Vibration Analysis of Heat Exchangers of a Nitric Acid Plant

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This paper deals with an operational problem of two heat exchangers of the nitric acid synthesis plant. Detailed calculation and vibration analysis were made and vibration problem of the nitrous gases cooler E-111 was identified. When the vibration problem was indicated, several steps were taken for its removal. For five different types of baffles a study was made to determine how their number and spacing influence the cross-flow rate of the fluid. Results showed that the vibration problem cannot be removed completely, but it can be reduced considerably with different types of baffles. Also, the analysis showed that the cooling water condenser E-114 does not have a vibration problem.

Key words:
Heat exchangers, vibration analysis, baffles, tube failure

Introduction

In this work, a detailed calculation and vibration analysis were performed for two *shell-and-tube* heat exchangers of the nitric acid plant (Fig. 1). Heat exchangers are used for cooling nitric gases before their entry into the absorption column. It is well known that lower temperature and higher pressure enable better absorption of nitric gases. Thus, the absorption efficiency depends on the amount of heat transferred through the two heat exchangers.[1]

In the first heat exchanger (E-111), compressed nitric gases from the top of the whitening column are cooled with purge gas from the top of the absorption column. In the second heat exchanger (E-114) nitric gases are cooled with cooling water (Fig. 2).

Practical experience has indicated that heat exchangers have a major operational and noise problem; therefore, a detailed vibration analysis was made. When a vibration problem of one heat exchanger was identified, an additional study was made for five different types of baffles. Simulation was performed using chemical process simulation software ChemCAD i.e. its integrated module CC-THERM, an interactive simulation tool for design and rating of the *shell-and-tube* heat exchangers.[2]

The results obtained in this work can be helpful for engineers to design heat exchangers of the nitric acid synthesis because the current design criteria are based on data collected with water and air, while the prediction of whether or not a given heat exchanger configuration will resist vibration and tube failure is not yet well defined.

Case study

Both analyzed heat exchangers are *shell-and-tube* type designed by TEMA standards[3] with single segmental baffles inside the shell. Based on their construction, they are classified as fixed-tubesheet heat exchangers with bonnet-type channel covers (TEMA class C/BEM). They have straight tubes that are secured at both ends to tubesheets welded to the shell (Fig. 3).

The most common materials of construction for TEMA heat exchangers are carbon and stainless steel and their properties and composition are specified by ASTM standards.[4] In most cases, all components are made of identical materials. Some heat exchangers are constructed from dissimilar metals and in that case extreme care in their selection is required since electrolytic attack may develop.[5] Table 1 shows basic data about these two heat exchangers.

Tube bundle and baffles

A tube bundle is the most important part of a tubular heat exchanger. Tubes are generally the most expensive part of the exchanger and they are most prone to corrosion. Tube sheets, baffles or support plates, tie rods and, usually, spacers complete the bundle.[5]

Baffles serve two important functions:

– Support the tubes during assembly and operation, helping to prevent vibration from flow-induced eddies.

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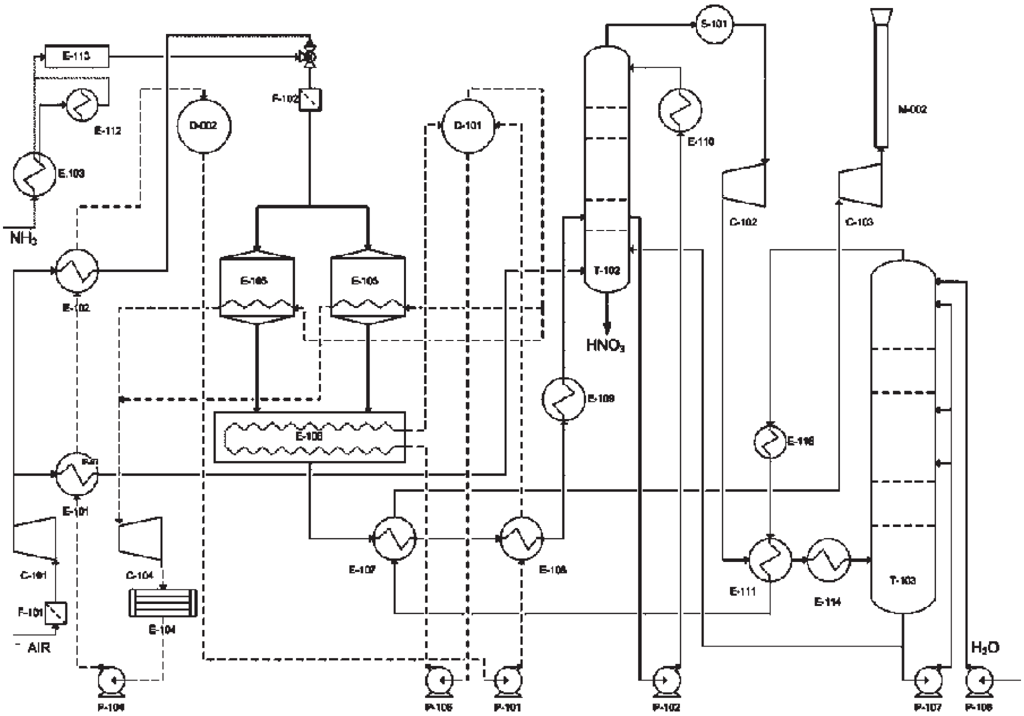
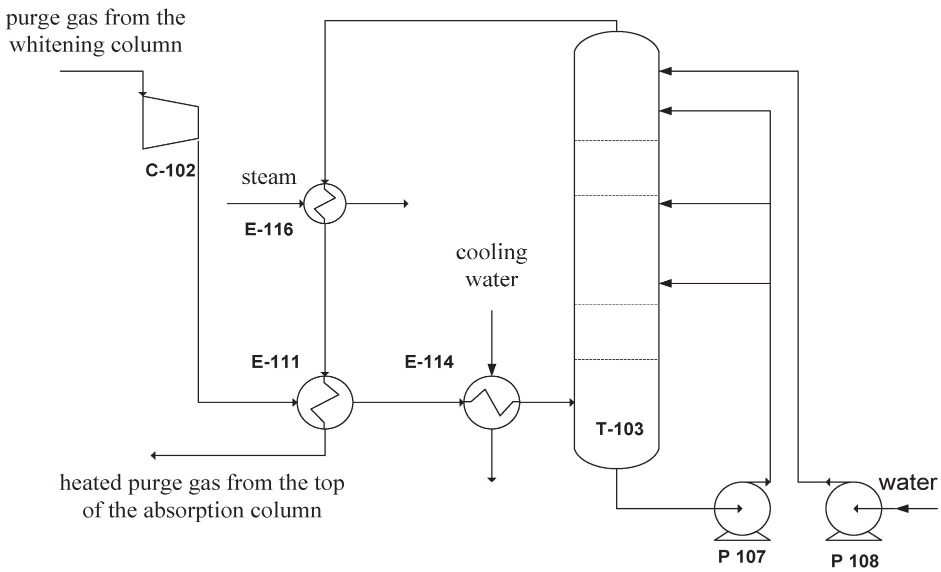
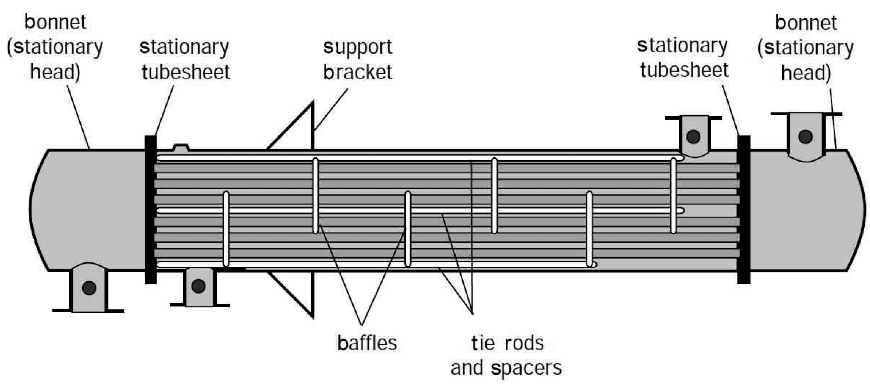


F i g . 1 – *PFD of the nitric acid plant*



F i g . 2 – *Observed section of the process*



F i g . 3 – *Fixed-tubesheet heat exchanger*[6]

Table 1 – *Elementary data about exchangers, E-111 and E-114*

	Nitrous gases cooler, E-111	Cooling water condenser, E-114
Φ/kW	2266.3	813.9
Tube side		
Fluid	Compressed nitrous gases	Compressed nitrous gases
q_m/kg h^{-1}	70 753.0	81 952.0
θ_i/°C	43.0	80.0
θ_o/°C	153.0	50.0
p_i/MPa	0.795	0.800
construction material	stainless steel; ASTM A 249 TP 304L	stainless steel; ASTM A 249 TP 304L
Shell side		
Fluid	Purge gas from T-103	Cooling water
q_m/kg h^{-1}	81 952.0	100 000.0
θ_i/°C	176.0	28.0
θ_o/°C	80.0	35.0
p_i/MPa	0.800	0.450
construction material	stainless steel; ASTM A 240 304L	carbon steel; ASTM A 285 C

– Direct the shell-side fluid back and forth across the tube bundle to provide effective velocity and heat transfer rates.

There are several types of baffles,[6] as shown in Fig. 4. Single segmental and double segmental baffles reduce cross-flow rate for a given baffle spacing. The triple segmental reduce both cross-flow and long-flow rates and are identified as the 'window-cut' baffles. In some special cases *no-tubes-in-window*, *disc-and-donut* and rod baffles are used. Rod or bar baffles have either rods or bars extending through the lanes between the rows of tubes. The shell-side flow is uniform and parallel to the tubes. Stagnant areas do not exist.[7]

Segmental baffles do not extend edge to edge but have a cut that allows shell-side fluid to flow to the next baffle chamber. For most liquid applications, the cuts areas represent 20 to 25 % of the shell diameter. For gases, where a lower pressure drop is desirable, baffle cuts of 40 to 45 % are common. Two type of baffle cut orientation are usually used (Fig. 5). Horizontal baffle cut is recommended for single-phase fluid on the shell side, because this minimizes accumulation of deposit at the bottom of the shell and also prevents stratification. In the case of a two-pass shell, a vertical cut is preferred for ease of fabrication and bundle assembly.[6]

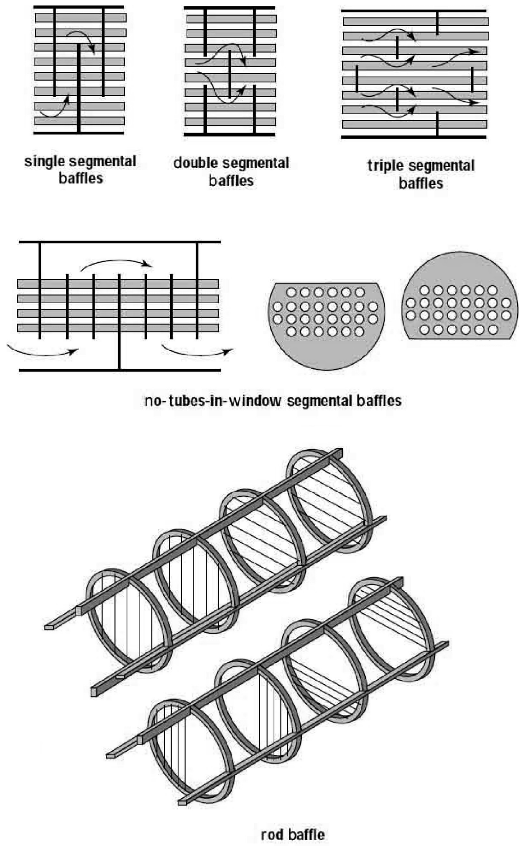


Fig. 4 – *Types of baffles*[6]

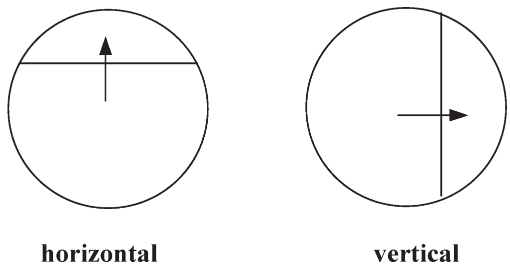


Fig. 5 – *Baffle cut orientation*[5]

Baffles must overlap at least one tube row in order to provide adequate tube support. They are spaced somewhat evenly throughout the tube bundle to provide even fluid velocity and pressure drop in each baffled tube section.

Noise and vibration problems in tube bundles

Tube bundles in heat exchangers are often subject to vibration and noise problems. Vibration can lead to wear and consequential tube failures.[8] Damage is more likely to occur with gases or vapours on the shell-side than with liquids. Flow-induced vibrations also occur with liquids, but the damage is often limited to localized areas of relatively high rate.

The tubes vibrate only at unique responding frequencies called their natural frequencies. The natural frequency of the tubes depends primarily on their geometry and material of construction.

Whenever gas flows over a tube bundle, in the in-line or staggered arrangement the vortices are formed and shed beyond the wake of the tubes resulting in harmonically varying force perpendicular to the flow direction. This vibration frequency is called the vortex shedding frequency. If vortex shedding frequency coincides with natural vibration frequency of the tubes, the resonance occurs which leads to bundle vibration.[9] Vortex shedding can be described by the dimensionless Strouhal number:

$$S = \frac{f_{vs} D}{v_{cross}} \tag{1}$$

Strouhal number data were reviewed for tube bundles of various configurations and tube pitch ratios ($X_p = P/D$)[10–12]. Vortex shedding occurs in the ranges $100 < \mathrm{Re} < 10^5$ and $\mathrm{Re} > 2 \cdot 10^6$ and dies out in between.

Another phenomenon that occurs with vortex shedding is the acoustic vibration, leading to noise and high pressure drops. Standing waves are formed inside the duct. The duct or the bundle enclosure vibrates when the vortex shedding frequency coincides with acoustic frequency. Such resonance normally causes intense acoustic noise and often serious tube and baffle damage. The acoustic frequency can be predicted by the following equation:

$$f_a = \frac{n U_S}{2d} \tag{2}$$

The lowest acoustic frequency is achieved when $n = 1$ and the characteristic length is the shell diameter. This is called the fundamental tone and higher overtones vibrate at acoustic frequencies 2, 3, or 4 times the fundamental (n = 2, 3, or 4) but reports of higher overtones in heat exchangers are rare. The velocity of the sound in a gas is given by the equation below:

$$U_S = \left(\frac{z \gamma R \theta}{M}\right)^{1/2} \tag{3}$$

The acoustic frequency always has the same value at the inlet, centre, and outlet of the exchanger.[13]

Turbulent buffeting is the name given to the fluctuating forces acting on tubes due to extremely turbulent flow of the shell-side fluid. The turbulence has a wide spectrum of frequencies distributed around a central dominant frequency which increases as the cross-flow velocity increases. This turbulence buffets the tubes which selectively extract energy from the turbulence at their natural frequencies from the spectrum of frequencies present. This is extremely complex form of excitation.[13] The empirical equation of Owen[14] was used to predict this frequency:

$$f_{tb} = \frac{v_{cross} D}{P_l P_t}\left[3.05\left(1 - \frac{D}{P_t}\right)^2 + 0.28\right] \tag{4}$$

This equation was developed for gases and may not be applicable to liquids.

Vortex shedding resonance and random excitation are not usually of concern in gas flow since the fluid density is generally low thereby resulting in relatively small excitation forces. However, both mechanisms should be considered in some gas heat exchangers. Acoustic resonance is possible in gas heat exchangers and it must be avoided. The experience with similarly sized units, tube spacing and tube size can be very helpful for predicting probability of vibration problems.

Results and discussion

Vibration analysis

A detailed calculation and vibration analysis was made for both heat exchangers at three different locations:

1. The entrance baffle span (between the tube sheet and the first baffle)

2. The centre baffle span (at a typical baffle centre span location)

3. The exit baffle span (between the last baffle and the rear tube sheet)

Table 2 shows geometry of the exchangers E-111 and E-114.

Detailed vibration analysis includes the following steps. The first step is determination of fluidelastic instability. Fluidelastic instability is by far the most important mechanism and must be avoided in all cases. It can be determined by Connors method:[15]

$$v_{crit} = \beta f_n \sqrt{\frac{2\pi \zeta_n m_t}{\rho}} \tag{5}$$

This topic was reviewed[16] and formulated in terms of dimensionless flow velocity, $v_{crit}/f_n D$ and dimensionless mass-damping parameter, $2\pi\zeta_n m_t/\rho D^2$:

$$\frac{v_{crit}}{f_n D} = \beta\left(\frac{2\pi \zeta_n m_t}{\rho D^2}\right) \tag{6}$$

Table 2 – *Geometry of heat exchangers E-111 and E-114*

	E −111	E −114
TEMA class	C/BEM	C/BEM
shell diameter, m	0.934	0.934
number of tubes	707	693
tube length, m	11.0	3.6
tube outer diameter, mm	25.0	25.0
tube pattern	triangular (30)	triangular (30)
tube pitch, mm	32.0	32.0
number of tube passes	1	1
baffle type	SSEG	SSEG
baffle cut percent, %	40	25
direction of baffle cut	horizontal	horizontal
inlet spacing, m	2.06	0.26
center spacing, m	1.70	0.27
outlet spacing, m	2.06	0.26
number of baffles	5	12

The damping ratio, ζ_n, is the total damping ratio in heat exchangers with gas on the shell-side as defined as friction between tubes and tube-supports, in percent:[17]

$$\zeta_n = 5\left(\frac{N-1}{N}\right)\left(\frac{L}{l_m}\right)^{1/2} \tag{7}$$

When the shell-side fluid is a gas, the Connors method is the most important determinant of the vibration problem. If the velocity in the given span exceeds the Connors' critical velocity, then four different criteria, based on the Chen[9] for vortex shedding frequency and Owen[14] method for turbulent buffeting frequency, have to be considered. So, the second step of vibration analysis is to check that the tube vibration level is below the permitted level and that unacceptable resonance is avoided,[18] based on the following four criteria:

1. The ratio of vortex shedding frequency to natural frequency is greater than 0.5.

2. The ratio of vortex shedding frequency to acoustic frequency is greater than 0.8 and less than 1.2.

3. The ratio of turbulence buffeting frequency to natural frequency is greater than 0.5.

4. The ratio of turbulence buffeting frequency to acoustic frequency is greater than 0.8 an less than 1.2.

The results of the vibration analysis for the heat exchanger E-111 are shown in Table 3. It may be concluded that vibration problems exist in all three considered locations of the E-111 heat exchanger. Cross-flow rate is more than 10 times greater than the critical value. The ratio of natural frequency to vortex shedding frequency is 40.70, indicating that resonant conditions cannot occur. However, the ratio of vortex shedding frequency to acoustic frequency is 1.02 and vibration of the tube bundle may occur, causing possible damage and noise.

Table 3 – *Vibration analysis results for heat exchanger E-111*

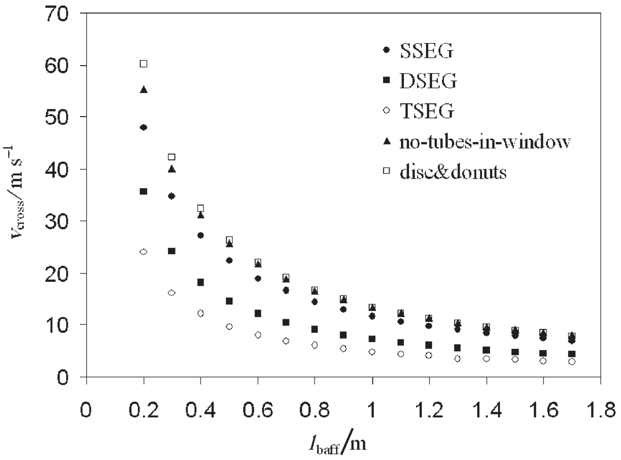
	Inlet	Center	Outlet
tube span, m	3.76	3.40	3.76
v_{cross}, m s^{-1}	5.73	6.95	5.73
v_{crit}, m s^{-1}	0.69	0.65	0.61
v_{cross}/v_{crit}	8.35	10.69	9.37
f_n, s^{-1}	4.75	4.75	4.75
f_a, s^{-1}	223.3	212.0	200.0
f_{vs}, s^{-1}	179.0	217.1	179.0
f_{tb}, s^{-1}	33.5	40.7	33.5
f_{vs}/f_n	37.70	40.70	37.70
f_{vs}/f_a	0.80	1.02	0.90
f_{tb}/f_n	7.05	8.56	7.06
f_{tb}/f_a	0.15	0.19	0.17
vibration exists	YES	YES	YES

One of the possibilities for vibration problem removal is to use different types of baffles. A baffle shape, cut, number, orientation, and spacing directly determine the fluid rate, which in turn has a major influence on vibration occurrence. For five different types of baffles, a study was made and simulation results are shown in Figs. 6-8.

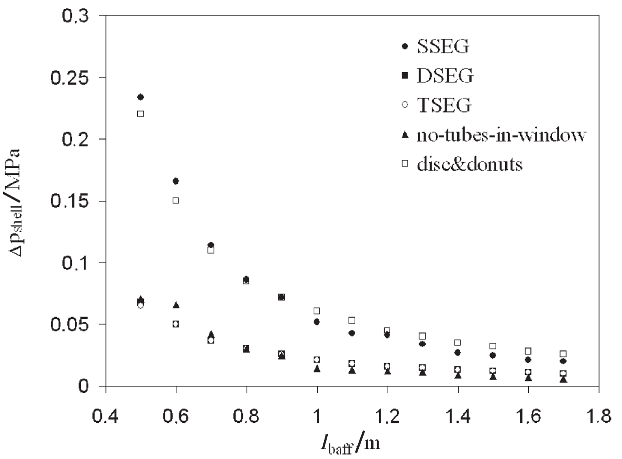
Fig. 6 shows fluid cross-flow rate dependence on baffle spacing for five different types of baffles. It can be seen that fluid cross-flow rate decreases with the increased baffle spacing. Also, the lowest cross-flow rates are achieved with double segmental and triple segmental baffles for spacing greater than 1.0 m.

Fig. 7 also shows that a lower pressure drop can be achieved using double and triple segmental baffles.

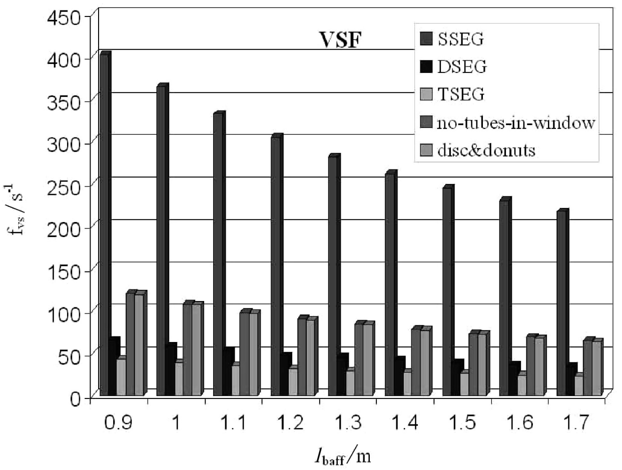
Fig. 8 shows how baffle spacing and type influence the vortex shedding frequency. The value of



F i g . 6 – *Baffle spacing influence on fluid cross-flow rate*



F i g . 7 – *Shell pressure drop dependence on baffle spacing*



F i g . 8 – *Vortex shedding frequency for different baffle spacing and different baffle types*

the vortex shedding frequency has an impact on the cross-flow rate of the fluid and vibration problem in heat exchanger. The vortex shedding frequency, in general, drops with the increased baffle spacing. It is important to notice that much lower value of vortex shedding frequency are achieved using double and triple segmental baffle, in comparison with single segmental baffles, constructed in the analyzed heat exchanger. In this case, two type of baffles, DSEG and TSEG, shows equally good results but DSEG baffles have the advantage because they are specialized for reduction of cross-flow rate for a given baffle spacing and they gave better critical to cross-flow rate ratio then TSEG.

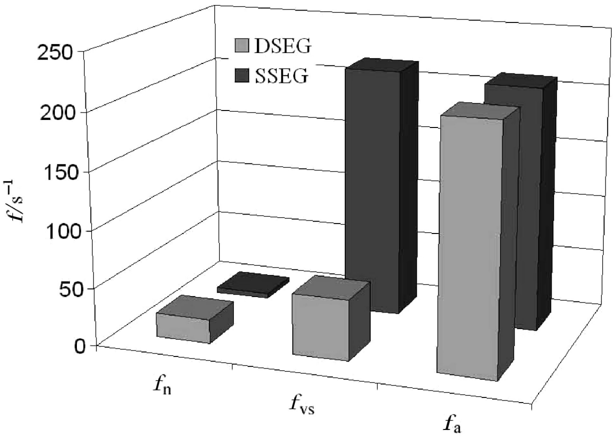
Based on obtained results, recommended solution of vibration problem of heat exchanger E-111 is to replace SSEG with DSEG baffles and to reduce the span between them from 1.70 m to 1.10 m so their number have to be increased from 5 to 8. In that case fluid cross-flow to critical rate ratio is 1.75, and the vortex shedding frequency is reduced by 4 times. This can remove the possibility of its coinciding with the bundle's tube natural frequency and acoustic frequency, eliminating vibration and noise problems. This approach retains the structural effectiveness of the tube bundle yet allows the gas to flow between alternating tube sections in a straighter overall direction, thereby reducing the effect of numerous direction changes.

The results obtained by simulation for 8 double segmental baffles with 1.10 m spacing are given in Table 4.

T a b l e 4 – *Results obtained by simulation for 8 DSEG baffles with 1.10 m spacing*

	Inlet	Center	Outlet
tube span, m	2.71	2.20	2.71
v_{cross}, m s^{-1}	4.58	6.71	4.58
v_{crit}, m s^{-1}	2.11	3.82	1.89
v_{cross}/v_{crit}	2.17	1.75	2.42
f_n, s^{-1}	12.3	23.5	12.3
f_a, s^{-1}	223.3	212.0	200.0
f_{vs}, s^{-1}	36.72	53.82	36.72
f_{tb}, s^{-1}	3.33	4.87	3.33
f_{vs}/f_n	3.00	2.30	3.00
f_{vs}/f_a	0.16	0.25	0.18
f_{tb}/f_n	0.27	0.21	0.27
f_{tb}/f_a	0.01	0.02	0.02

Fig. 9 shows that double segmental baffles greatly reduce vortex shedding frequency and that the differences between other frequencies are large enough not to coincide.



F i g . 9 – *Comparison of NF, VSF and AF values for SSEG and DSEG types of baffles*

Vibration analysis for heat exchanger E-114 was made also and the same procedure was followed as for heat exchanger E-111. The results of vibration analysis for E-114 are shown in Table 5.

T a b l e 5 – *Vibration analysis results for heat exchanger E-114*

	Inlet	Center	Outlet
tube span, m	0.53	0.55	0.53
v_{cross}, m s^{-1}	0.25	0.24	0.25
v_{crit}, m s^{-1}	5.36	5.39	5.36
v_{cross}/v_{crit}	0.05	0.04	0.05
f_n, s^{-1}	205.8	205.7	205.9
f_a, s^{-1}	0.0	0.0	0.0
f_{vs}, s^{-1}	7.68	7.38	7.68
f_{tb}, s^{-1}	1.44	1.38	1.44
f_{vs}/f_n	0.04	0.04	0.04
f_{vs}/f_a	0.0	0.0	0.0
f_{tb}/f_n	0.01	0.01	0.01
f_{tb}/f_a	0.0	0.0	0.0
vibration exists	NO	NO	NO

Fluidelastic instability was determinate by Connors method[15] and from the obtained results it can be seen that fluid cross-flow rate is below the critical one ($v_{cross} < v_{crit}$) which indicates that E-114 has no vibration problems. This is also confirmed by calculated low value of vortex shedding frequency and the value of acoustic frequency equalling zero.

Conclusion

A detailed calculation and vibration analysis of two heat exchangers of the nitric acid synthesis plant were made. Based on obtained results, the following may be concluded:

– Nitrous gases cooler E-111 has a vibration problem.

– Vibrations of E-111 are caused by coinciding values of the vortex shedding frequency and the acoustic frequency.

– The vortex shedding frequency can be reduced using double segmental baffles, rather than the existing single segmental, and therefore frequencies coinciding can be removed. This reduces the possibility of tube bundle vibration problems and noise.

– Vibrations cannot be entirely removed, because high-rate gas flows through the shell and tubes and the designed length of the tubes is 11.0 m.

– Heat exchanger E-114 has no vibration problem.

N o m e n c l a t u r e

d – shell inside diameter, m
D – tube outside diameter, m
f_n – tube natural frequency, s^{-1}
f_{vs} – vortex shedding frequency, s^{-1}
f_a – first acoustic frequency of the bundle, s^{-1}
f_{tb} – turbulent buffeting frequency, s^{-1}
l_m – span length, mm
l_{baff} – baffle spacing, m
L – support thickness, mm
m_t – linear mass density of tube, kg m^{-1}
M – molar mass, kg mol^{-1}
n – mode number, dimensionless integer
N – number of span
p_i – inlet pressure, MPa
Δp_{shell} – shell-side pressure drop, MPa
P – tube pitch, mm
P_l – longitudinal tube pitch, mm
P_t – transverse tube pitch, mm
q_m – mass flow rate, kg h^{-1}
R – gas constant, 8.314 J mol^{-1} K^{-1}
Re – Reynolds number
S – Strouhal number
U_s – velocity of sound in the shell side fluid, m s^{-1}
v_{cross} – cross-flow rate, m s^{-1}
v_{crit} – critical cross-flow rate, m s^{-1}
X_p – tube pitch ratio
z – compressibility factor, dimensionless

Greek letters

β – Connors' constant
θ – absolute temperature, °C
θ_i – inlet temperature, °C
θ_o – outlet temperature, °C
ζ_n – modal damping ratio
ρ – shell side fluid density, kg m^{-3}
γ – specific heat ratio, C_p/C_v
Φ – heat load, kW

Abbreviations

C/BEM – Tema class
SSEG – single segmental baffles
DSEG – double segmental baffles
TSEG – triple segmental baffles
VSF – vortex shedding frequency

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