72AE14

BRITISH ACOUSTICAL SOCIETY

SPRING MEETING: 5th-7th APRIL, '72.

AERODYNAMIC NOISE SOURCES IN INDUSTRY SESSION: University of Loughborough.

NOISE AND VIBRATION DUE TO TUBE VORTEX SHEDDING AND BUNDLE ACOUSTIC RESONANCE IN THE SHELLSIDE OF A TRANSVERSE FLOW HEAT EXCHANGER

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SUMMARY
Heat exchanger noise and, in particular, vibration in recent years have been a source of considerable concern in chemical engineering plant. Recorded are details of the mechanism and solution of a pure tone noise that was generated within a heat exchanger and which resulted in immediate public reaction from distances up to three miles from the factory perimeter. The paper includes a number of theoretical and practical concepts of heat exchanger noise and vibration, as part of the investigation into the cause of this significant practical problem.
It was deduced that the noise was generated by vortex shedding from the tubes, at a frequency which matched an acoustic resonant frequency across the guide plates containing the tube bundle thus resulting in considerable magnification of the noise due to vortex shedding alone. It was concluded that flow induced tube vibration damage was unlikely and that the noise and shell vibration would be reduced by destruction of the acoustic resonance.
Laboratory tests showed that a 0.5 mm thick aluminium sheet would act as a suitable acoustic baffle. Correctly positioned 0.5 mm baffles were fitted in the exchanger and found to be completely effective.

1 INTRODUCTION

During the commissioning of a new plant excessive noise and vibration was observed to be emanating from an exchanger which recovers heat from the gas leaving a power recovery turbine. The gas enters the exchanger,(Fig. 1), through a large gentle taper inlet duct and flows transversely across the tubes and is deflected through 180° and back across the tubes to the outlet duct. The bundle construction is rectangular and the flow is contained within gas guide plates and support plates. The gas is discharged from the exchanger to atmosphere via a 385 ft stack thus providing an extremely effective "transmission aerial" for the noise developed within the exchanger. At local housing areas the noise, assessed using the Noise Rating system corrected for the pure tone nature, had risen considerably: the changes were from +9 to + 24 Noise Rating units and, as may be expected, this provoked strong reaction,(Table 1).
Several tests were carried out on the exchanger and the results were compared with calculated predictions from selected references. This combined experimental and theoretical approach provided an explanation of the mechanism and basis upon which a practical solution could be resolved.

2 EVALUATION OF TESTS AND CORRELATION WITH CALCULATIONS

The exchanger and stack emitted a pure tone noise during the initial stages of the plant commissioning; at the time the exchanger shellside fluid was air. Measurements showed that the frequency of the pure tone noise was at 245 Hz: this is shown in the spectrum analysis, (Fig. 2).

Calculations using Owen[1] and Chen[2] gave a tube vortex shedding frequency of 236 Hz. The calculated tube mechanical vibration natural frequency of 270 Hz suggested that there was a possibility of a resonant tube vibration due to aerodynamic instability. Further calculations showed that the pitch between the gas guide plates was such that an acoustic resonance could be set up at 252 Hz.

Further tests were carried out when the exchanger operated at its normal conditions. The effect of the change to process gas on the shellside was to change the "singing" frequency to 288 Hz: this is shown in the spectrum analysis, (Fig. 3).

Calculation of the vortex shedding and the acoustic resonant frequencies for the process gas conditions gave a similar order of agreement to the above correlation for the air test. This effect only occurred at rates in excess of 80% of design rate and as a temporary measure the plant was "turned down" to avoid the problem.

From the experimental evidence and theoretical analysis, it was concluded that the noise was produced by vortex shedding across the tubes which was being magnified by an acoustic resonance between the gas guide plates. The noise was transmitted to the atmosphere through the stack wall and the stack exhaust thus providing a considerable noise radiation source.

Vortex shedding distribution through the exchanger was expected to be uniform and not location dependent as can be the case in normal multi-pass shell and tube heat exchangers of circular cross-section. This was because:

(a) There were only two passes of the bundle, thus the leakage losses were minimised.

(b) The velocity distribution for any section of flow was very uniform due to the type of bundle and gas guide configuration.

(c) The temperature and pressure distribution of the gas was such that they tended to cancel out the effect of each other on the actual gas volume rate.

Calculations using the data on tube vibration in Lentz[3], Thorngren[4] and Connors[5] indicated that tube vibration damage was unlikely. A physical examination of the bundle revealed no evidence of damage due to buffeting flow or coincidence of vortex shedding and tube mechanical resonant frequencies. Strain gauges fitted to the tube showed the displacements at the mid-span for the acoustic resonant condition to be negligible.

3 CONCLUSIONS

The main conclusions from the tests and calculations are:

(1) the frequency of the noise correlates well with the predicted tube vortex shedding frequencies[1, 2] in the exchanger for the two operating conditions examined in detail, i.e. with air and process gas.

(2) the vortex shedding frequency coincides with an acoustic resonance across the gas guide plates for the two operating conditions and this magnified the pressure waves set up by the vortices shed from the tubes.

(3) the "safe" prediction for the exchanger design using Refs 3, 4 and 5 seemed to confirm the subjective examination and no problem of internal damage due to

flow induced vibration can be envisaged.

(4) to remove the effect it would be necessary to destroy the coincidence between the acoustic resonant and vortex shedding frequencies.

4 ACTION

Having established the mechanism beyond reasonable doubt it was necessary that the match of vortex shedding frequency and acoustic resonance had to be de-tuned by:

(a) changing the vortex shedding frequency, or

(b) removing the acoustic resonance across the gas guide plates.

To effect remedy (a) would mean either considerable modifications to the exchanger or permanently reducing the rate through the exchanger thus reducing the vortex shedding frequency. Both solutions were ruled out because of practical and economic disadvantages but the second measure was used to keep the plant on-line while detailed laboratory investigations on various methods of de-tuning the acoustic resonance were undertaken. The removal of the acoustic resonances is not a complex theoretical task but it did pose some practical problems as to how it could be achieved with minimum disturbance.

Acoustic resonances can be avoided by:

(i) longitudinal baffles pitched to avoid half wavelengths as shown in Fig. 4

(ii) fitting acoustic absorption material in the 'D' section side of the gas guide plates, as shown in Fig. 5.

Method (ii) would have involved considerable exchanger modifications and it was decided that method (i) was the most practical solution. Tests showed that a 0.5 mm thick aluminium sheet was the most suitable baffle material from both the practical implementation and acoustic point of view. Although every effort was made to simulate the actual exchanger arrangement with the test rig it was appreciated that there would be some differences between the test conditions and actual conditions. Because:

(a) the effectiveness of the test rig acoustic baffle could not be fully guaranteed for the full-scale exchanger, and

(b) it was imperative not to cause any further public disturbance,

it was decided to install a stack silencer capable of attenuating sufficient pure tone noise at the exit of the exchanger so as not to cause any further external disturbance. The silencer was designed using data from Mason[6] and King[7] and a number of materials practically suited for the duty were investigated for their absorption properties at 280 H_z. The silencer that was installed is shown in Fig. 6.

5 EPILOGUE

Following the installation of the acoustic baffles and stack silencer the plant was recommissioned and the acoustic baffles were found to be 100% effective. Fig. 7 shows the noise spectra adjacent to the exchanger before and after the modifications. A frequency analysis of the noise after the modifications shows that the vortex shedding frequency noise is still present (as would be expected since the flow pattern has not been affected). However, the noise at the vortex shedding frequency was reduced in magnitude by complete destruction of the acoustic resonance.

Subjective consideration of the vibration level in the vicinity of the exchanger has shown that there has been considerable improvement and the level is no greater than on any of the other transverse flow heat exchangers on the plant. The effect of the modifications on the noise near to the exchanger and stack is shown in Fig. 8: this noise spectrum is part of a full noise survey carried out in and around the plant. Fig. 8 shows clearly the effect of the acoustic baffles on the 250 Hz mid-octave component and shows that the stack silencer is not totally redundant since it is effectively attenuating the high frequency machine noise being passed to the vent stack via the heat exchanger. Noise surveys outside the factory perimeter have shown that the levels are restored to the previously established background levels.

Successful solution of this serious noise problem was achieved by detailed and logical plant experimental and laboratory work combined with a theoretical investigation into the problem.

TABLE 1 - NOISE RATINGS AROUND WORKS PERIMETER

Location	Position A	Position B	Position C
Normal Accepted Background Rating	41	46	55
Vessel "singing" (uncorrected for Pure Tone)	55	45	58
Vessel "singing" (corrected for Pure Tone)	65	55	68

ACKNOWLEDGEMENTS

The authors wish to thank the Directors of ICI Agricultural Division for permission to publish this paper.

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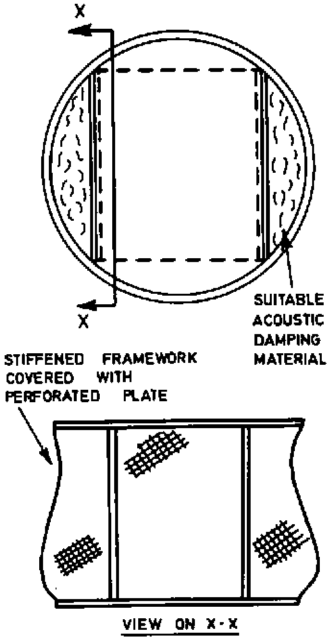


FIG. 5 ARRANGEMENT FOR ACOUSTIC DAMPING

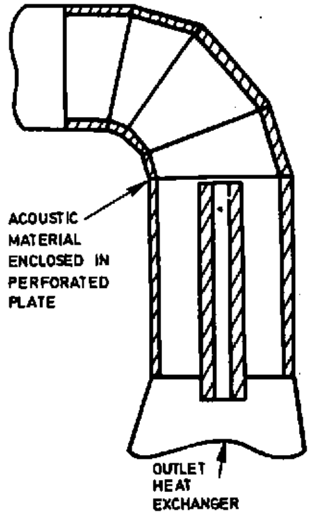


FIG. 6 STACK SILENCER

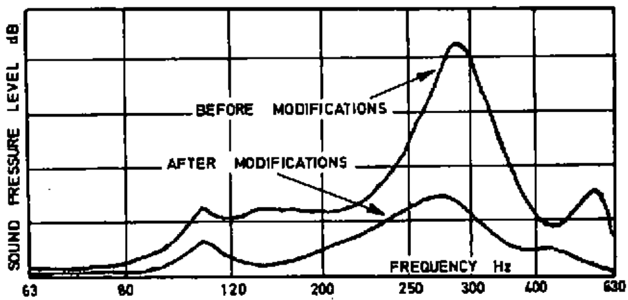


FIG. 7 NOISE SPECTRA THROUGH SHELL WALL OF EXCHANGER BEFORE AND AFTER MODIFICATIONS

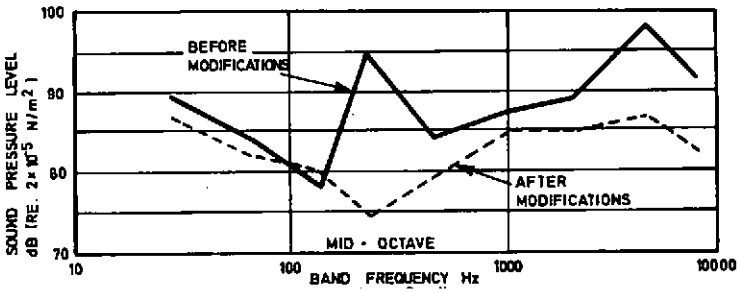


FIG. 8 PLANT NOISE SPECTRUM, BEFORE AND AFTER MODIFICATIONS TO THE EXCHANGER, FITTING THE SILENCER AND PLANT LAGGING

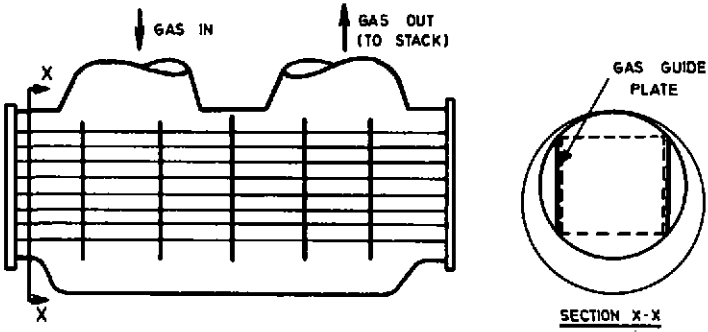


FIG. 1 BUNDLE CONSTRUCTION

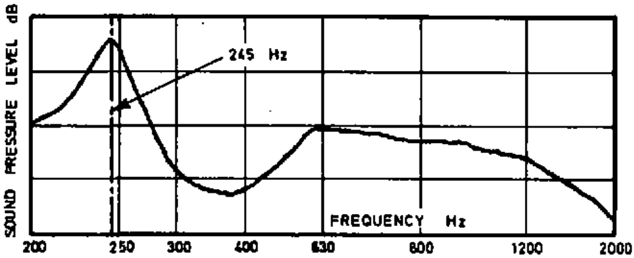


FIG. 2 NOISE SPECTRUM :- AIR ON SHELL SIDE

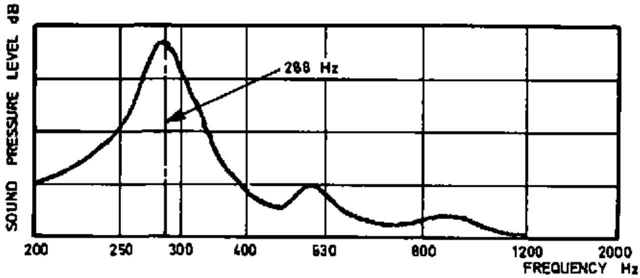
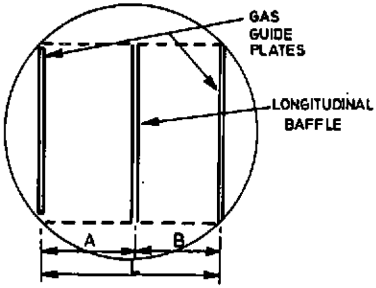


FIG. 3 NOISE SPECTRUM :- PROCESS GAS ON SHELL SIDE



TO AVOID 'ACOUSTIC RESONANCE'

A OR $B \neq \frac{N\lambda}{2}$

λ = WAVELENGTH OF PREDOMINANT FREQUENCY NOISE

N = ANY INTEBER

FIG. 4 LOCATION OF LONGITUDINAL BAFFLE