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107. Influence of turbocompressor working mode on vibroactivity

V. Barzdaitis[1], R. Jonušas[1], E. Juzėnas[1], V. Volkovas[1], V. Mureika[2]
[1] Kaunas University of Technology, Lithuania
[2] "Achema" Ltd, Lithuania
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Abstract

This paper examines variation of turbocompressor vibroactivity under varying conditions. The results of compressor vibromonitoring and experiments made under different compressor working regimes are presented.

Keywords: turbocompressor, vibrations

1. Subject of Research

When exploiting various compressor, air blovers and similar devices, dynamics of the compressed air stream has a great effect on their vibroactivity. The infringed air steam, the increased resistance in compression or suction ducts of the compressor can frequently cause high level vibrations which might be unnoticed under the normal operation of the compressor.

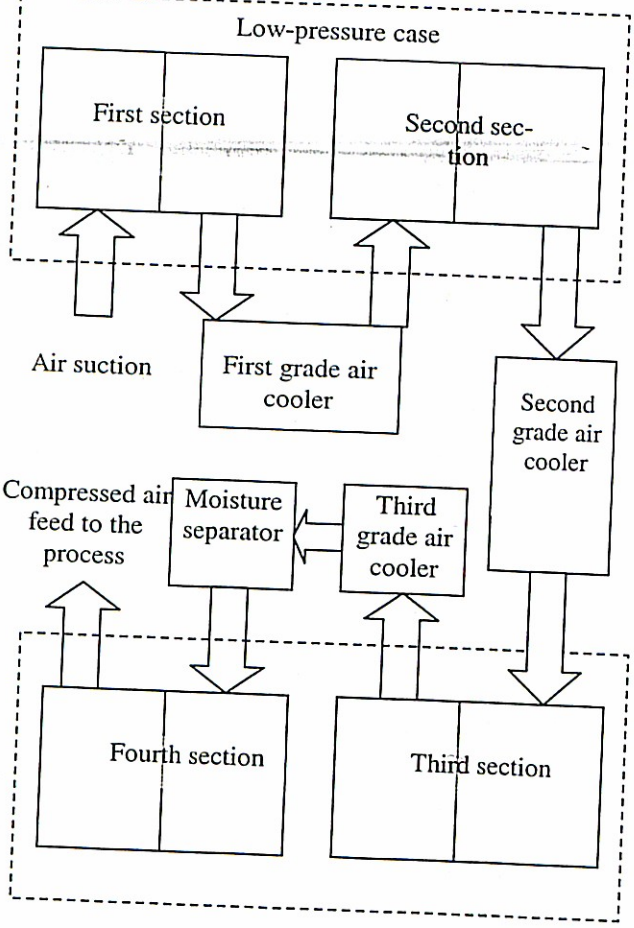


Fig. 1. Diagram of air streams in turbocompressor

When vibromonitoring the air turbocompressor K1290-212-1 in "Achema" Ltd ammonia plant, its raised vibroactivity was observed. That turbocompressor consists of two low and high pressure cases and has 4 compression sections. The compressed air flows from one section to the other though intermediate coolers in which it is cooled by the water stream (Fig. 1). After the first compression section the air pressure is about 0,13 MPa, after the second - about 0,43 MPa, after the third – about 0,95 MPa, after the fourth – about 3,2 MPa, the yield being 65000 m^3/h. The given diagram indicates that the system comprises the air suction and compression collectors with their own tanks causing resistance to the air stream flow, several devices, namely, the first, second, and third grade air coolers and a moisture separator with their own tanks also causing the resistance to the air stream flow [1]. In addition, when examining the forced vibrations of that compressor, both the tanks of the separate parts of the compressor case and the pressure changes among separate parts should also be evaluated [2]. The examined system is very complicated and, practically, the possible pumping modes cannot be determined because of the resistance variation to the air stream at any of the parts.

The rotor of the turbocompressor low-pressure case is rotated by a steam turbine at 3250 – 3350 r/min speed, then the motion is transmitted by a multiplicator to the high pressure case rotor rotating at about 9250 – 9550 r/min.

2. Results of Vibration Measurement

When measuring the vibrations of the compressor rotors supports it was determined that vibration level in the high-pressure case was significantly raised when the free end of rotor was measured in the vertical direction. Fig. 2 presents the variation of vibrations level in the final bearing of the high-pressure case.

As seen from Fig. 2 the vibration level of the high-pressure case varies from 1 mm/s to 8 mm/s and this variation does not coincide with that of turbine rotation frequency. The spectrum analysis of the signal has revealed that largest component is 48.44 Hz when the rotor revolution frequency is 154.7 Hz, thus making about 30% (Fig.3.). Longer observation of the compressor vibroactivity made it possible to determine that the frequency of that component remains constant when the frequency of rotor revolution changes. Therefore, it can be concluded that the main vibrations source may be the vibrations of the compressor case or the rotor excited either by the air stream or the lubricants of sliding bearings [3, 4]. This analysis has resulted in determination of the following feasible vibrations sources:

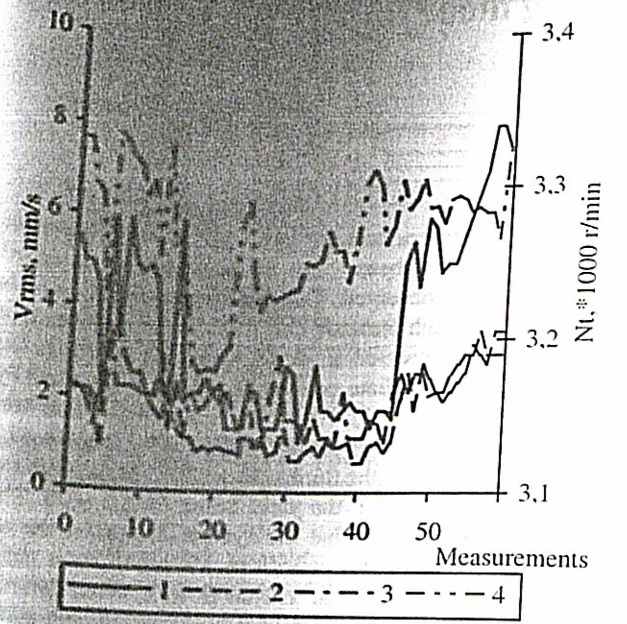


Fig. 2. Results of measuring the 10th bearing vibrations in the high pressure case of the turbocompressor: 1 – vibrations level of the case when measuring in the vertical direction; 2 – vibrations level of the case when measuring in the horizontal direction; 3 – vibration level of the case when measuring in the axial direction; 4 – rotation frequency of the turbine rotating the turbocompressor.

1) vibrations excited by the airflow in the suction collectors of the compressor;

2) air pressure variation in the compression collectors of the separate compressor sections;

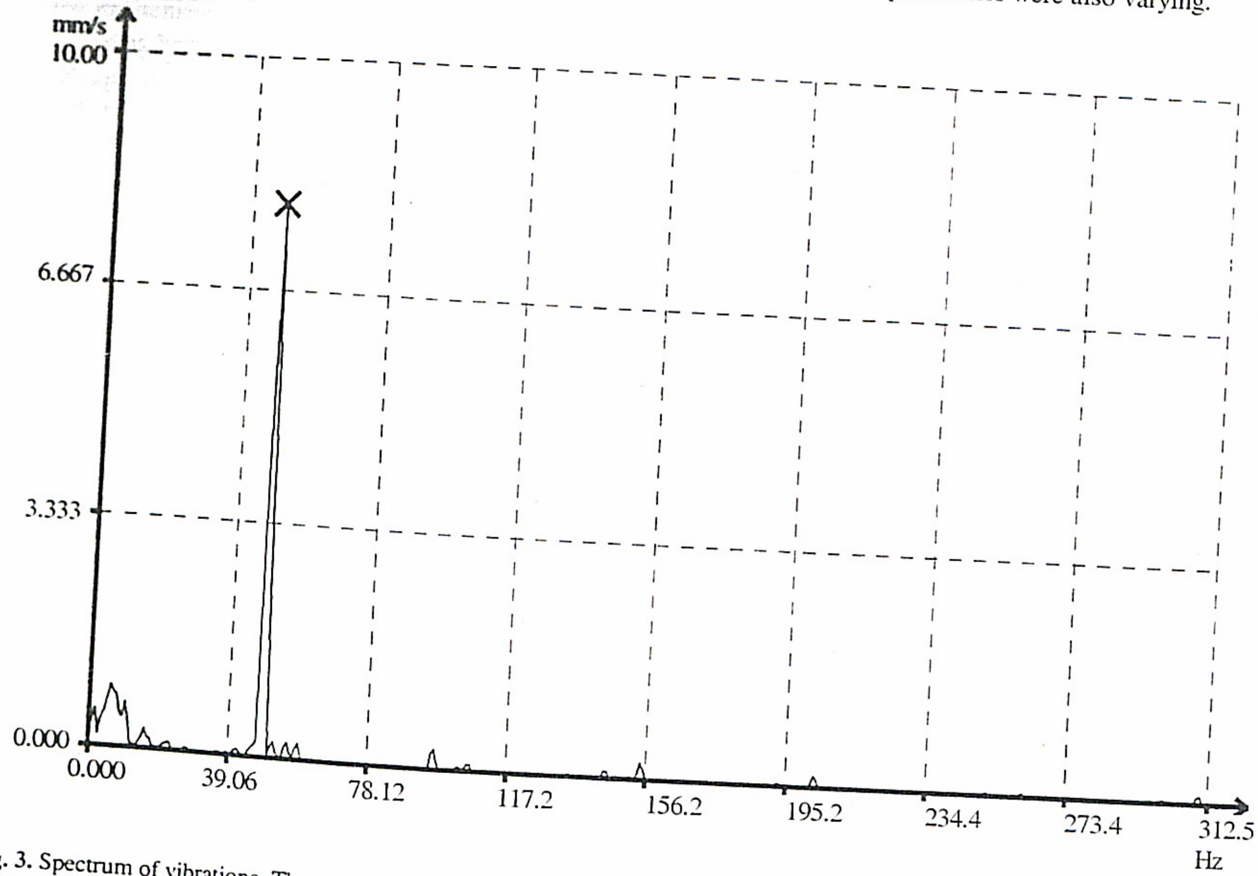
3) auto vibrations excited by the lubricants of sliding bearings.

For this reason, a number of measurements and experiments have been made for determining the potential vibrations source. Primarily, the vibration level of the separate parts of the high-pressure case, the coolers and the moisture separator has been measured. All measurements have been made in the vertical direction. The obtained results are given in Table 1.

As seen in Table 1 the highest vibrations levels on the compressor case at the suction collector to the fourth section. Estimation of the obtained measurement results allows stating that the main source of vibrations lays either in the suction collector or in the second section of the high-pressure case.

In order to determine the sources of vibrations more accurately, the compressor operation parameters have been registered and attempts have been mode to determine the vibrations dependence on the compressor efficiency while the temperature of the sucked air and the intermediate pressures in the suction and compression lines have been varying.

Fig. 4 indicated the variation of vibrations of the compressor bearing case in the vertical direction while the above-mentioned parameters were also varying.



ig. 3. Spectrum of vibrations. The marked point – 48,44 Hz, 7,903mm/s, general vibrations level – V_{RMS} 5,679 mm/s

Table 1.

Spectrum composition of vibrations of the individual parts of the compressor.

Measurement site	V_{RMS}, mm/s	Three largest components of vibrations, Hz		
		1	2	3
Front bearing cases	3,2	48,44	54,69	153,32
Compressor case at the front bearing	4,0	48,44	9,77	153,32
Compressor case at the suction to the third section	5,5	48,44	3,91	6,84
Compressor case at the exit from the third section	6,0	48,44	9,77	153,32
Compressor case at the suction to the fourth section	10,0	48,44	6,84	3,91
Compressor case at the exit from the fourth section	7,0	48,44	54,69	4,88
Compressor case at the back bearing	9,0	48,44	3,91	93,75
Back bearing case	5,7	48,44	5,469	9,375
First grade cooler	1,8	1314,45	48,44	1972,66
Second grade cooler	2,0	48,44	9,77	54,69
Moisture separator	4,6	48,44	1314,45	54,69
Third grade cooler	3,4	48,44	54,69	1073,24

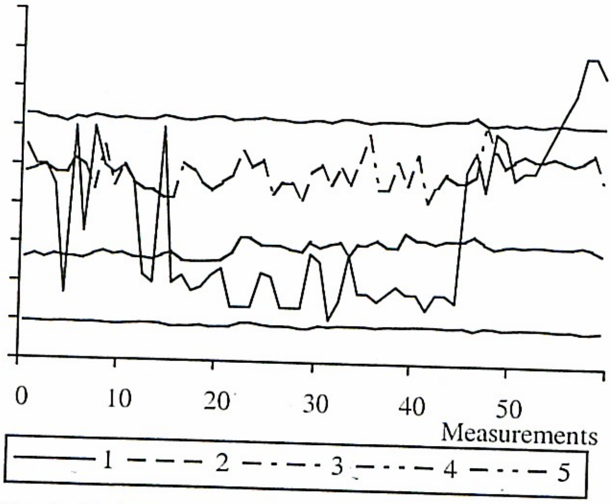


Fig. 4. Variation of vibrations level under different working modes: 1 – vibrations rate V_{RMS}; 2 –general efficiency of the compressor, 1000 m^3/h; 3 – pressure of the air sucked into the second section of the high case; 4 – temperature; 5 – air pressure in the compression collector.

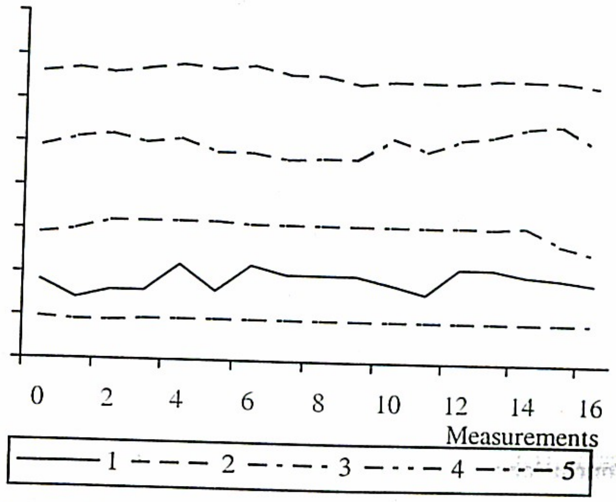


Fig. 6. Variation of vibrations level under various working modes: 1 – vibration rate V_{RMS}; 2 – general compressor efficiency, 1000 m^3/h; 3 – pressure of the sucked air into the second section of the high pressure case; 4 – temperature; 5 – air pressure in the compression collector.

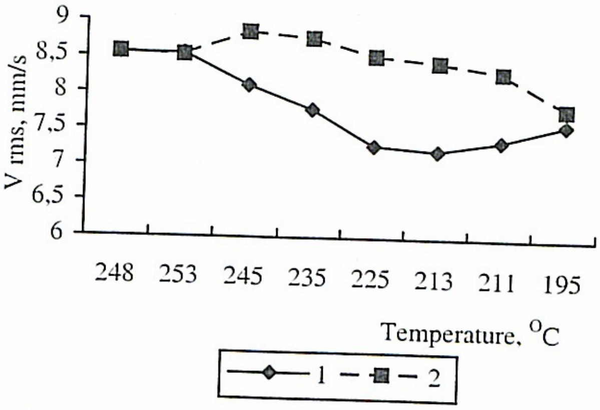


Fig. 5. Variation of the compressor vibrations at the additional water supply to the suction of the fourth section: 1 – curve obtained by raisin the water quantity; 2 – curve obtained by reducing the water reducing the water quantity.

In order to reduce the temperature of the sucked and compressed air and to increase the mass of the compressed material an additional experiment was made, namely, the water was supplied to the fourth section suction. The obtained results are given in Fig.5. The quantity of the supplied water was determined according to the temperature of exhausted air.

Having analyzed the monitoring and experimental data for precise determination og the defects causing 48,44 Hz frequency vibrations, during the overhaul it was recommended:

1) opening the high pressure case to check the alignment of diaphragms with respect to the rotor;

2) to check the state of the rotor labyrinths seals;

3) to check sealing of the separating plane;

4) to check the technical state of diaphragms and diffusers;

5) to inspect the rotor sliding bearings and evaluate their state.

After the overhaul of the turbocompressor no rise of the vibrations level was noticed. Fig. 6 shows the variation of vibrations level under various modes after the overhaul.

As seen from Fig.6 after the overhaul the compressor work is more stable and its efficiency is higher for the same revolutions of the steam turbine. After the overhaul the vibrations level of the compressor enters the zone B according to ISO 10816-3 standard.

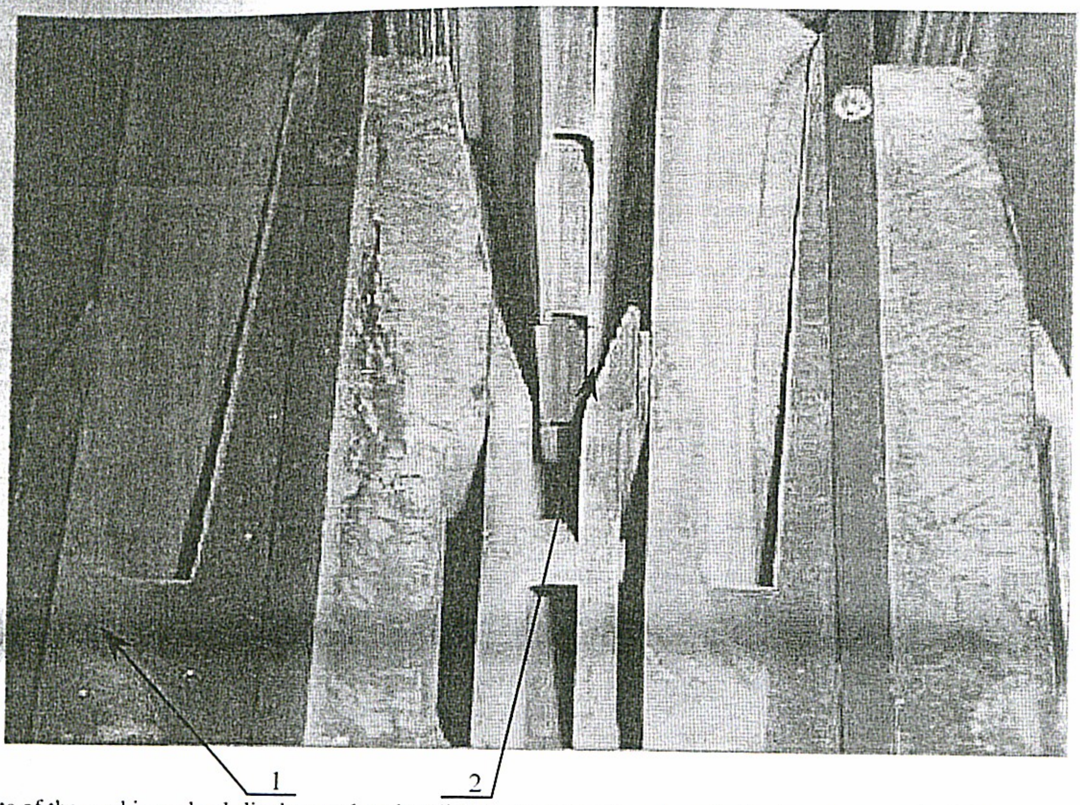


Fig. 7. Defects of the working wheel-diaphragm 1 and sealing in the separating plane 2 of turbocompressor K 1290-121-1.

Conclusions

1 After the observation and above-mentioned experiments made in the turbocompressor it was found that the vibrations level in the primary stage of the compressor defect development, depended on the compressor load i.e. on the compressed air volume and its mass and temperature (Fig. 2, 1-20 measurements). Afterwards, from 45 – 48 measurements the compressor defects exerting the rise of the vibrations level were unpredictably developing and the vibrations level tended to rise.

2 The air mass increase in the suction by additional supply of a small quantity of water to it had a short-term effect. Later on the vibrations level started rising and only after full stopping of water supply the initial vibrations level set in.

3 Taking into the account the experiments made it was determined that the defect exerting a relatively high vibrations rise lies in the compressor high-pressure case close to the fourth section. It may be the air stream, leakage lither through the seals in the separating plane of the compressor case or though the seal labyrinths. This assertion was confirmed after the stoppage of the compressor for overhaul: the leakage was detected in the seals of the compressor separating plane and the air stream penetrated from the fourth section into the third one. One more defect possibly causing the emergence of low frequency vibrations was one of the working wheels improperly installed in the axial direction with respect to the diffusers in the compressor fourth section (Fig.7).

References

1. **Bucharin N. N.** Modeling of Characteristics of Centrifugal Compressors (in Russian). Leningrad. Mashinostrijenije. 1983 - 214 p.

2. **Kazakevich V. V.** Natural Vibrations (Pumpage) in Compressors (in Russian). Moskva. Mashonostrojenije. 1974 - 264 p.

3. **Simonovskyi V. I.** Rotor Stability and Non-Linear Vibrations in Centrifugal Machines (in Russian). Kharkov. Visha shkola. 1986 - 128 p.

4. **Muszynska A., Bently D. E.** Fluid – Induced Instabilities of Rotor: Whirl and Whip –Summary of Results. Orbit: Bently Nevada. March 1996. vol. 17. No. 1 7-15 p.p.