CHAPTER 9

EXPERIMENTAL EVALUATION

With all the parameters of the experimental model known, the validity of the simulation models could be determined by comparing the real vibration measurements with the values predicted by the mathematical models.

9.1. Natural frequency comparison

As a preliminary check to ensure that the theoretical and experimental models were comparable, the measured natural frequencies from section 7.5 was compared to the predicted natural frequencies from section 8.3.1. The comparison is shown in Table 17.

Table 17: Comparison between predicted and measured natural frequencies

	Predicted Values		Measured Values	
	Stiff steel-mounted case	Soft rubber-mounted case	Stiff steel-mounted case	Soft rubber-mounted case
First natural frequency [Hz]	19.387	10.038	18.375	8.500
Second natural frequency [Hz]	603.805	28.683	-	24.500

The measured natural frequencies compared favourably with the predicted natural frequencies in all the cases, being within 20% of the predicted values.

Due to the close correlation between all the predicted and measured values, it was seen fit to continue with the experimental measurements.

9.2. Testing procedure

To estimate the effectiveness of the vibration-control concept, the dynamic forces in the plates and other elastic elements were compared between the stiff steel-mounted case (comparable to the current assembly of the column-top condensers) and the case where soft rubber mounts and compensators isolated the model from the frame.

A two-channel FFT analyser was used to measure the acceleration of both the top of the heat exchanger frame ($\ddot{x}_1$) and the bottom of the plate pack ($\ddot{x}_2$).

Therefore, the model had to be assembled for the stiff steel-mounted case (as illustrated in Figure 83) and the displacement of the plates determined at two different frequencies by means of acceleration meters and the vibration analyser.

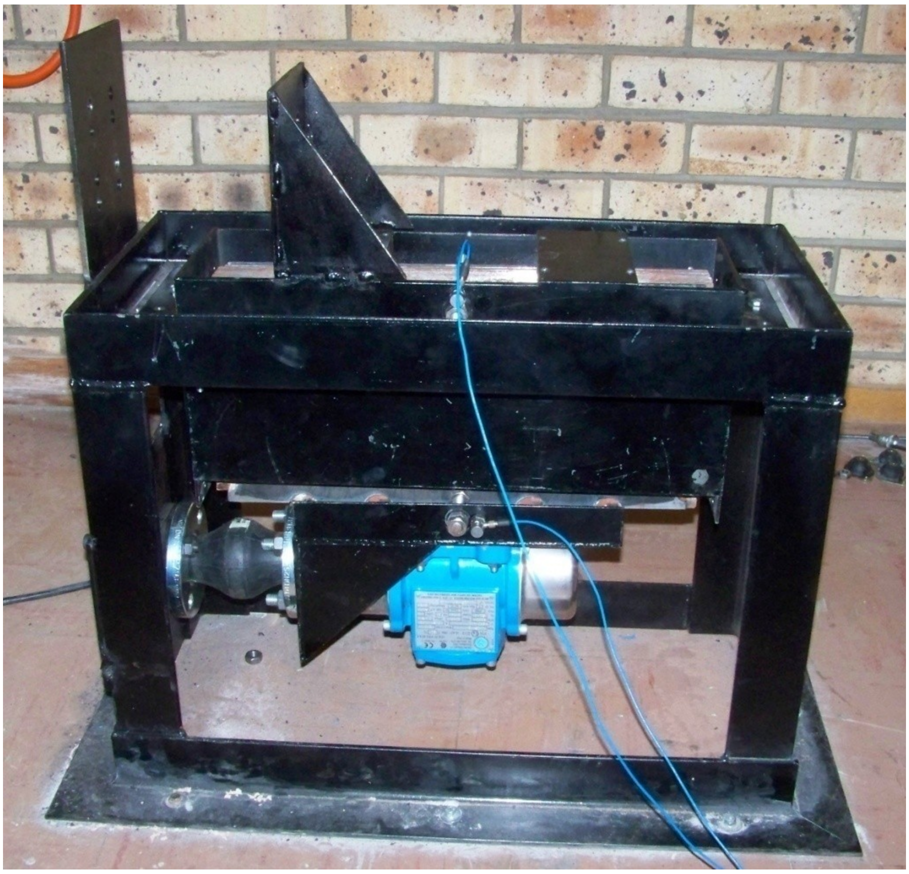

Figure 81: The attachment of the accelerometers on the stiff steel-mounted model

The two frequencies chosen for experimental measurements were 12.125 Hz (± 75 rad/s) and 17 Hz (± 106 Hz), which indicated relatively good reductions in dynamic force in the plate pack (F_{k1}).

From the measured acceleration values for both the selected forcing frequencies, the RMS velocities and displacements were calculated. From these values, together with the characterised values, the resulting forces in the different elements could be determined, using the Equations in section 5.2.

The model then had to be assembled for the soft rubber-mounted case, with the mounts and the top compensator included, as illustrated in Figure 82.

Figure 82: The connection of accelerometers on the soft rubber-mounted model

Measurements were taken at the same forcing frequencies as taken in the stiff steel-mounted case. The same equations used in the soft rubber-mounted case were used to calculate the resulting forces in the different elements of the model.

As the measurements for both the mounting cases were taken at the same forcing frequencies, the calculated forces in the different elements could be compared to each other in order to determine the effect of the change in mounting system design. This comparison was done by calculating the ratio of the force in the elements when the model was mounted with the soft rubber mounts to the force in the element when the model was mounted with the stiff steel bolts.

9.3. Experimental results

The results of the experiments were captured with a two-channel FFT vibration analyser. These values were then graphically represented, using Matlab.

9.3.1. Measured response at 12.125 Hz

The response of the system was measured for the stiff steel-mounted and soft rubber-mounted configuration at a value that closely corresponded with the natural frequency of the stiff steel-mounted system. This value therefore indicated a case where periodic wake shedding could be causing excessive vibration amplitudes.

9.3.1.1. *Stiff steel-mounted case*

The model was mounted stiffly with bolts to the frame and the vibrating motor was set to a forcing frequency of 12.125 Hz. The resulting measured acceleration of the top frame ($\ddot{x}_1$) is illustrated in Figure 83.

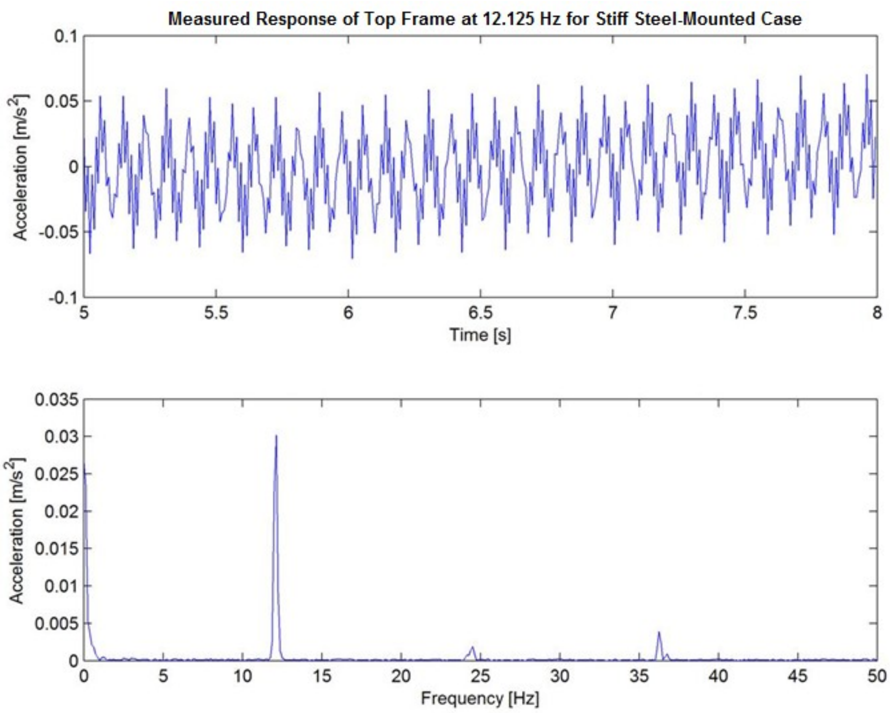

Figure 83: Measured acceleration of the top frame at a frequency of 12.125 Hz, stiff steel-mounted case

The acceleration for the bottom frame is illustrated in Figure 84.

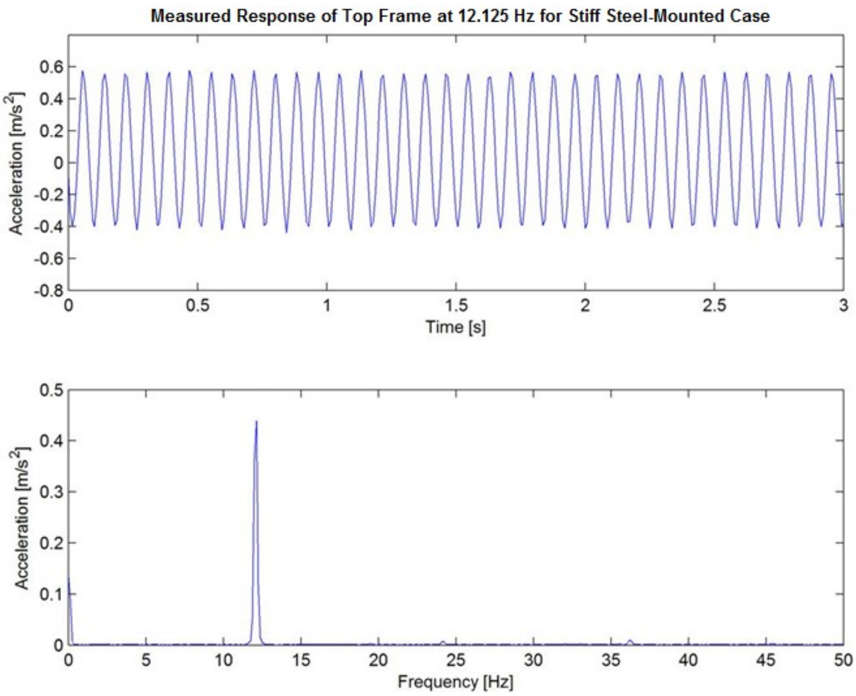

Figure 84: Measured acceleration of the bottom frame at a frequency of 12.125 Hz, stiff steel-mounted case

9.3.1.2. *Soft rubber-mounted case*

When soft rubber mounts were included to isolate the system, the resulting forces in the plates were predicted to reduce the theoretical models.

As in the case where the model was mounted to the structure with the stiff steel bolts, the measurements were taken at 12.125 Hz. The measured acceleration of the top frame is illustrated in Figure 85.

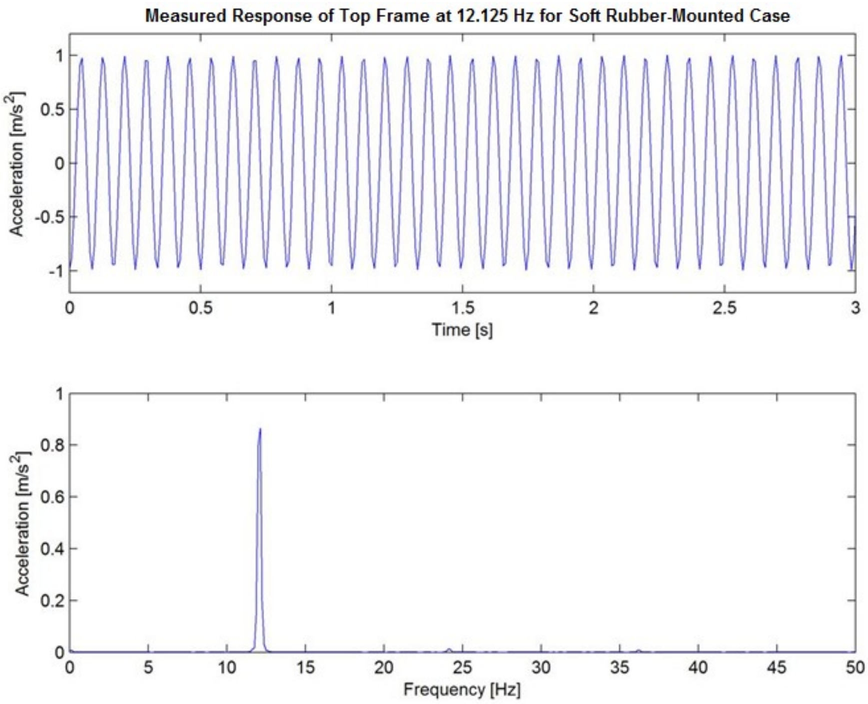

Figure 85: Measured acceleration of the top frame at a frequency of 12.125 Hz, soft rubber-mounted case

The measured acceleration of the bottom frame is illustrated in Figure 86.

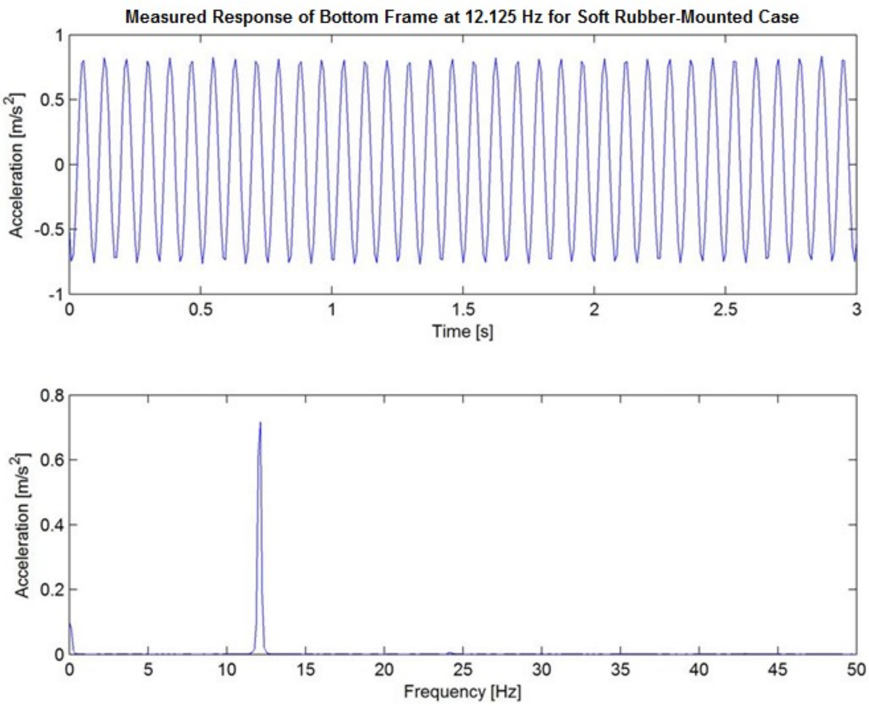

Figure 86: Measured acceleration of the bottom frame at a frequency of 12.125 Hz, soft rubber-mounted case

9.3.2. Measured response at 17 Hz

The other frequency chosen to verify experimentally, was 17 Hz.

9.3.2.1. *Stiff steel-mounted case*

When the model was mounted with stiff steel bolts to the frame, the measured acceleration of the top frame showed the following pattern, as is illustrated in Figure 87.

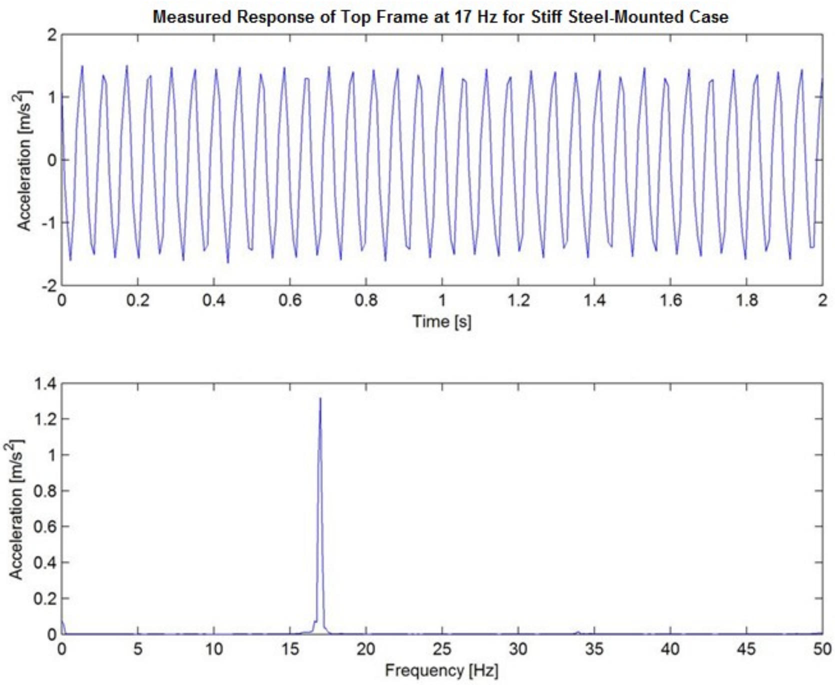

Figure 87: Measured acceleration of the top frame at a frequency of 17 Hz, stiff steel-mounted case

The measured acceleration of the bottom frame is illustrated in Figure 88.

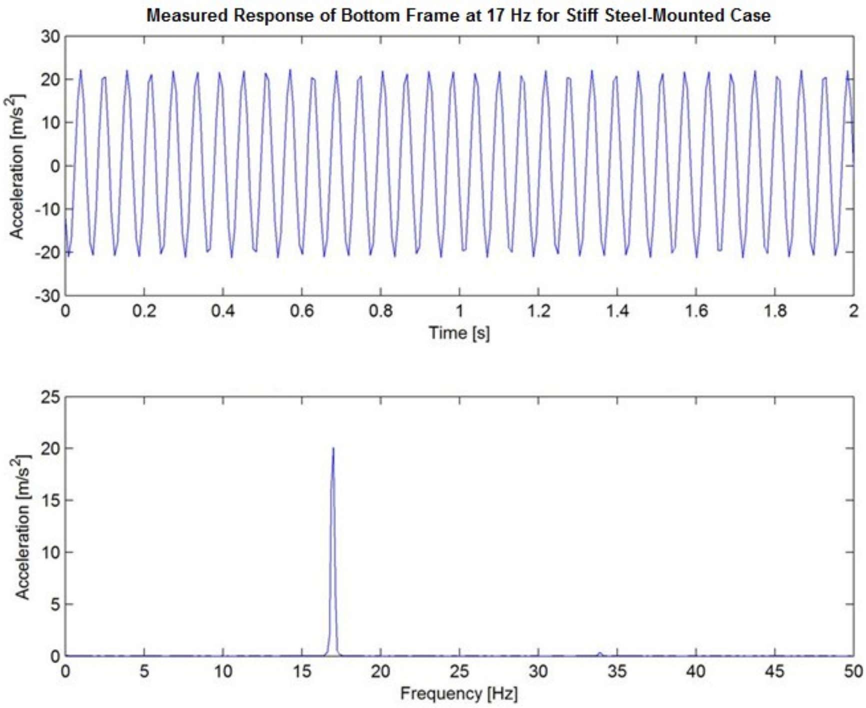

Figure 88: Measured acceleration of the bottom frame at a frequency of 17 Hz, stiff steel-mounted case

9.3.2.2. *Soft rubber-mounted case*

The measurements were also taken at 17 Hz for the case where the model had been isolated from the frame by soft rubber mounts. The measured acceleration of the top frame is illustrated in Figure 89.

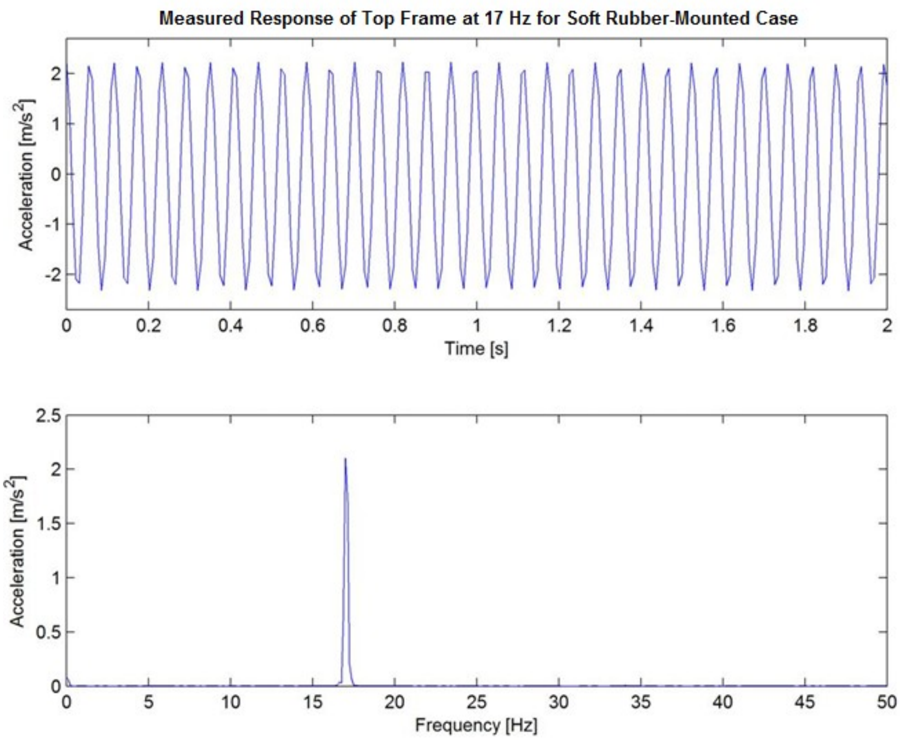

Figure 89: Measured acceleration of the top frame at a frequency of 17 Hz, soft rubber-mounted case

The measured acceleration of the bottom frame is illustrated in Figure 90.

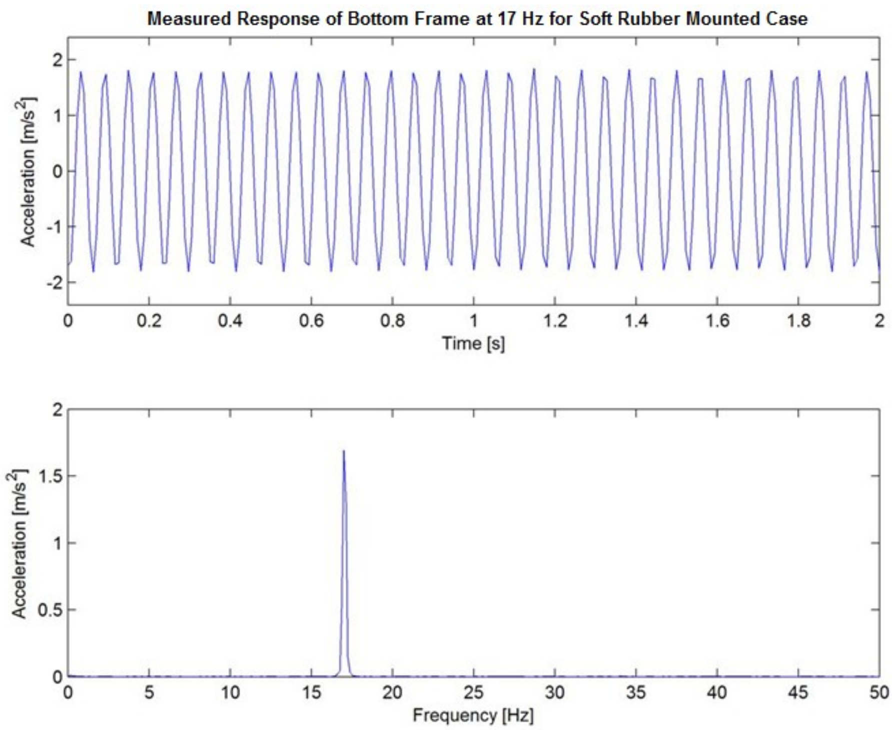

Figure 90: Measured acceleration of the bottom frame at a frequency of 17 Hz, soft rubber-mounted case

9.3.3. Processing of measured data

The measured acceleration data for the different forcing frequencies and mounting cases was compiled and converted into velocity and displacement values. The results are shown in Table 18.

With the displacement and velocity known, the resulting dynamic forces in each of the elements can be calculated with the equations from section 5. The calculated values are shown in Table 19.

A good approximation for the RMS value of an oscillating force is given by multiplying the amplitude with 0.707. This was done to the values in Table 19. The results are shown in Table 20.

The values indicated in Table 20 are the values that will be compared to the values predicted by the mathematical models.

9.4. Comparison with theoretical predictions

Given the complexity and cost of building an experimental model of each heat exchanger to determine the parameters of an effective mounting system, a mathematical model that can be used for any heat exchanger is preferable. The accuracy of such a model is very important.

The accuracy of the models in this case can be determined by comparing the experimental values with the theoretical predictions. If the models were accurate enough, they could be used to design mounting systems for other dimple plate heat exchangers, without the need of extensive experimental work.

As a starting point, the measured data was summarized as illustrated in Table 18, Table 19 and Table 20.

Table 18: Converted measured data summary

Forcing frequency		12.125 Hz (≈75 rad/s)		17 Hz (≈106 rad/s)	
Mounting case		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Measured acceleration amplitude of m_1 [m/s^2]	$\ddot{X}_1$	3.019E-02	8.652E-01	1.319E+00	2.104E+00
Velocity amplitude of m_1 [m/s]	$\dot{X}_1$	3.962E-04	1.136E-02	1.235E-02	1.970E-02
Displacement amplitude of m_1 [m]	X_1	5.201E-06	1.491E-04	1.156E-04	1.844E-04
Measured acceleration amplitude of m_2 [m/s^2]	$\ddot{X}_2$	4.393E-01	7.172E-01	2.011E+01	1.691E+00
Velocity amplitude of m_2 [m/s]	$\dot{X}_2$	5.767E-03	9.414E-03	1.882E-01	1.583E-02
Displacement amplitude of m_2 [m]	X_2	7.570E-05	1.236E-04	1.762E-03	1.482E-04

Table 19: Calculated dynamic forces from measured data

Forcing frequency		12.125 Hz (≈75 rad/s)		17 Hz (≈106 rad/s)	
Mounting case		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Force in element k_1 [N]	F_{1k}	1664.252	23.976	37005.524	29.657
Force in element k_2 [N]	F_{2k}	22.559	8.162	526.926	11.572
Force in element k_3 [N]	F_{3k}	2.765	4.514	64.375	5.415
Force in element c_1 [N]	F_{1c}	0.000	3.410	0.000	5.914
Force in element c_2 [N]	F_{2c}	0.235	0.085	7.686	0.169
Force in element c_3 [N]	F_{3c}	0.155	0.253	5.050	0.425
Combined force in element k_1 and c_1 [N]	F_1	1664.252	24.218	37005.524	30.241
Combined force in element k_2 and c_2 [N]	F_2	22.560	8.162	526.982	11.573
Combined force in element k_3 and c_3 [N]	F_3	2.769	4.521	64.573	5.431

Table 20: RMS values for the calculated dynamic forces (from measurements)

Forcing frequency		12.125 Hz (≈75 rad/s)		17 Hz (≈106 rad/s)	
Mounting case		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
RMS force in element k_1 [N]	F_{1kRMS}	1176.626	16.951	26162.905	20.968
RMS force in element k_2 [N]	F_{2kRMS}	15.949	5.770	372.537	8.182
RMS force in element k_3 [N]	F_{3kRMS}	1.955	3.191	45.513	3.828
RMS force in element c_1 [N]	F_{1cRMS}	0.000	2.411	0.000	4.181
RMS force in element c_2 [N]	F_{2cRMS}	0.166	0.060	5.434	0.119
RMS force in element c_3 [N]	F_{3cRMS}	0.109	0.179	3.571	0.300
RMS combined force in element k_1 and c_1 [N]	F_{1RMS}	1176.626	17.122	26162.905	21.381
RMS combined force in element k_2 and c_2 [N]	F_{2RMS}	15.950	5.771	372.576	8.182
RMS combined force in element k_3 and c_3 [N]	F_{3RMS}	1.958	3.196	45.653	3.840

9.4.1. Comparison at 12.125 Hz

The predicted values from the two DOF model without damping (75 rad/s) and the experimentally measured data (12.125 Hz) are shown in Table 21.

The mathematical model predicted that there would be a reduction in the amplitude of the force in element 2, which is the plate pack. This was confirmed with the experimental data.

The mathematical model, however, predicted an increase in the amplitude of the forces in element 1. However, this was contradicted by the experimental values indicating that a reduction was achieved in this element.

This variation is due to the neglecting of damping effects in this mathematical model, resulting in higher predicted values close to natural frequencies. The over prediction is conservative in nature and is therefore would not be harmful to the use of this model as a design tool.

The mathematical model predicted that the force in element 3 would increase, which was confirmed by the experimental data.

The experimental values were also compared to the values predicted by the two DOF model that included damping. This comparison is shown in Table 22.

The mathematical model incorrectly predicted that the forces in elements 1 and 2 would increase if the mounting system were changed from the stiff steel-mounted case to the soft rubber-mounted case.

The experimental values show that there is a marked reduction in the dynamic forces in all but one element.

The experimental values prove that there is a significant reduction in the dynamic forces acting in on the plate pack, reducing the force value by almost two thirds to 36.18% of the original force.

Table 21: Comparison between two DOF model without damping and experimental values for a forcing frequency of 12.125 Hz

		Two DOF model without damping		Experimentally determined values	
		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Acceleration of top frame [m/s²]	$\ddot{X}_1$	0.007	21.347	0.030	0.865
Acceleration of bottom frame [m/s²]	$\ddot{X}_2$	7.687	25.593	0.439	0.717
Displacement of top frame [m]	X_1	5.689E-08	-1.705E-04	-5.201E-06	-1.491E-04
Displacement of bottom frame [m]	X_2	5.693E-05	-1.895E-04	-7.570E-05	-1.236E-04
Dynamic force in element k_1 [N]	F_{k1}	18.205	27.426	1664.252	23.976
Dynamic force in element k_2 [N]	F_{k2}	18.198	6.080	22.559	8.162
Dynamic force in element k_3 [N]	F_{k3}	2.079	6.923	2.765	4.514
(F_{k1} for soft rubber-mounted)/ (F_{k1} for stiff steel-mounted)		150.65%		1.44%	
(F_{k2} for soft rubber-mounted)/ (F_{k2} for stiff steel-mounted)		33.41%		36.18%	
(F_{k3} for soft rubber-mounted)/ (F_{k3} for stiff steel-mounted)		332.93%		163.24%	

Table 22: Comparison between two DOF model with damping and experimental values for a forcing frequency of 12.125 Hz

		Two DOF model with damping		Experimentally determined values	
		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Acceleration of top frame [m/s²]	$\ddot{X}_1$	0.007	21.347	0.030	0.865
Acceleration of bottom frame [m/s²]	$\ddot{X}_2$	7.687	25.593	0.439	0.717
RMS force in element k_1 [N]	F_{1kRMS}	12.8753	18.7902	1176.626	16.951
RMS force in element k_2 [N]	F_{2kRMS}	12.8995	79.0172	15.949	5.770
RMS force in element k_3 [N]	F_{3kRMS}	1.4711	4.757	1.955	3.191
RMS force in element c_1 [N]	F_{1cRMS}	0	2.6382	0.000	2.411
RMS force in element c_2 [N]	F_{2cRMS}	0.2138	1.3101	0.166	0.060
RMS force in element c_3 [N]	F_{3cRMS}	0.0812	0.2629	0.109	0.179
RMS combined force in element k_1 and c_1 [N]	F_{1RMS}	12.8753	18.9745	1176.626	17.122
RMS combined force in element k_2 and c_2 [N]	F_{2RMS}	12.9013	79.028	15.950	5.771
RMS combined force in element k_3 and c_3 [N]	F_{3RMS}	1.4733	4.7643	1.958	3.196
(F_1 for soft rubber-mounted)/ (F_1 for stiff steel-mounted)		147.37%		1.46%	
(F_2 for soft rubber-mounted)/ (F_2 for stiff steel-mounted)		612.56%		36.18%	
(F_3 for soft rubber-mounted)/ (F_3 for stiff steel-mounted)		323.38%		163.24%	

9.4.2. Comparison at 17 Hz

The experimentally measured values at 17 Hz (≈106 rad/s) were also compared to the values predicted by the mathematical models.

The predicted values from the two DOF model without damping and the experimentally measured data are shown in Table 23. This model correctly predicted a reduction in the dynamic forces of all the elements.

The experimental values were also compared to the values predicted by the two DOF model with damping. The results are shown in Table 24. This model correctly predicted a reduction in the dynamic forces in all the elements.

The forces inside the plate pack (F_{2RMS}) were experimentally proven to have reduced to only 2.2% of the original force. By only changing the mounting system from a stiff steel-mounted system to a soft rubber-mounted system, a 97.8% reduction in the forces was experimentally measured.

This reduction in dynamic forces indicates an enormous reduction in stresses in the panels and potentially increases the life of the heat exchanger panels significantly.

9.4.3. Conclusion

The experimental measurements indicated that the replacement of the soft rubber mounting of the previous stiff steel-mounted model significantly reduced the forces in the plate pack under exactly the same forcing frequency conditions.

The two DOF model without damping proved to be the most reliable theoretical model in predicting the effect of the change in mounting stiffness. This was especially true in the case of the dynamic forces in the plate pack, where it did not only indicate when the forces would decrease, but was also quite accurate in the prediction of the amount to which the forces were decreased.

Table 23: Comparison between two DOF model without damping and experimental values for a forcing frequency of 17 Hz

		Two DOF model without damping		Experimentally determined values	
		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Acceleration of top frame [m/s^2]	$\ddot{X}_1$	0.073	24.431	1.319	2.104
Acceleration of bottom frame [m/s^2]	$\ddot{X}_2$	78.458	19.006	20.107	1.691
Displacement of top frame [m]	X_1	2.908E-07	9.770E-05	1.156E-04	1.844E-04
Displacement of bottom frame [m]	X_2	2.909E-04	7.046E-05	1.762E-03	1.482E-04
Dynamic force in element k_1 [N]	F_{k1}	93.055	15.714	37005.536	29.657
Dynamic force in element k_2 [N]	F_{k2}	92.982	8.717	526.926	11.572
Dynamic force in element k_3 [N]	F_{k3}	10.625	2.574	64.375	5.415
(F_{k1} for soft rubber-mounted)/ (F_{k1} for stiff steel-mounted)		16.89%		0.08%	
(F_{k2} for soft rubber-mounted)/ (F_{k2} for stiff steel-mounted)		9.37%		2.20%	
(F_{k3} for soft rubber-mounted)/ (F_{k3} for stiff steel-mounted)		24.22%		8.41%	

Table 24: Comparison between two DOF model with damping and experimental values for a forcing frequency of 17 Hz

		Two DOF model with damping		Experimentally determined values	
		Stiff steel-mounted	Soft rubber-mounted	Stiff steel-mounted	Soft rubber-mounted
Acceleration of top frame [m/s^2]	$\ddot{X}_1$	0.030	0.865	1.319	2.104
Acceleration of bottom frame [m/s^2]	$\ddot{X}_2$	0.439	0.717	20.107	1.691
RMS force in element k_1 [N]	F_{1kRMS}	65.6181	11.0777	26162.905	20.968
RMS force in element k_2 [N]	F_{2kRMS}	65.7004	38.0508	372.537	8.182
RMS force in element k_3 [N]	F_{3kRMS}	7.4925	1.8389	45.513	3.828
RMS force in element c_1 [N]	F_{1cRMS}	0	2.1973	0.000	4.181
RMS force in element c_2 [N]	F_{2cRMS}	1.5385	0.8915	5.434	0.119
RMS force in element c_3 [N]	F_{3cRMS}	0.5847	0.1437	3.571	0.300
RMS combined force in element k_1 and c_1 [N]	F_{1RMS}	65.6181	11.2935	26162.905	21.381
RMS combined force in element k_2 and c_2 [N]	F_{2RMS}	65.7184	38.0612	372.576	8.182
RMS combined force in element k_3 and c_3 [N]	F_{3RMS}	7.5153	1.8445	45.653	3.840
(F_1 for soft rubber-mounted)/ (F_1 for stiff steel-mounted)		17.21%		0.08%	
(F_2 for soft rubber-mounted)/ (F_2 for stiff steel-mounted)		57.92%		2.20%	
(F_3 for soft rubber-mounted)/ (F_3 for stiff steel-mounted)		24.54%		8.41%	

The two DOF model that included damping was not as accurate, especially at the lower forcing frequency. The inaccuracy of the predictions could most likely be traced to the difficulty in characterising the damping parameters of the rubber mounts and compensators, due to the frequency and amplitude sensitivity of these values.

This clearly indicates that the forces in the internals of a heat exchanger can be significantly reduced over a number of forcing frequencies by reducing the stiffness of the mounting system.

The two DOF models that are presented provide the opportunity to estimate the effect of a reduction in stiffness beforehand, in order to aid the design of a mounting system.