δ, the damping; ρ, the density and m, mass per unit length. The relation between velocity and resonance frequency of tube is $u_{crit} = 1.54 f_n$. The velocity necessary to produce instability is very high (above 20 m/s).

3. MEASUREMENT RESULTS

The heat exchanger under analysis is a cross flow type with tube geometry distribution as show in Fig. 3. A 350 HP fan blower forced the air flow trough the exchanger. In order to identify the problem of excessive noise and vibration, the fan rotational speed was varied from 450 RPM to 850 RPM, which correspond to upstream air velocities of 2.90 m/s to 5.50 m/s, respectively.

The dimensions of the heat exchanger are 3.80 m in height, 3.50 m wide and 6.70 m in length. The tubes have 63 mm in external diameter, 3.80 m length and the wall is 4 mm thick. Tubes are distributed over 26 rows by 54 rows, and arranged as shown in Fig. 3.

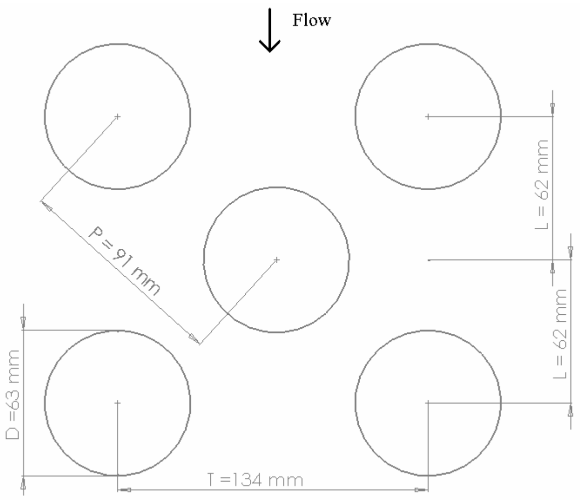

Figure 3. Cross-section of experimental test section.

For the configuration of the heat exchanger analysed in this paper, the ratio $X_D = \frac{P}{D} = 1.45$ (pitch ratio), then $ST_v = 1.0$ (Ziada, 2006).

Duct vibration and external sound pressure spectra were measured for several fan speed and results are shown in Fig.4 and 5. It is noticeable a dominant peak at 53 Hz, and its correspondent harmonics. The vortex shedding frequency varies linearly with air velocity as illustrated in Fig. 4, and also shown by Eq. 1.

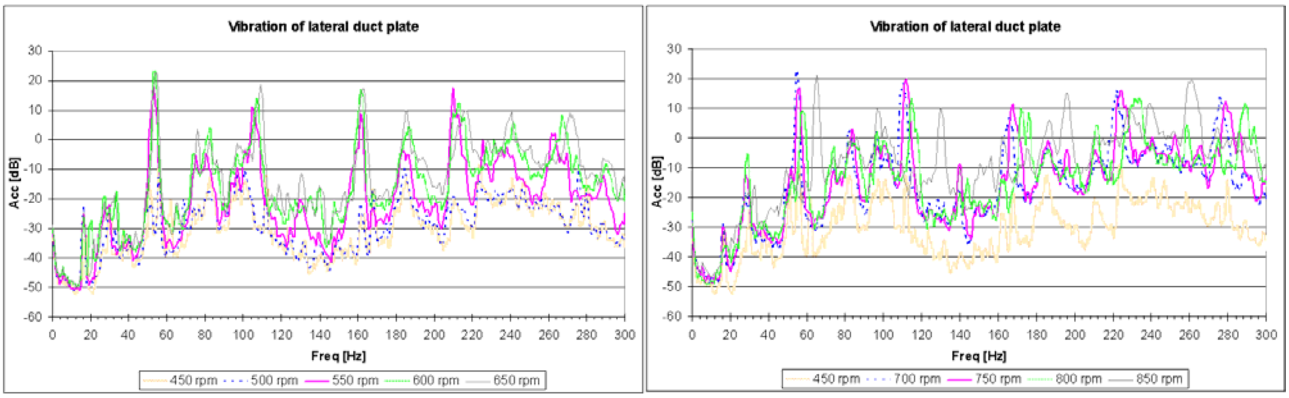

Figure 4. Acceleration of lateral plate.

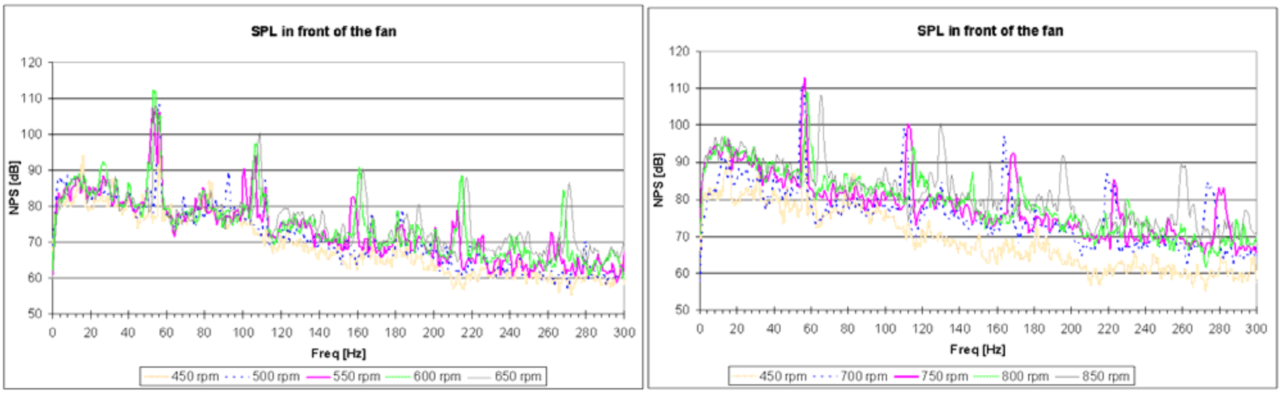

Figure 5. Sound pressure at fan inlet.

The resonances of several heat exchangers components were analysed in order to identify the source of the excessive vibration and noise generated. Tubes vibrations, lateral duct plate vibrations and acoustic modes were analyzer by Finite Elements seeking the identification of the resonances.

4. NUMERICAL SIMULATIONS

4.1. Tube vibrations

A small group of tubes was modeled by Finite Elements using shell elements, coupled to 20 mm thick plates at the tubes ends. The objective was to obtain frequency ranges for the first resonant modes. The simulation of the complete set of tubes would obviously be an expensive task. Figure 6 shows a typical frequency response of the tube excited by a point force at one of them. Figure 7 shows typical response for the first three modes. Table 1 indicates the frequency ranges for the three first modes. It is clear that the peak response at 53 Hz during operation is not related to tubes resonances.

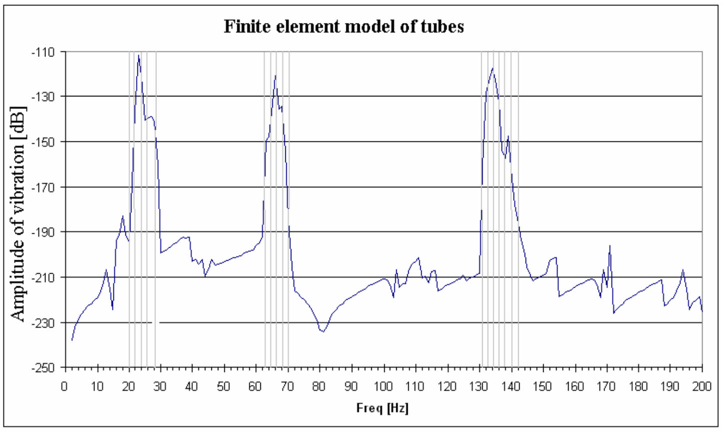

Figure 6. Frequency response of tubes.

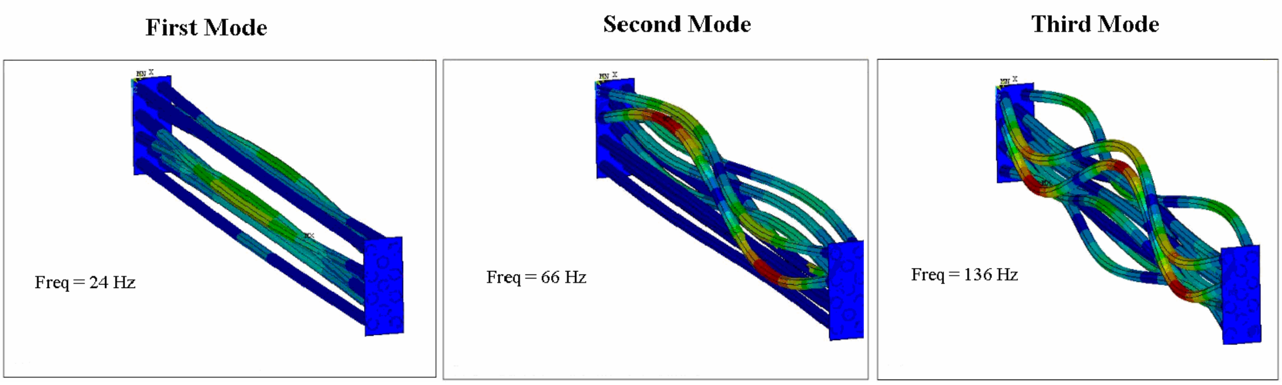

Figure 7. Modes of tubes.

Table 1. Vibration Modes of tubes.

Modes	Range of resonances
1	21 a 27 Hz
2	62 a 68 Hz
3	130 a 138 Hz

4.2. Acoustic modes

This simulation considered air with a variable temperature distribution over the length of the exchanger, from 30°C at blower exit up to 225°C at the end of the exchanger. The speed of the sound in longitudinal and transversal direction was considered 95% of that in air due to the "porosity" caused by the tubes. Modes in the transverse direction only were analysed since these can be excited by vortex shedding formation. A typical frequency response is shown in Fig. 8. The first three resonances in the transverse direction are 51 Hz, 55 Hz and 59 Hz, as show in Fig. 9. This indicates the presence of acoustic resonances in the range of 53 H, which can easily be excited, and being responsible for the vibration and noise problem.

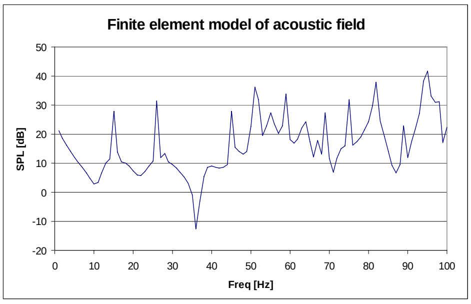

Figure 8. Frequency response of acoustic field.

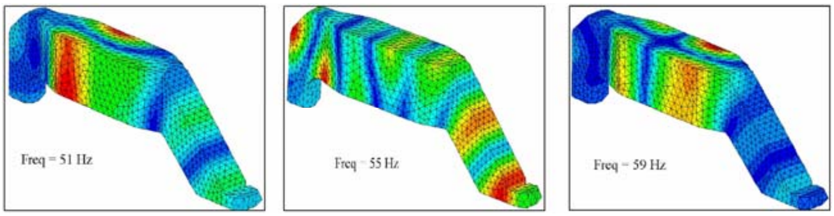

Figure 9. Some heat exchanger acoustic modes.

4.3. Lateral duct plate vibration

High vibration levels at the lateral duct plates were observed during operation. Severe fatigue problems required permanent welding repairements due to strong excitation levels.

A Finite Element model of a detailed and full length lateral duct plate was developed using shell and beams elements for the reinforcements. A typical vibration response is show in Fig. 10. It can be observed the presence of a global mode resonance at 51 Hz, which is very close to the acoustic mode and the high response frequency (53 Hz) observed during the measurements.

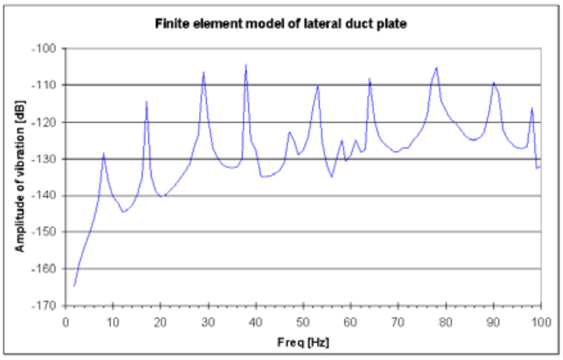

Figure 10. Frequency response of lateral duct plate.

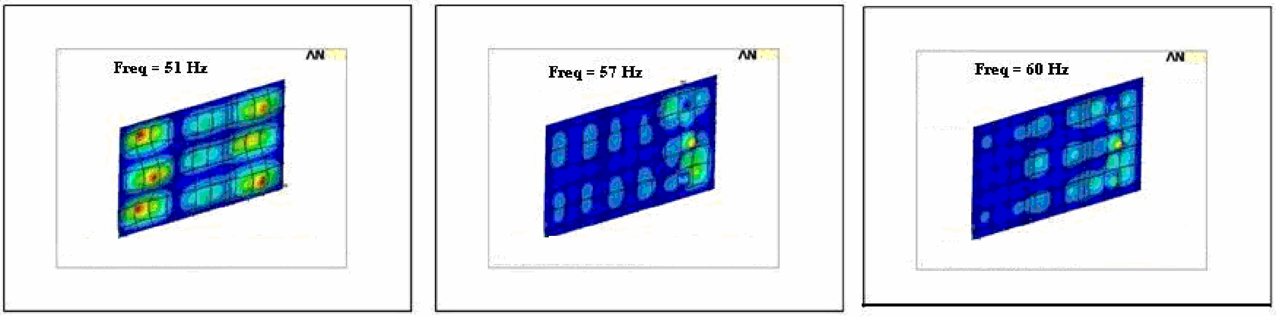

Figure 11. Vibration modes of lateral duct plate.

Three simulated plate vibration mode having frequencies close to 53 Hz are show in Fig. 11.

5. VIBRATION AND NOISE PROBLEM IDENTIFICATION

Figure 12 shows a diagram of system resonance and the vortex-shedding frequency variations with air velocity. In the velocity values axis are indicated the correspondent fan rotation, in RPM.

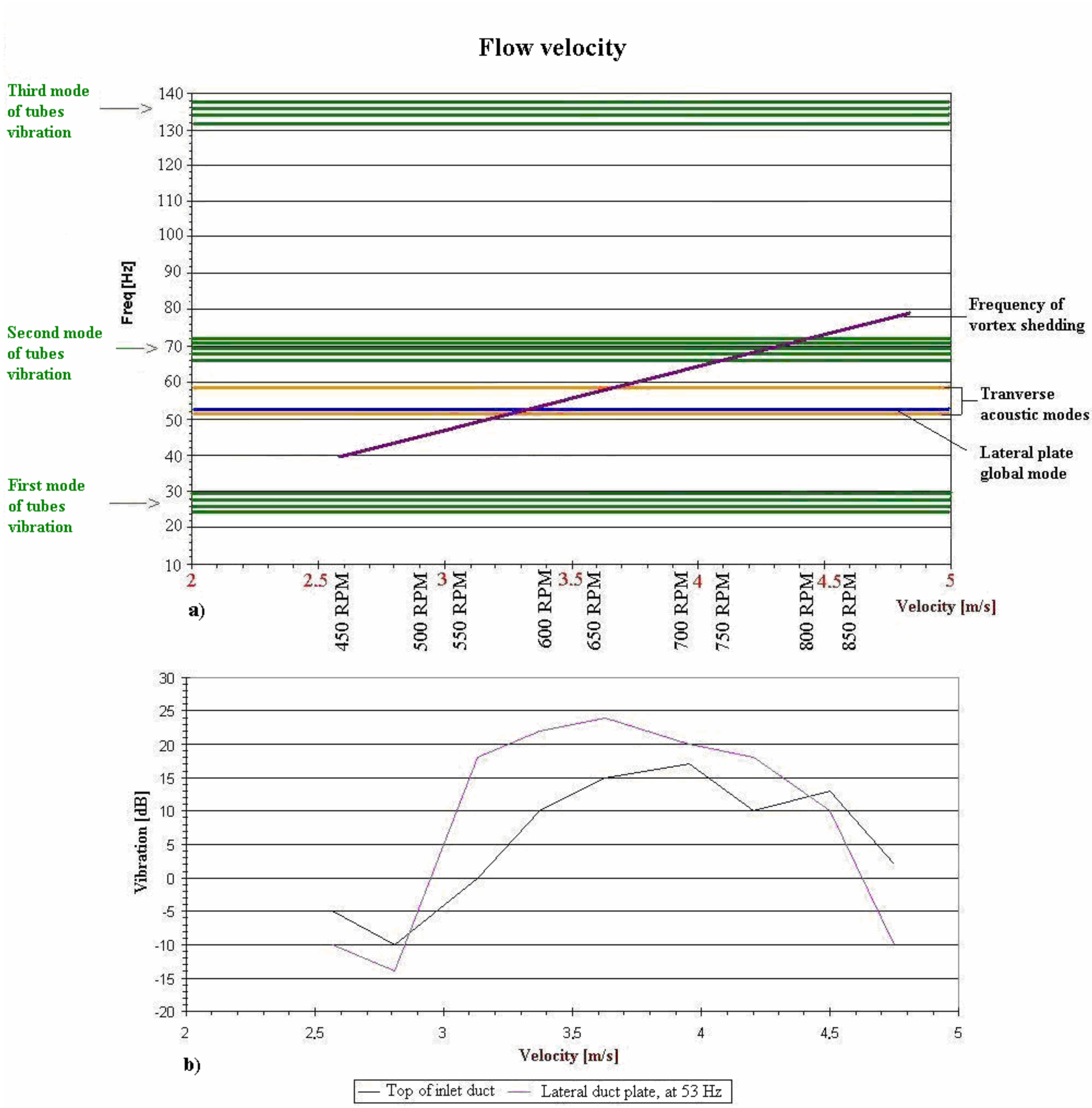

Figure 12. Frequency of vortex shedding (a) and responde at 53 Hz, as function of flow velocity (b).

The lower part of this figure shows the vibration and sound pressure response measured at 53 Hz for the various fan rotation values. It is clear that large response initiates at air velocity value for which the transverse acoustic mode and with a lateral duct plate global mode. Tubes vibrations are not taking part of the excitation mechanism. Another evidence of the vortex shedding acoustic mode excitation can be seen from the noise spectra measured at 1 m from the fan inlet, show in Fig. 5. The clear peak response at 53 Hz shows the strong acoustic mode excited in the duct.

6. THE PROPOSED SOLUTION

In order to avoid excitation of the acoustic modes in the transverse direction, the exchanger width has to be shortened. For this reason, two longitudinal rows of tubes were removed and in the empty space, 4 mm thick plates were placed, and welded at the top and bottom borders. Figure 13 shows the location of the plates. This procedure did not interfered significantly in the exchanger efficiency.

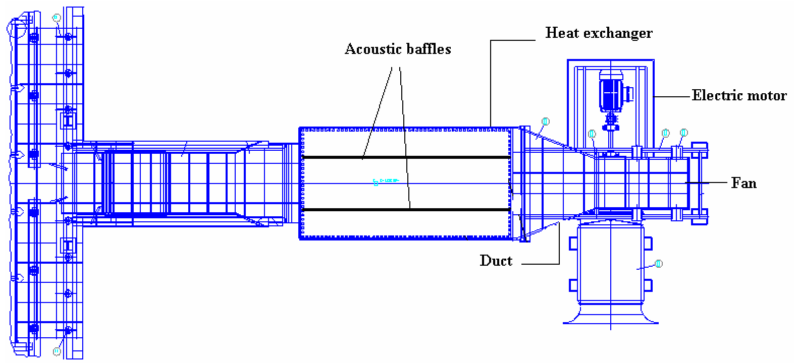

Figure 13. Acoustic baffles in the heat exchanger.

News measurements of vibrations and sound pressure levels were taken, and shown in Fig. 14 and 15. Both vibration and sound pressure levels are greatly reduced. Over all noise levels at the plant are now considered perfectly acceptable, as expected for this kind of industry. The sound pressure level shown in Fig. 15, measured at 1 m from the fan inlet, indicates that the acoustic resonance was perfectly avoided by this procedure.

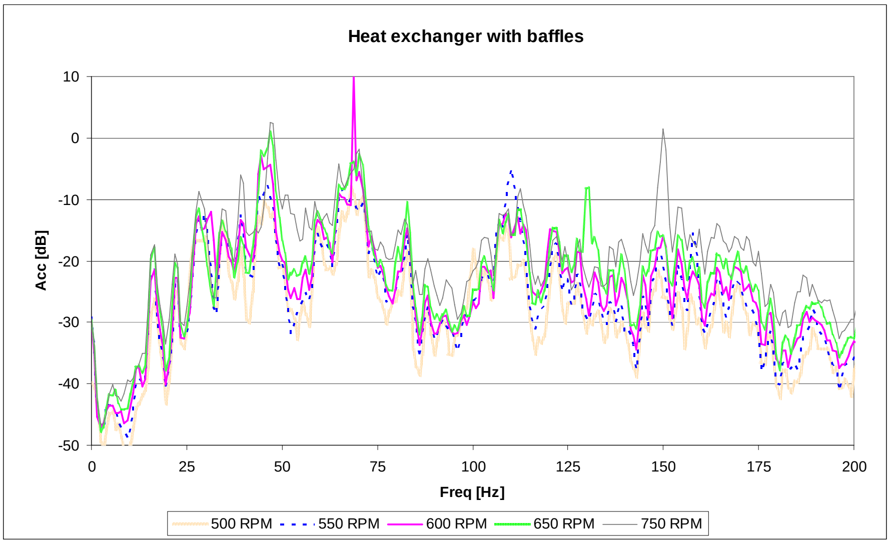

Figure 14. Vibration of lateral plate with acoustic baffles in the heat exchanger.

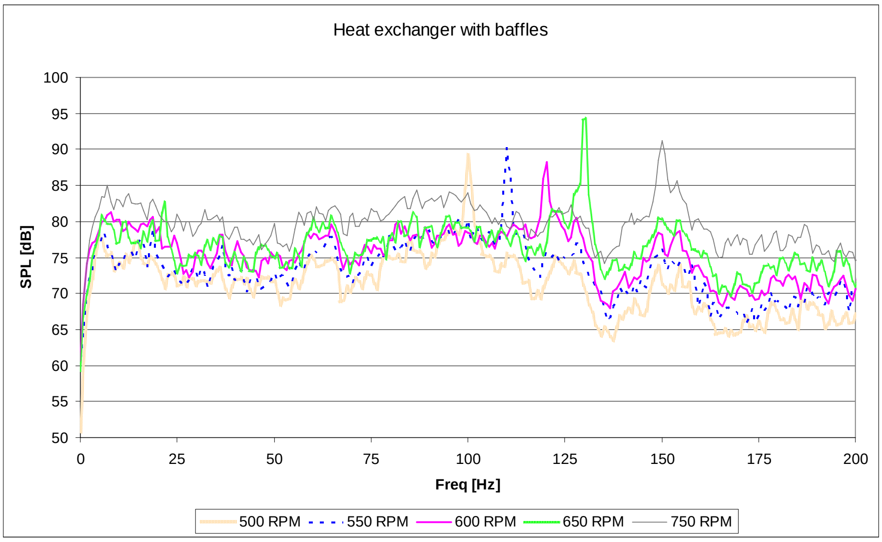

Figure 15. SPL at inlet of the fan with acoustic baffles in the heat exchanger.

7. CONCLUSIONS

Vortex shedding was identified as the mechanism of transverse acoustic excitation in a heat exchanger, when the frequency of vortex formation coincided with an acoustic resonance. This also coincided with a global vibration mode lateral plate resonance, aggravating the vibration and noise problem. Tubes vibrations were not considered as part of one main excitation mechanism. The use of two longitudinal plates to avoid transverse acoustic resonant frequencies in the range of the exchanger operation has shown to be an efficient solution. Noise levels measured after this modification indicated that the problem was totally eliminated.

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