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ACOUSTIC COUPLING OF A CENTRIFUGAL COMPRESSOR

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ABSTRACT
During the tests in similitude conditions at low pressure and low load, a centrifugal compressor exhibited, at low flow rate, unusual very low frequency vibrations. An acoustic study of the piping system of the test loop was performed by TECHNICATOME/SMVI with the LACTUS® software (a computer-aided design tool for noise-free and reliable piping networks). It was found that the acoustic resonances were in accordance with the vibration frequencies measured on the compressor, and that the levels of the vibration peaks were proportional to the acoustic transfer function amplitudes. The absence of vibrations during the full load tests confirmed the assumption of the acoustic coupling between the compressor and the piping system of the test loop whose acoustic resonances amplified the pressure pulsations and forced the low frequency vibration of the rotor.

1 - INTRODUCTION

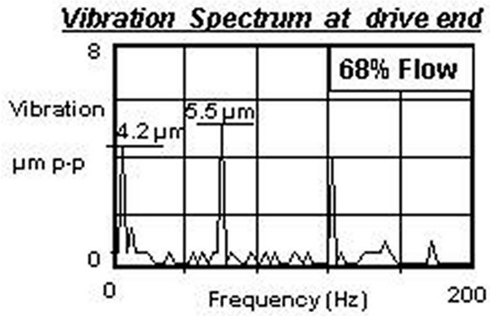
The centrifugal compressors for the oil and gas industries, designed according to API 617 standard, are submitted to acceptance tests in similitude conditions with an inert and pure gas in order to measure accurately their aerodynamic performance. For this acceptance test, the compressor is installed in a special test rig with a test loop including a pressure reducing valve and a cooler in a piping system of a very limited size. Because the sound velocity is lower and the pipe length is smaller than that of the site, the first acoustic natural frequencies are not negligible regarding rotational speed, and acoustic-aerodynamic couplings may induce resonances with significant stationary pressure pulsations.

2 - COMPRESSOR AND PERFORMANCE TESTS

The multistage centrifugal compressor is made of five impellers in one section. The aerodynamical performances of the centrifugal compressors are usually checked during shop tests. These tests are generally carried out at a limited pressure, in order to limit the required power and the heat evacuation. A pure gas, very well thermodynamically characterised, is also used for the tests. API 617 and ASME PTC 10 standards define the similitude conditions, in order to respect the ratio of the suction volume flow to the rotational speed, and the ratio of the suction and discharge volume flows. According to the selected gas test, the pressure and the rotational of the test are derived. Sometimes, another shop test is also performed at nominal pressure, nominal speed, nominal power, and with a gas of the same molecular weight as the contract gas. The purpose of this full load test is to check the behaviour of the compressor placed in its future actual operating conditions, for the sealing system, the vibratory stability of the rotor, the absence of aerodynamical instabilities, and of the complete train with its control and monitoring system. The test loop used for the full load test is different from the loop of similitude tests.

3 - SIMILITUDE TEST SUMMARY

The test loop includes the compressor driven by a variable speed electric motor through a gear, one or two coolers in parallel to eliminate the heat generated by the compressor, throttle valves to reduce the pressure level from the discharge to the suction conditions. A piping system including the temperature, pressure and flow elements for the acquisition of the aerodynamic performance measurements, connects all these components. At nominal flow, the vibration spectrum is very simple, only peaks at the rotational frequency and its first harmonics appear. When reducing the flow rate at constant speed, a small emergence at low frequency begins at 77 % nominal flow, whose level increases when the flow decreases (Fig. 1):



Vibration level (μm p-p) versus flow rate			
Flow rate	@ 3.5 Hz	@ 5.3 Hz	@ 7.1 Hz
80 %	< 0.1	< 0.1	< 0.1
77 %	0.77	< 0.1	< 0.1
73 %	1.9	< 0.1	< 0.1
70 %	2.5	0.6	0.35
68 %	4.2	0.7	1.2

Figure 1: Vibration spectrum at various flows.

The subsynchronous vibration level, transposed to the site conditions by a rotordynamic simulation, was not allowed for a safe vibratory operation for flows lower than 74 %.

3.1 - Modification of the similitude test loop

The test loop includes two coolers in parallel, and the possibility to throttle the flow either at the compressor discharge, or just downstream the coolers. In order to investigate the phenomenon which caused the subsynchronous vibration, the test conditions were changed, with three modifications of the loop arrangement:

- throttling the compressor discharge and using one cooler (operation for low flow rate only)
- throttling the cooler outlet (operation at lower pressure due to the designed pressure of the cooler installed at the discharge pressure of the compressor).

The levels of the subsynchronous vibrations were found dependent from the arrangement of the test loop for the same flow rate (75% nominal flow) − they were worth respectively 4.2 μm, 0.65 μm and 0.3 μm. Then, it clearly appeared that the frequency and the level of the vibration was not due to the compressor alone. This dependence of the vibration level upon the test loop arrangement might be an indication that the pressure pulsations of the compressor are amplified by the acoustic resonances of the test loop (the compressor being a source of broad band excitation).

3.2 - Acoustic study of the test loop

An acoustic study of the piping system of the test loop was performed with the LACTUS® software for the three arrangements described. LACTUS® software analyses piping networks' behaviour by simulating pressure and velocity fluctuations in the network and its surroundings. A piping network is assumed to be broken into a group of components (pumps, valves, pipes,...) which act as acoustic sources and/or wave guides. In addition to these components, there are "end" or "termination" type components which impose certain boundary conditions such as acoustic impedance. The analysis is performed in the low frequency domain corresponding to plane waves hypothesis. The acoustic state at the ports of each component is then described completely by two variables: the acoustic pressure p and the volume velocity q or the impedance $\boldsymbol{Z}=\boldsymbol{p}/\boldsymbol{q}$. If a component shows an acoustic coupling between the input and output ports it should be described as an acoustic two-port ([1]) characterised, at each frequency, by a passive 2 ×2 transfer matrix M, that describes the transmission of pressure waves through the component, and two active source variables, a source pressure p_s and a volume velocity q_s:

$$\begin{pmatrix} p_{output} \\ q_{output} \end{pmatrix} = M \begin{pmatrix} p_{input} \\ q_{input} \end{pmatrix} + \begin{pmatrix} p_s \\ q_s \end{pmatrix}$$

In order to model the test loop, several hypothesis were made:

- as the desired results consisted on the dynamic behaviour of the test loop (modulus of the transfer functions only), the hypothesis that is made upon the nature of the acoustic source is irrelevant; therefore the compressor was considered a unitary velocity source (p_s=0, q_s=1 m^3/s);

- the transfer matrix of the compressor was assumed to be the identity matrix ([2], [3]), considering that the impedance of the compressor is negligible ($M{=}I$);
- the pressure reducing valve with sonic flow was modelled by a "velocity node" boundary condition;
- air and water coolers were replaced by straight pipes, widenings and narrowings, the bundle of pipes being replaced by a single pipe of equivalent thickness and diameter;
- the pipe walls material was considered rigid in the frequency band used ($0-10$ Hz), and because of the weak coupling between the structure and the internal fluid (low density).

The acoustic transfer function of the loop was determined by calculating the term $H_i(\omega) = p_i(\omega)/q_s(\omega)$ in the internal fluid, with the LACTUS® software.

3.3 - Comparison between measurements and simulation
It was found that the three low frequencies found in the vibration spectra (3.5 Hz, 5.3 Hz and 7.1 Hz) are also in accordance with the acoustic resonances of the piping system. The vibration level at the frequency of 3.5 Hz and at 75 % flow, was quasi proportional to the magnitude of the transfer function for the same resonance. This quasi proportionality between acoustic resonances and vibration levels among the three configurations brings the following statements:

- the throttling of the valves doesn't influence the modal response of the loop at that frequency,
- in this case, the vibration levels are directly amplified by the acoustic resonance of the test loop.

It was concluded from those results that the compressor alone is not responsible for the vibration levels measured which are caused by the amplification by the test loop of vibrations coming from the compressor.

4 - FULL LOAD TEST

According to the contract, this unit was full load tested (ASME PTC 10 Class 1) on a specific test bench. Because the high generated heat had to be evacuated, several coolers of the test bench were used (4 water coolers and 3 air coolers). An acoustic study of the piping system of the test loop was performed with the LACTUS® software. The acoustic hypothesis taken here were identical to those taken in the similitude test case. The acoustic transfer function calculated with LACTUS® showed that the most energetic resonances are much lower in frequency than the fluctuations possible from the compressor (7 % rotational speed), and that there is no risk of acoustic coupling. The behaviour of the compressor was free of subsynchronous vibration down to 68 %, very much lower than the 74 % found in the similitude tests. The absence of vibrations during the full load tests confirmed the assumption of the acoustic coupling between the compressor and the piping system of the similitude test loop whose acoustic resonances amplified the pressure pulsations and forced the low frequency vibration of the rotor.

5 - CONCLUSION

The compressor exhibited subsynchronous vibrations during the similitude tests. The level and the frequency of the vibration were dependent on the arrangement of the test loop, and in a rather good accordance with the simulation of the acoustic transfer functions by LACTUS®. During the full load tests, the subsynchronous vibrations began at a lower flow rate, and at a conventional frequency for the compressor. This means that the aerodynamical excitation generated by the compressor during its operation at low flow rates, but before the development of any instability, were amplified by the acoustic resonances of the test loop, and caused the compressor to vibrate. The modelling of pulsation and vibration phenomena becomes essential for the success of the design of machines and piping networks. The LACTUS® software has proven to be an efficient and effective tool for the prediction and analysis of dynamic phenomena in piping networks. In particular, the LACTUS® software can help predicting acoustic resonances in piping network with good accuracy, therefore enabling the network builder to avoid coincidences between acoustic modes of the network and peak frequencies of the external devices (pumps, fans,...).

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