Sound propagation on a high pressure gas pipe

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At a two stage high pressure gas compressor with intermediate cooler, the cooler radiates a tone at the compressor rotational frequency of approx. 160 Hz. For the design of noise reduction devices, the mechanism of sound propagation in between the compressor and the cooler on a DN 400 steel pipe with 24 mm wall thickness and 160 bar internal gas pressure was to be determined.
By non-invasive vibration measurements on the pipe, the dominance of the fluid borne sound (natural gas) over the structure borne sound was found.
Later, pressure measurements in the pipe clearly approved this result.

1 Introduction

A large compressor station radiates an audible low frequency tone at approx. 160 Hz into its (very quiet) neighbourhood. To develop noise reduction measures for this plant, it is necessary to have a good understanding of the sound transmission within the installation. The non-invasive determination of the vibrational and fluid-based power flow through a fluid-filled pipe shows up to be an important task. This measurement including the data analysis procedure is described in this paper.

2 Some facts on the compressor plant

The tested plant is a two stage high pressure gas turbo-compressor with a rotational frequency of 160 – 170 Hz. On the low-pressure side natural gas is fed into the compressor with a long-distance pipeline pressure of approx. 55 bar where it is compressed to a medium pressure of approx. 160 bar. Then the heated gas is piped via the medium pressure line to a big air cooler, subsequently, after cooling-down, the gas is compressed up to 330 bar. The electrical driving power is 38 MW and the transport capacity 12 Mio. Norm-m³/day. Strong vibrations with the rotational frequency and its harmonics as well as broad-band flow excitations can be measured on the connecting pipelines. There are no details as for the excitation mechanism, a fluid excitation however is not typical for rotational frequencies with turbo-compressors. It's for noise protection reasons that the compressor is housed in a separate power house, furthermore the compressor's bedplate is grounded separately elastically decoupled from the building's fundament etc. Starting at the job definition to reduce the sound emission at rotational frequency the question can be deducted how to reduce the excitation of the emitting coolers. For this purpose, first the excitation mechanism of the coolers has to be found. Obviously excitation can only propagate via the pipelines (and there apparently predominantly via the medium pressure line). Still the exact dominating propagation path has to be found what for it requires to determine the energy flow for the several propagation paths resp. modes (structure borne vibration modes on the pipe wall, gas-inherent fluid borne sound).

3 Sound propagation on an in-vacuo pipe

It is not possible to give here a complete description of sound propagation on an in-vacuo pipe. Reference may be made to relevant literature.

It is appropriate to model the pipe by thin shell theory. Via the equations of motion this leads to dispersion relations as a polynomial of degree 8 in wave number for each circumferential mode number *m*. Indeed most of the modes/wave forms show lower cut-on frequencies for their propagation. In the present case we can restrict ourselves to low frequencies. The following modes/wave forms are always able to propagate on the pipe wall:

- Torsional wave (m=0 transverse wave)
- m=0 (quasi)-bending wave (partly described as dilatational wave, yet not to be mixed with the longitudinal wave)
- m=1 bending wave in 2 polarization directions, both plus near field.

4 Sound propagation on a fluid-filled pipe

Also for a fluid-filled pipe only summarizing statements can be made here. For details reference is made to relevant literature.

Basically, with fluid-filled pipes, the fluid-structure-interaction has to be considered. Therefore the fluid loading has to be added to the equation of motion of the thin shell theory. In the dispersion relation, for each circumferential mode an infinite number of radial modes in the fluid occur additionally, which can be considered theoretically by an appropriate series expansion.

For low frequencies the following propagable modes/wave types result:

- Torsional wave, not coupling with the fluid
- m=0 (quasi)-bending wave, strongly coupling with the fluid
- m=1 bending waves with near fields, are cross-sectional preserving and therefore weakly coupling with the fluid
- m=0 plane fluid wave, coupling strongly with the pipe wall.

5 Relevant pipe data

The medium pressure line has a nominal diameter of DN 400 with a wall thickness of 24 mm and is made of steel. Inside of it there is natural gas with a pressure of

160 bar and a temperature of 383 K, which results in a density of 93,7 kg/m³ and a speed of sound in the infinite fluid of 481 m/s.

Considering the fluid-structure-coupling by the pipe wall's compliance there will be virtually no change in the acoustical velocity ($c_{F'}/c_F$ = 0,9992), thus the pipe is rigid for the gas.

The cut-on frequency for the propagation of the first higher gas mode inside the pipe is at 787 Hz, the cut-on frequency for the m=2 bending mode for the vacuo-pipe at 392 Hz and for the fluid-filled pipe at 382 Hz.

It is not necessary to consider higher modes for this task defined to 160 Hz.

6 Measurements

Comprehensive vibration measurements have been executed on the medium pressure line. Figure 1 shows the set-up used with 20 tri-axial accelerometers in 5 rings at 90°-positions each. The raw time series of these 60 channels (as well as further additional channels) were recorded synchronously using a PAK MKII measuring system, the subsequent reports and standard evaluations were also done with this system. Only for the final evaluation procedures (separation of waves, energy flow calculation) data was exported and calculations carried out in Matlab.

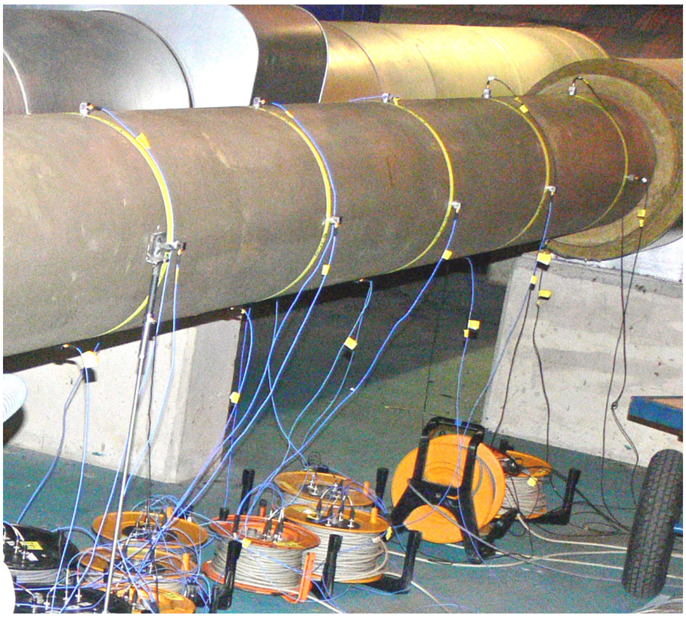

Figure 1: Measuring set-up on the gas pipeline: 5 rings with 4 tri-axial accelerometers each in 90°-positions.

7 Analyses

First the acceleration time series were transformed into the frequency domain for each sensor. Based on complex spectra the modal decompositions of the circumferential modes (circumferential mode order, axial/radial/tangential acceleration) were carried out for each sensor ring. Again based on these (also complex) amplitudes of the single directions of motion per circumferential mode waves were separated into m=0 quasi-bending waves running to and fro and plane fluid waves as well as m=1 bending waves running to and fro with near fields (separated for both polarisations) based on theoretical wave numbers.

The resulting modal amplitudes then can be converted by using appropriate modal impedances into power flows of the single modes/wave types and finally get balanced.

8 Results

The following power flows averaged for typical operating conditions result from the analyses (with positive power flow from the compressor to the cooler and negative power flow from the cooler to the compressor):

- Torsion -0,05 W
- m=1 bending wave, Pol. A 0,05 W
- Pol. B 0,49 W
- m=0 (quasi-)bending wave -0,94 W
- plane fluid wave 7,23 W
- Total sum 6,78 W
- (Thereof: radiated sound power) 0,04 W)

The power flows clearly give evidence that in this case the main power transport is done by fluid borne sound from the compressor to the cooler. This result was backed impressively by a verifying pressure measurement carried out later (on one measuring point only).

Thus the applied method of indirect power flow determination can be considered as verified and applicable in the field including the fluid-borne part of exclusive measurements of accelerations on the pipe wall.

References

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