1996

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International Compressor Engineering Conference. Paper 1200.  
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CHARACTERISATION OF COMPRESSOR NOISE
PROPAGATING THROUGH CONNECTED PIPES

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ABSTRACT

Fundamental features of fluid-borne and structure-borne noise propagation through compressor pipes are discussed. The noise level is shown to depend on the observed position. One possible method of noise characterisation in compressor pipes, developed at CETIM, is presented. The method uses a specific test circuit with anechoic properties to ensure uniqueness of results, and the Intensity Method adapted to pipes for reproducible measurement of noise emission.

INTRODUCTION

Noise emitted by compressors propagates to the surroundings in three different ways:
- as air-borne noise directly to the ambient space
- as structure-borne and fluid-borne noise through the connected pipes
- as structure-borne noise through the compressor feet

The relative contributions of these different noise paths to the total noise level depend on the type of machine considered. Neither of the three paths should be a priori neglected. However, noise propagation through the pipes is of specific significance as noise can be transmitted far away from the source due to weak spatial attenuation in pipes.

Noise emitted from a source depends on acoustic properties of the surrounding space. In particular, noise propagating from the compressor through the connected pipes depends on the impedance of the piping circuit. In order to characterise compressors as a noise source in an unambiguous way, the external acoustic conditions have to be uniquely defined, and later controlled during the characterisation process.

This paper describes a procedure for characterisation of compressor noise propagating through the connected pipes which has been conceived and developed at CETIM. The procedure comprises both a specific test circuit and a specific measurement method for noise characterisation. The former, having well controlled anechoic properties, is necessary to ensure uniqueness of noise emission conditions which otherwise depend on the properties of the piping circuit. The latter, based on Intensity Method adapted to fluid-filled pipes, ensures the reproducibility of results irrespective of the chosen measurement location.
BASIC ACOUSTICAL FEATURES OF FLUID-FILLED PIPES

Noise propagates through a pipe in the form of pressure pulsations and structure-borne vibrations. The pipe is in the acoustical sense a waveguide with elastic walls. All the characteristic features of acoustical waveguides apply therefore to pipes. Some of these are listed below:(1)

- Different types of acoustical waves simultaneously take place. The proportion of each particular wave in the total acoustical motion depends on the distribution and frequency of pipe excitation,
- At lower frequencies, the majority of the wave types are evanescent, i.e. decaying exponentially from the pipe terminations. As the frequency increases, the number of wave types which propagate without any appreciable decay is rising,
- Due to the elasticity of the pipe wall, a coupling exists between the wall and the contained fluid which induces the transfer of the acoustical energy from the wall to the fluid and vice versa.

The acoustical phenomena in pipes are most usually related to so called normalised frequency \( \Omega \) which is a function of frequency \( f \) [Hz], mean pipe diameter \( d \) [m] and speed of compressional waves in the pipe material \( c_s \) [ms\(^{-1}\)].

\[
\Omega = \pi fd / c_s
\]

In practical cases only low values of \( \Omega \) will be of interest. In these cases fairly simple laws apply(2):

- Four wave types only can effectively take place: fluid, longitudinal, torsional and flexural waves. Out of these four wave types, only flexural waves can have a strong evanescent component, while the other three wave types are essentially purely propagative,
- A light fluid, such as gas, will not strongly couple to the pipe wall. The coupling will increase with frequency. The speed of sound in a light fluid will be only marginally affected by the wall elasticity.
- The energy of a particular wave type will be almost completely contained either in the fluid (for fluid waves) or in the wall (for longitudinal, torsional and flexural waves). This makes it possible to consider these wave types as independent from each other.

This simple behaviour of pipes extends up to so called first "cut-on" frequency, where other, more complex wave types start propagating energy. The cut-on frequency depends on both the pipe and fluid properties, in particular on the thickness-to-diameter ratio. Figure 1 shows the cut-on frequency for a copper pipe containing Forane 22 at 20 bar and 70°C. It can be seen that this limiting frequency is above 1 kHz for small pipes up to 40 mm in diameter providing the pipe thickness is above 1 mm.

Figure 1: First cut-on frequency for a copper pipe containing Forane 22: 20 bar, 70°C
One of the most important quantities which affects noise propagation in pipes is the speed of each particular wave type. At not too high frequencies, the wave speed is independent of frequency for all but flexural waves. The latter propagate at the speed which depends much on frequency. Figure 2 shows the normalised speed of flexural waves (actual speed divided by the speed of compressional waves) in dependence of normalised frequency for some typical values of thickness/diameter ratio. The pipe parameters are the same as in the previous example, i.e. copper/Forane 22 at 20 bar and 70°C.

Figure 2: Normalised speed of flexural waves for a copper pipe containing Forane 22: 20 bar, 70°C

The wave speed of fluid and longitudinal waves, independent of frequency at lower \( \Omega \), depends still on pipe geometry and material. For light fluids this dependence is quite negligible.

FLUID-BORNE AND STRUCTURE-BORNE NOISE LEVELS ALONG PIPE

Compressor noise is essentially of periodic nature. The noise spectrum thus consists of a number of harmonics. Each harmonic component is contributed by waves propagating in opposite directions of the pipe axis. As a result interference takes place, producing considerable variations of the noise level along the pipe. Variation of level occurs even at a single frequency.

Figure 3 shows the spread of RMS level of fluid pulsations for a single harmonic in dependence of the reflection coefficient at the pipe termination which is not excited. The value 1 corresponds to the mean RMS level. The case 100% reflection corresponds to a fully standing wave where the maxima and minima in the RMS level assume extreme values of 1.41 and 0 respectively. At 50% reflectivity, which can be considered as a typical value, the variations in level are between 1.34 and 0.45, i.e. almost 10 dB. This means that the measurement of pipe noise should not be made in one point only, as the result obtained may be far from the mean value.
The RMS level distribution for a single harmonic is spatially repetitive, with the periodicity of half the wavelength. When several harmonics exist, the repeatability in total RMS level does not hold any more.

It follows that the variations in the noise level can be suppressed by reducing the reflections from the pipe termination. Under normal circumstances, the reflection will exist always as different components in the piping circuit, valves, mufflers etc., necessarily revert one portion of the incoming acoustical energy. The overall reflectivity as seen by the discharge or suction pipe of the compressor will therefore depend on the particular piping circuit connected to these pipes. This means that the same compressor, connected to two different piping systems, will emit different noise levels in the two cases. Such a feature is not acceptable for the characterisation of compressor pipe noise as it would give rise to ambiguous results.

OPTIMISED CHARACTERISATION PROCEDURE

The characterisation procedure developed at CETIM compensates the deficiencies of the classical measurement methods. It simultaneously takes into account the need for

• uniqueness of characterisation conditions
• independence of the measurement position on results

Figure 4 depicts the basic measurement set-up. The discharge and intake pipes are fitted with so called TAG's, which is the French abbreviation for the anechoic gas termination. The measurement sections are fitted with transducers for fluid-borne and structure-borne intensity measurement.

Figure 4: Measurement set-up for characterisation of compressor pipe noise

A TAG is an acoustical device, the purpose of which is to reduce reflections of pressure pulsations. It consists essentially of a straight, perforated, long pipe inserted into a cavity shield. The diameter of the perforation holes, which act as Helmholtz orifices, continuously increases along the pipe length in order to produce a gradual alteration of the impedance.

Figure 5: Cross section of a TAG
This is essential for achieving low reflectivity. The outer shield, acting as the resonator volume, is divided length-wise into a number of smaller compartments for an optimum acoustic performance. Optimisation of an anechoic termination has to be done for each fluid and for each pipe diameter$^3$.

The pipe intensity, usually measured in mW, has the meaning of the rate of change of the acoustical energy along the pipe (energy flow). The acoustical energy flow along a pipe is practically constant: in the absence of internal acoustical sources within the pipe, the energy flow will only slightly diminish due to fluid and structural damping which is usually negligible. Possible turbulence noise, originating from the pipe, will not normally interfere with the compressor harmonics. The result of energy flow measurement thus becomes independent of the measurement position. Next advantage comes from the fact that the energy contributions of different wave types are simply added for obtaining the total energy flow.

The pipe intensity is a novel concept, requiring expertise in fluid-borne and structure-borne acoustics. An analogous concept, that of the acoustical air-borne intensity has been in use for almost two decades, backed up with commercially available instrumentation - sound intensity meters. So far, four international congresses on intensity techniques, the last two of which dealt exclusively with its structure-borne aspect, have revealed a considerable interest of the acoustical community for the subject$^4$. In spite of the interest shown, industrial applications of intensity techniques other than air-borne ones remain virtually non-existent due to complexity of the phenomena and lack of an appropriate equipment.

The fluid-borne intensity measurements are done using piezoelectric pressure transducers. If the wave speed of the fluid is known, two transducers suffice for measurement, otherwise a third one is necessary. The structure-borne intensity measurements are done as a rule using an array of miniature accelerometers. For basic measurements four accelerometers are needed for an engineering-grade estimation of structure-borne intensity. More accurate measurements require the simultaneous use of eight accelerometers. Once the signal acquisition is done, the post-processing for the determination of either the fluid-borne or structure-borne intensity is carried out on the basis of theoretical modelling of pipe pressure pulsations and wall vibrations. This procedure is simple when fluid-borne case is concerned, but fairly complex in the structure-borne case where separation of different wave types is needed.

Table I shows the acoustical performance of a new-generation TAG for refrigeration gases, which has been developed recently at CETIM. The reflection coefficient is well below 20% throughout the frequency range up to 2 kHz. This feature guarantees low pressure variation of the order of 3 dB. Even so, the reproducibility of measurement results is increased by using intensity instead of level measurement. Here the TAG serves predominantly for ensure some well specified measurement conditions, but it also improves the measurement reproducibility which increases with the reflectivity decreasing.

### Table I: Reflection coefficient of a TAG optimised for refrigeration gases

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>50 Hz</th>
<th>100 Hz</th>
<th>200 Hz</th>
<th>500 Hz</th>
<th>1000 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>17%</td>
<td>18%</td>
<td>2%</td>
<td>10%</td>
<td>9%</td>
<td></td>
</tr>
</tbody>
</table>

It has been found that all three types of structure-borne waves, longitudinal, torsional and flexural, can be of similar importance in transmitting the energy of structure-borne noise. Measurements on small compressors have revealed that the fluid-borne intensity normally exceeds the structure-borne one. This can be seen on Figures 6 and 7, showing the intensity of fluid waves and flexural structure-borne waves obtained on the discharge and suction pipes of a small refrigeration compressor.
CONCLUSIONS

Characterisation pipe noise in compressors needs very careful analysis of noise propagation in pipes before any practical measurement method is set up. A novel method, using the anechoic terminations for pipes and intensity techniques yields unbiased and reproducible characterisation results.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the support of this work by the Working Group "Refrigeration Machinery" of the Association of French Mechanical Industries.

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