Chapter 2

Selection of heat exchanger and HTRI analysis

2.1 Introduction

To determine at which flow rates vibration may occur, a number of HTRI analyses were done on the selected heat exchanger. The natural frequency equation from Chapter 1 (eq. 1.1), as well as FEM analyses, were used to determine additional natural frequencies of the tube.

2.2 Selection criteria

To select a heat exchanger from the approximately 2000 shell-and-tube heat exchangers at Sasol Synthetic Fuels (SSF), the following criteria were used to select a suitable heat exchanger to analyse.

- The shell- and tube-side flow must be single-phase flow. If the fluid on the tube- or shell-side is a gas and condensation or evaporation takes place in the heat exchanger, solving the problem becomes more complex.
- The shell-side fluid must be a gas with temperature not exceeding 100°C. The temperature range of equipment available to measure vibration on the heat exchanger is limited. If shell-side flow is a gas, all six mechanisms that can cause tube vibration (see paragraph 1.2.4) are present. Acoustic resonance only occurs if the fluid is a gas or vapour.
- The heat exchanger should not be covered with thermal insulation. For the experimental investigation, free access to the shell was needed for accelerometers and strain gauges.
- It must be possible to vary the load through the heat exchanger. Vibration measurements at different load conditions need to be taken to compare these values with predicted values.
- The flow rates, operating pressure and temperature at the inlet and outlet, should be known. The flow rate, pressure and temperature through the heat exchanger changes, depending on the process. When comparing the results, it is important to compare results with the same flow velocities, temperatures and

pressures. It is furthermore important to measure the flow velocities to see if the inlet flow velocity changes are causing any vibration in the heat exchanger.

The heat exchanger should also have a history of vibration-related problems. From the list of possible heat exchangers provided by SSF, the following two exchangers were examined in more detail:

- ES X06 Trim cooler
- ES 208 Tail gas heat exchanger

2.2.1 ES X06 Trim cooler

The ES X06 heat exchanger was a good candidate for vibration measurements, because it had serious vibration problems. The gas that passes through the shell-side of the heat exchanger consists of 224 components, which can be divided into three categories: Tail gas, reaction water and unstabilised light oil. Condensation of some of these components takes place, making this a two-phase problem. The heat exchanger also had unconventional baffles that could not be analysed with Heat Transfer Research Institute (HTRI) or Heat Transfer and Fluid flow Service (HTFS). The ES 208 Tail gas heat exchanger was therefore analysed, for it did not pose any such problems.

2.2.2 ES 208 Tail gas shell-and-tube heat exchanger

The heat exchanger is a CEN 1020-6100 type (figure 2.1) shell-and-tube heat exchanger with a square tube configuration and triple segmental baffles. A HTRI analysis was used to obtain vibration information because of its triple segmental baffle capabilities (figure 2.2).

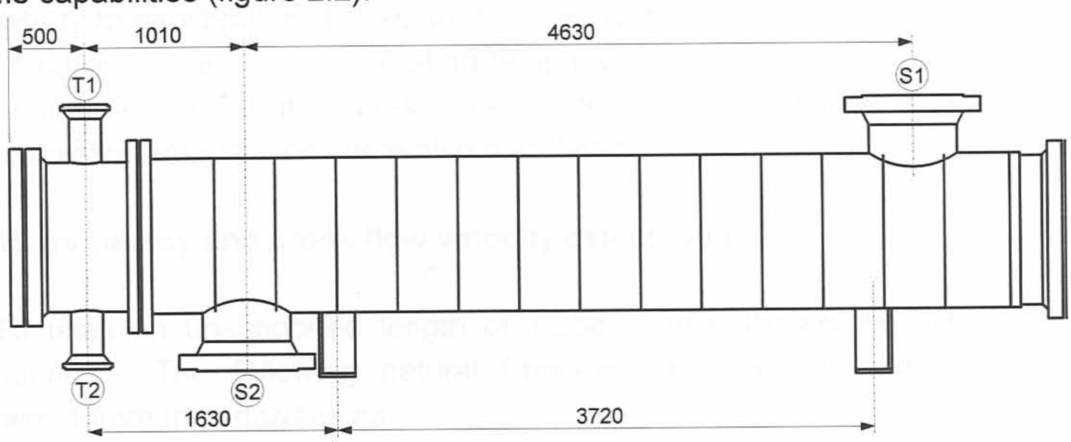

Figure 2.1: CEN type heat exchanger

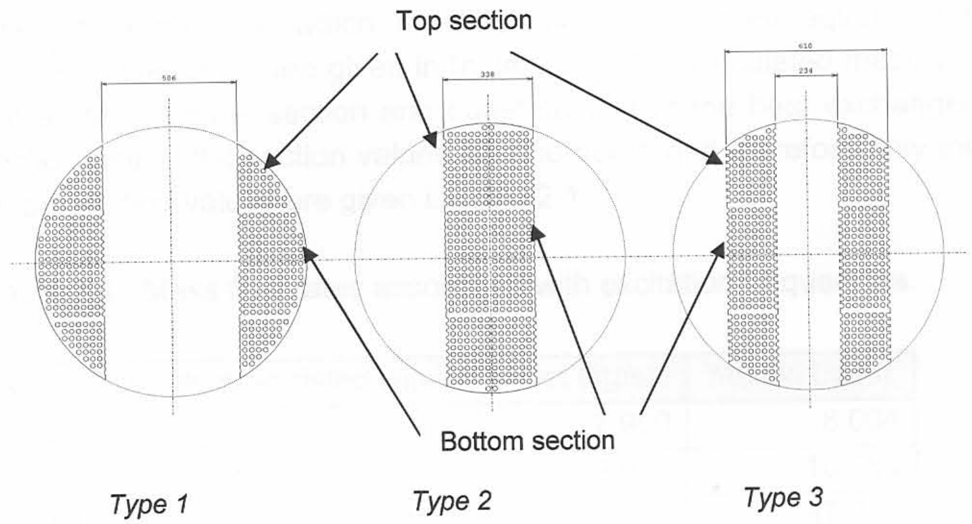

Figure 2.2: Triple segmental baffles

The heat exchanger has one shell-side pass with an inside diameter of 1.02m and six tube-side passes with 1100 tubes each 6.1m long with an outside diameter of 19mm. Tail gas with a density of *21 kg/m*^{3} (Appendix A) enters the shell-side of the heat exchanger at S1 (see figure 2.1) with a temperature of 55°C and exits the heat exchanger at S2 with a temperature of 47°C. On the tube-side, water enters the heat exchanger at T1 with a temperature of 30°C and exits at T2 after six tube passes with a temperature of 43°C. The operating pressure on the shell-side of the heat exchanger is 3.225 MPa and 0.35 MPa on the tube-side.

2.3 HTRI Analysis

The HTRI software is mainly used to design heat exchangers, but it also has the capability to do vibration checks on the heat exchanger tubes. HTRI analyses on the heat exchanger at the operating temperature and pressure, were performed for a number of mass flow rates. The temperature and pressure dependence of the excitation frequencies, were also calculated.

2.3.1 HTRI frequency and cross flow velocity calculation

HTRI uses an unsupported length of 1.268 m to calculate the lowest natural frequency. The following natural frequency and acoustic frequency were obtained from the analyses as:

$$f_n = 28.58Hz \qquad f_a = 174.75Hz \qquad (2.1)$$

The mass flow rates at which the excitation frequencies equal the natural frequencies of the tubes, are given in table 2.1. HTRI calculated these values at the inlet section, middle section and outlet section of the heat exchanger. The inlet section and outlet section values are the same and therefore only the inlet- and middle section values are given in table 2.1.

Table 2.1: Mass flow rates associated with excitation frequencies.

Mass flow rate associated with:	Inlet (kg/s)	Middle (kg/s)
Vortex shedding (f_{vs})	7.950	8.004
Turbulence buffeting (f_{tb})	9.982	10.054
Acoustic vortex shedding (f_{vsa})	46.713	47.073
Acoustic turbulence buffeting (f_{tba})	66.134	66.601

The acoustic frequency of the heat exchanger is not within a range of 20 percent of the natural frequency of the tubes. This means that no tube vibration due to acoustic excitation, will take place. For this reason only the vortex shedding frequency and turbulence buffeting frequency will be looked at in more detail.

The maximum vibration amplitude of the tubes, as calculated by the HTRI analyses for a mass flow range between 5 kg/s and 100 kg/s, is 0.3 mm. Figure 2.3 a) shows the amplitude of vibration for the inlet section (blue) and outlet section (red) as functions of the mass flow rate.

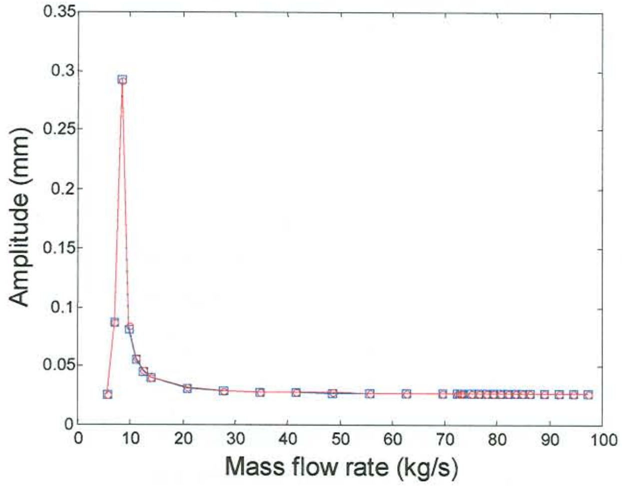

Figure 2.3 a): Vibration amplitude.
(blue – inlet section, red – outlet section)

The vibration amplitude is relatively small and no collision damage will occur because the minimum clearance between adjacent tubes is 7 mm. The vibration may cause fatigue and baffle damage (refer to paragraph 1.2.6) as well as tube sheet damage, especially if there are thermal stresses involved.

In the close-up view of the vibration amplitude graph (figure 2.3 b), the maximum amplitude is at the mass flow rate where the vortex shedding frequency coincides with the tube's natural frequency.

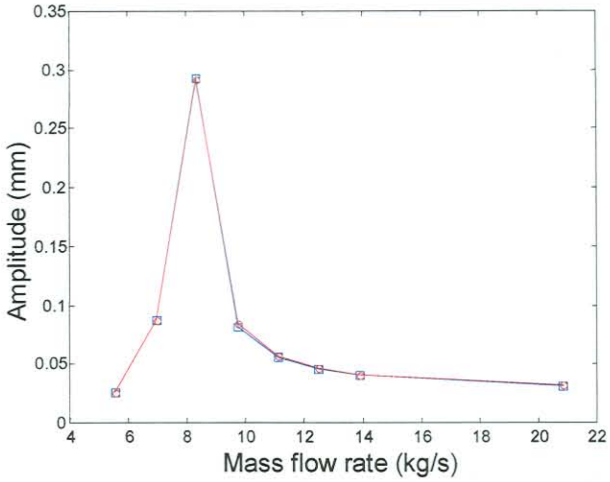

Figure 2.3 b): Close-up view of vibration amplitude.
(blue – inlet section, red – outlet section)

At a mass flow rate of about 14.234 kg/s the HTRI average cross flow velocity is equal to the HTRI calculated fluid-elastic instability critical velocity.

$$c_r = 2.89 m/s \qquad 2.2$$

No vibration amplitude calculation is given for fluid-elastic instability, but the tube vibration is assumed to increase as the cross flow velocity increases beyond the critical value, until the neighbouring tubes limit the amplitude, causing collision damage (refer to figure 1.3). If the fluid-elastic instability vibration curve published by HTFS is superimposed on to the vibration curve given by the HTRI analysis, (figure 2.3), the vibration amplitude curve shown in figure 2.4 is obtained. It is important to remember that the assumption is made that the entire unsupported tube length is subjected to the single average cross flow velocity. This is hardly ever the case in shell-and-tube heat exchangers, as the average cross flow velocity is also a function of the flow patterns through the heat exchanger, as caused by the baffle plates.

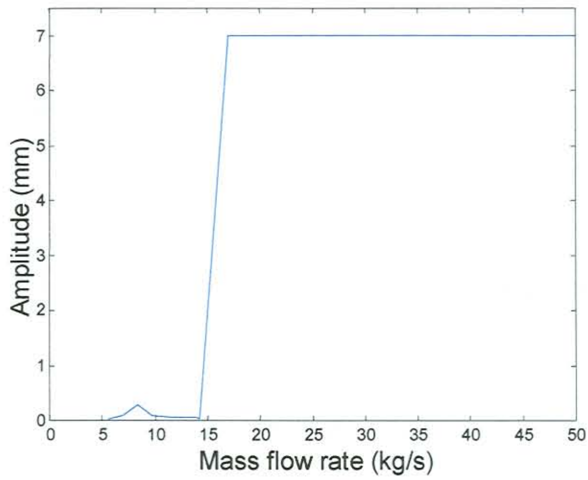

Figure 2.4: Vibration curve including FEI

2.3.2 Pressure and temperature dependence

The pressure and temperature dependence of the excitation frequencies, were obtained at a mass flow rate of 73 kg/s. Figures 2.5 and 2.6 give the inlet temperature dependence of the vortex shedding and turbulence buffeting frequencies respectively. These two frequencies are not functions of the inlet pressure. The acoustic vortex shedding and turbulence buffeting frequency on the other hand are only dependent on the inlet pressure and not the temperature.

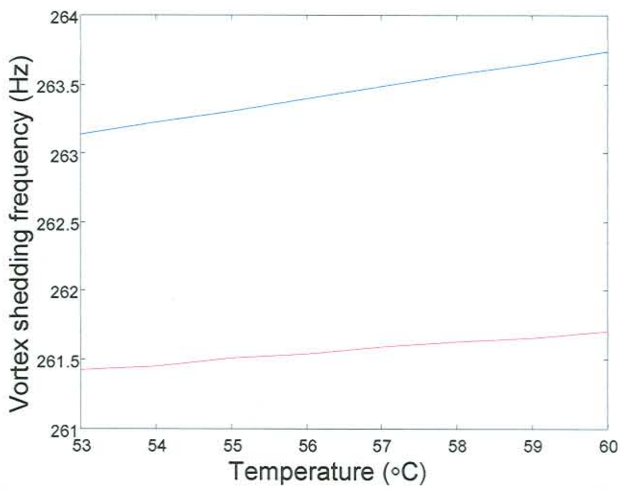

Figure 2.5: Temperature dependence of the vortex shedding frequency (blue – inlet section, red – middle section)

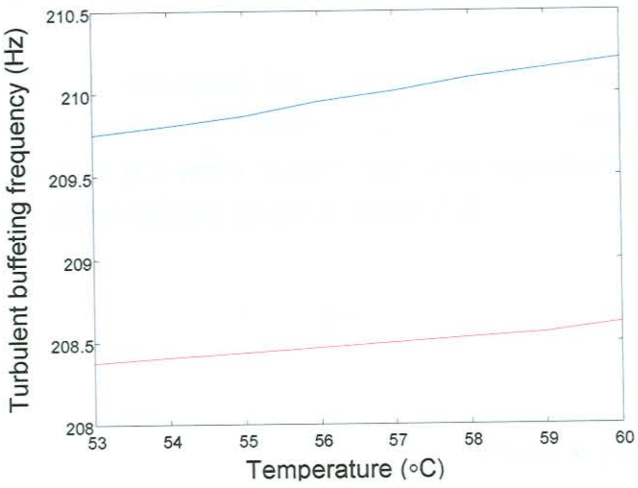

Figure 2.6: Temperature dependence of the turbulence buffeting frequency (blue – inlet section, red – middle section)

The difference in vortex shedding and turbulence buffeting frequency over the given temperature range, is less than 0.4 percent and can therefore be neglected when working in the specified temperature range.

2.3.3 Pressure drop through the heat exchanger

The HTRI-calculated pressure drop through the heat exchanger, is given in figure 2.7 for a range of mass flow rates. These values will be compared with CFD results (Chapter 3) to obtain the mass flow rates through the heat exchanger for measuring purposes (Chapter 4).

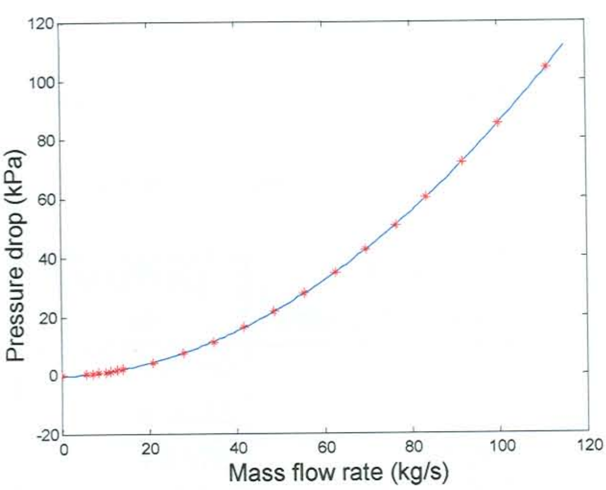

Figure 2.7: HTRI calculated pressure drop through the heat exchanger

2.4 Additional natural frequency calculations

The HTRI analysis only calculated the lowest natural frequency. To obtain all the natural frequencies, equation 1.1 (repeated as eq. 2.3) was used. The different combinations of lengths (L) were substituted into equation 2.3 and the different natural frequencies obtained are given in table 2.2.

$$f_n = \frac{C_n}{2\pi}\left(\frac{EI}{M_e L^4}\right)^{0.5} \qquad 2.3$$

C_n = 10 from figure 1.1 and E = 210 GPa. The moment of inertia (eq. 2.4) and the equivalent mass (eq. 2.5) were calculated as follows:

$$I = \frac{\pi\left(D_o^4 - D_i^4\right)}{64} = 3.415\times10^{-9} \qquad 2.4$$

$$M_e = M_m + M_t + M_s = 0.916 kg/m \qquad 2.5$$

The equivalent mass per unit length (M_e) is the sum of the fluid inside the tube per unit length (M_t), the tube material per unit length (M_m) and the virtual mass per unit length (M_s) of the tube for shell-side fluid displaced by the tube (equation 2.6).

$$M_s = \frac{k\rho_s \pi D_o^2}{4} = 1.393\times10^{-2} kg/m \qquad 2.6$$

The added mass coefficient (k) is a function of the tube pitch to diameter ratio as well as the tube layout as given by the TEMA standards (1988).

Table 2.2: Natural frequencies of unsupported tube lengths

Length (L) (m)	Natural frequency (Hz)
0.8450	62.45
0.8635	59.80*
1.2675	27.76
1.2860	26.96*

* Value calculated using equation 2.3 with assumption that both ends of the tube are pinned.

If the above-calculated natural frequency for an unsupported length of 1.2675 m is compared to the value given by the HTRI analyses, it differs by 0.82 Hz or 3 percent. This is due to variations in added mass and natural frequency coefficients.

This is not the longest unsupported length. There are a small number of tubes that have an unsupported length of 1.286 m at the inlet and outlet section of the heat exchanger, because of the 0.441 m spacing between the tubesheet and baffles.

The natural frequency calculation in table 2.2 was done using equation 2.3. This equation was formulated with the assumption that the tube ends are pinned at the baffle plates. At the tubesheets this is not the case and the natural frequencies need to be calculated using fixed constraints at the one end (tubesheet) and pinned constraints at the other end (baffle plate).

A Finite Element Method Analysis (FEA) was used to determine the mode shapes and natural frequencies with pinned-pinned and fixed-pinned supports for the four different tube lengths.

In all the FEA's an equivalent density was used to compensate for the tube-side fluid mass and virtual shell-side fluid mass.

Figure 2.7 shows a tube between baffle plates and this was analysed using pinned supports at both ends. It represents the mode shape of the first natural frequency (61.64 Hz) of the tube. This value compares well to the TEMA calculated value (62.45 Hz) in table 2.2.

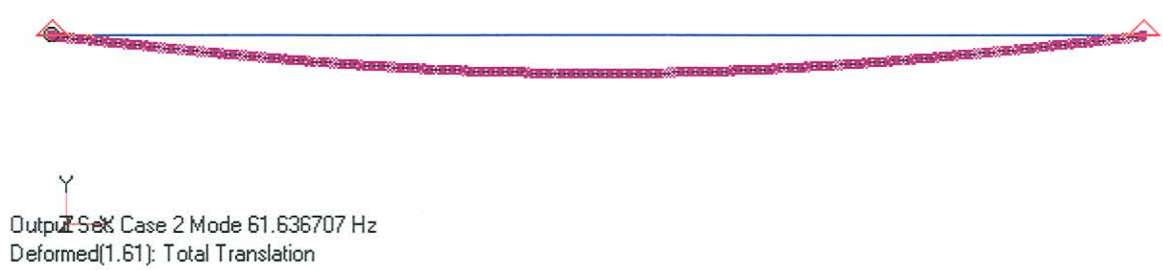

Figure 2.7: Mode shape of tube with length 0.845 m

In figure 2.8 a) the natural frequency associated with pinned supports is 59.023 Hz which is very close to the TEMA calculated value (59.8 Hz) in table 2.2.

This frequency is inaccurate because of the assumption of pinned-pinned supports mentioned earlier. If the tube is analysed with the correct constraints (fixed-pinned) as shown in figure 2.8 b) a natural frequency of 92.21 Hz is obtained.

Figure 2.8 a) Mode shape of tube with length 0.8635 m with pinned supports

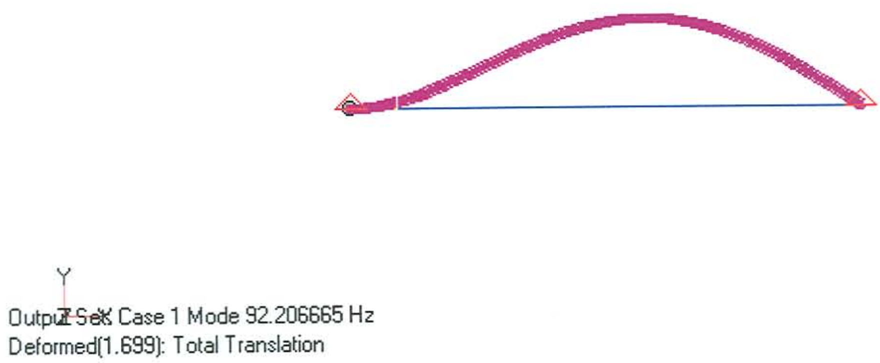

Figure 2.8 b) Mode shape of tube with length 0.8635 m with fixed-pinned supports

For the tube section in figure 2.9, the tube is again between two baffle plates and the assumption used to calculate the natural frequency holds. The natural frequency given by TEMA is 27.76 Hz which compares well with the FEA value of 27.39 Hz.

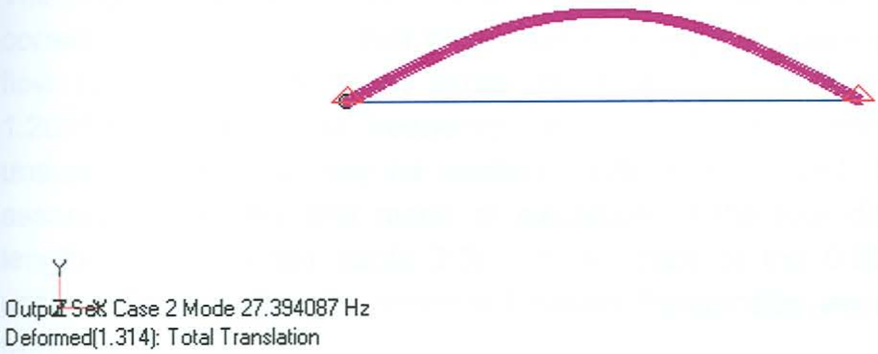

Figure 2.9: Mode shape of tube with length 1.2675 m with pinned supports

Figures 2.10 a) and b) give the mode shapes for pinned-pinned and fixed-pinned supports respectively. In the actual heat exchanger, the one end of the tube is fixed at the tubesheet and the associated natural frequency that should be used, is 41.57 Hz.

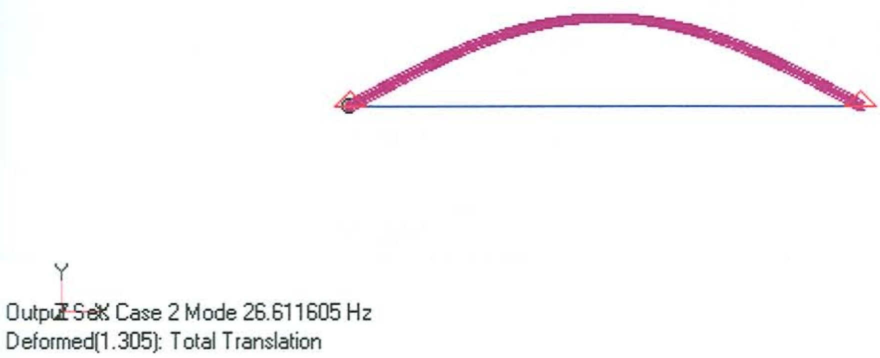

Figure 2.10 a) Mode shape of tube with length 1.286 m with pinned supports

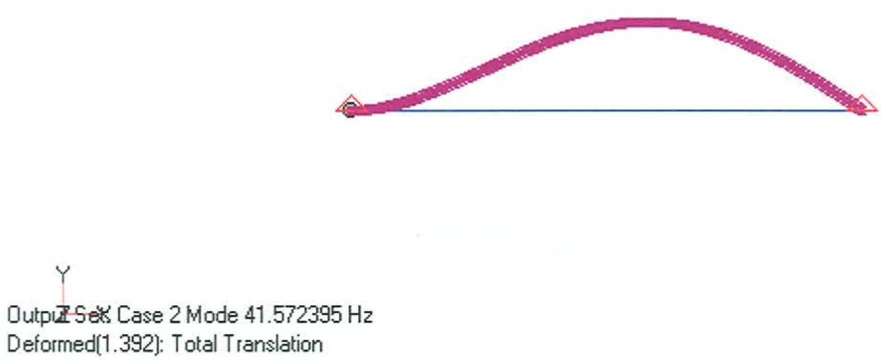

Figure 2.10 b) Mode shape of tube with length 1.286 m with fixed-pinned supports

This implies that the lowest natural frequency calculated by the HTRI analysis, is correct. The problem is that if the heat exchanger is operated at a higher mass flow rate, which would not excite the tubes with an unsupported length of 1.2675 m at its lowest frequency, one of the other tubes (with a different unsupported length) may be excited. With this in mind, the mass flow rate associated with the first mode of excitation of the four different unsupported lengths was obtained (table 2.3). In the case of the 0.8635 m and 1.285 m unsupported lengths, the corrected natural frequencies were used as obtained from the FEA.

Only the first natural frequencies for the four different unsupported lengths were used, because the other natural frequencies were above 100 Hz and the vibration amplitudes related to those frequencies, were assumed to be small relative to the amplitudes of the first natural frequency.

Table 2.3: Mass flow rates associated with tube vibration

		Associated mass flow rate (kg/s)			
Natural frequency		27.76Hz	41.57Hz	59.80Hz	92.21Hz
Vortex shedding frequency	Inlet	7.75	11.61	16.70	25.69
	Middle	7.80	11.75	16.84	25.87
Turbulence buffeting frequency	Inlet	9.78	14.58	20.98	32.2
	Middle	9.85	14.68	21.12	32.42

2.5 Comparison of natural frequencies and associated mass flow rates

In table 2.4 the lowest natural frequency value obtained from TEMA equation (table 2.3) and the finite element analysis are compared to values obtained in the HTRI analyses (equation 2.3) of 28.58 Hz.

Table 2.4: Comparison of natural frequencies to HTRI values

	Frequency	Difference (%)
TEMA	27.76	2.869
FEM	27.39	4.164

In table 2.5, the mass flow rates associated with vortex shedding and turbulence buffeting for the lowest natural frequencies obtain from the TEMA calculation, (table 2.3) are compared to the HTRI calculated value (table 2.1).

Table 2.5: Comparison of mass flow rates

		Associated mass flow rate (kg/s)		
		HTRI	TEMA	% difference
Vortex shedding frequency	Inlet	7.95	7.75	2.52
	Middle	8.00	7.80	2.50
Turbulence buffeting frequency	Inlet	9.98	9.78	2.00
	Middle	10.50	9.85	1.99

There are only small differences between the HTRI and TEMA natural frequencies and associated mass flow rate values as shown in tables 2.4 and 2.5. The difference between the HTRI and TEMA natural frequencies is due to a small difference in the calculation of the effective mass per unit length (M_e) in equation 2.5.

The HTRI results compare well with the TEMA results and can therefore be used for comparison purposes with the CFD results in Chapter 3 as well as the experimental results in Chapter 4.

If the above-calculated natural frequency for an unsupported length of 1.2675 m is compared to the value given by the HTRI analyses, it differs by 0.82 Hz or 3 percent. This is due to variations in added mass and natural frequency coefficients.

This is not the longest unsupported length. There are a small number of tubes that have an unsupported length of 1.286 m at the inlet and outlet section of the heat exchanger, because of the 0.441 m spacing between the tubesheet and baffles.

The natural frequency calculation in table 2.2 was done using equation 2.3. This equation was formulated with the assumption that the tube ends are pinned at the baffle plates. At the tubesheets this is not the case and the natural frequencies need to be calculated using fixed constraints at the one end (tubesheet) and pinned constraints at the other end (baffle plate).

A Finite Element Method Analysis (FEA) was used to determine the mode shapes and natural frequencies with pinned-pinned and fixed-pinned supports for the four different tube lengths.

In all the FEA's an equivalent density was used to compensate for the tube-side fluid mass and virtual shell-side fluid mass.

Figure 2.7 shows a tube between baffle plates and this was analysed using pinned supports at both ends. It represents the mode shape of the first natural frequency (61.64 Hz) of the tube. This value compares well to the TEMA calculated value (62.45 Hz) in table 2.2.

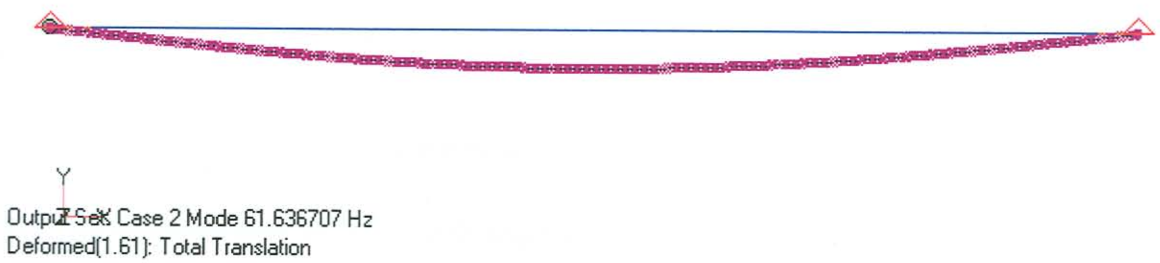

Figure 2.7: Mode shape of tube with length 0.845 m

In figure 2.8 a) the natural frequency associated with pinned supports is 59.023 Hz which is very close to the TEMA calculated value (59.8 Hz) in table 2.2.

This frequency is inaccurate because of the assumption of pinned-pinned supports mentioned earlier. If the tube is analysed with the correct constraints (fixed-pinned) as shown in figure 2.8 b) a natural frequency of 92.21 Hz is obtained.

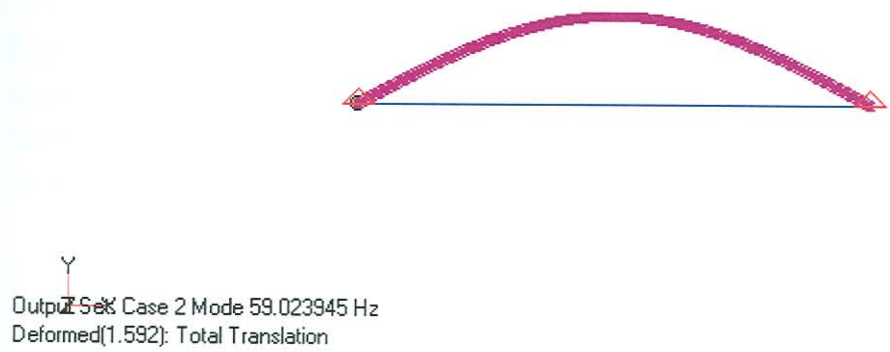

Figure 2.8 a) Mode shape of tube with length 0.8635 m with pinned supports

Figure 2.8 b) Mode shape of tube with length 0.8635 m with fixed-pinned supports

For the tube section in figure 2.9, the tube is again between two baffle plates and the assumption used to calculate the natural frequency holds. The natural frequency given by TEMA is 27.76 Hz which compares well with the FEA value of 27.39 Hz.

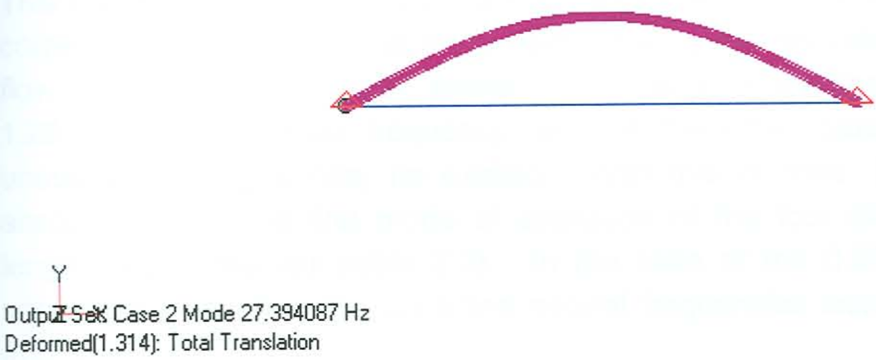

Figure 2.9: Mode shape of tube with length 1.2675 m with pinned supports

Figures 2.10 a) and b) give the mode shapes for pinned-pinned and fixed-pinned supports respectively. In the actual heat exchanger, the one end of the tube is fixed at the tubesheet and the associated natural frequency that should be used, is 41.57 Hz.

Figure 2.10 a) Mode shape of tube with length 1.286 m with pinned supports

Figure 2.10 b) Mode shape of tube with length 1.286 m with fixed-pinned supports

This implies that the lowest natural frequency calculated by the HTRI analysis, is correct. The problem is that if the heat exchanger is operated at a higher mass flow rate, which would not excite the tubes with an unsupported length of 1.2675 m at its lowest frequency, one of the other tubes (with a different unsupported length) may be excited. With this in mind, the mass flow rate associated with the first mode of excitation of the four different unsupported lengths was obtained (table 2.3). In the case of the 0.8635 m and 1.285 m unsupported lengths, the corrected natural frequencies were used as obtained from the FEA.

Only the first natural frequencies for the four different unsupported lengths were used, because the other natural frequencies were above 100 Hz and the vibration amplitudes related to those frequencies, were assumed to be small relative to the amplitudes of the first natural frequency.

Table 2.3: Mass flow rates associated with tube vibration

		Associated mass flow rate (kg/s)			
Natural frequency		27.76Hz	41.57Hz	59.80Hz	92.21Hz
Vortex shedding frequency	Inlet	7.75	11.61	16.70	25.69
	Middle	7.80	11.75	16.84	25.87
Turbulence buffeting frequency	Inlet	9.78	14.58	20.98	32.2
	Middle	9.85	14.68	21.12	32.42

2.5 Comparison of natural frequencies and associated mass flow rates

In table 2.4 the lowest natural frequency value obtained from TEMA equation (table 2.3) and the finite element analysis are compared to values obtained in the HTRI analyses (equation 2.3) of 28.58 Hz.

Table 2.4: Comparison of natural frequencies to HTRI values

	Frequency	Difference (%)
TEMA	27.76	2.869
FEM	27.39	4.164

In table 2.5, the mass flow rates associated with vortex shedding and turbulence buffeting for the lowest natural frequencies obtain from the TEMA calculation, (table 2.3) are compared to the HTRI calculated value (table 2.1).

Table 2.5: Comparison of mass flow rates

		Associated mass flow rate (kg/s)		
		HTRI	TEMA	% difference
Vortex shedding frequency	Inlet	7.95	7.75	2.52
	Middle	8.00	7.80	2.50
Turbulence buffeting frequency	Inlet	9.98	9.78	2.00
	Middle	10.50	9.85	1.99

There are only small differences between the HTRI and TEMA natural frequencies and associated mass flow rate values as shown in tables 2.4 and 2.5. The difference between the HTRI and TEMA natural frequencies is due to a small difference in the calculation of the effective mass per unit length (M_e) in equation 2.5.

The HTRI results compare well with the TEMA results and can therefore be used for comparison purposes with the CFD results in Chapter 3 as well as the experimental results in Chapter 4.