AERODYNAMIC SUPPORT OF A BIG INDUSTRIAL TURBOBLOWER ROTOR

Jiří Šimek*, Jan Kozánek**, Milan Šafr***

Aerodynamic bearing support for the rotor of a 100 kW input industrial turboblower with operational speed of 18 000 rpm was designed and manufactured. Rotor with mass of about 50 kg is supported in two tilting-pad journal bearings 120 mm in diameter, axial forces are taken up by aerodynamic spiral groove thrust bearing 250 mm in diameter. Some specific features of the bearing design are described in the paper and the results of rotor support tests are presented. The paper is an extended version of the contribution presented at the Colloquium 'Dynamics of Machines 2006'.

Key words: *industrial turboblower, aerodynamic bearing, tilting-pad journal bearing, spiral groove thrust bearing, rotor-dynamic calculation, rotor relative vibration measurement*

1. Introduction

Aerodynamic support of ČKD NE industrial turboblower, using VÚES Brno high frequency motor, was designed in 2004 [1]. The machine with the input of 100 kW has operational speed of 18 000 rpm. The turboblower rotor was originally supported in ceramic angular-contact ball bearings, which had insufficient durability. Preliminary rotor-dynamic calculation showed, that it is possible to increase the rotor stiffness and to shift the 1st bending critical speed high above operational speed. This was necessary condition for successful reconstruction of the blower to aerodynamic bearing support. The problems arising from manufacturing inaccuracies were subsequently eliminated and several tests were carried out. The final phase of tests was accomplished with the turboblower completed with the compressor impeller and spiral casing.

2. Bearing support design

The original rotor mass was about 35 kg, but its necessary stiffening and introduction of relatively big thrust bearing runner increased its mass to about 50 kg. With thrust runner made of titanium the load is distributed almost evenly to both journal bearings. In order to achieve sufficient bearing life, the specific loading of journal bearings should not exceed 0.015 MPa. This value determined the journal bearing dimensions – diameter and length of 120 mm. The axial force not being specified, the thrust bearing diameter was chosen as the maximum possible according to allowable stress at operational speed, i.e. 250 mm.

Overall machine design without compressor impeller is apparent from Fig. 1. The rotor 1, equipped with stiffening sleeves 2 and 3, is supported in two identical aerodynamic

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tilting-pad journal bearings. The bearing bodies 4, 5, fastened to the stator of electromotor, had three radial openings, 120° one from another. Into the openings are inserted freely moving pad supports 7 holding the pins 18, on which the pads 6 can tilt both in circumferential and transversal directions. Position of the two lower supports, which bear the static load of the rotor mass, is secured by stops 8. The upper support position can be set by means of adjustable stop 11. Between the stop 11 and the upper support there is the spring 10, which together with the socket 9 allows the upper support to move in radial direction, in case the bearing clearance decreased due to temperature dilatations. The spring preload can be adjusted by means of screw 19, so that it is lower than load carrying capacity of the pad. This arrangement ensures, that the bearing clearance cannot decrease below a safe limit. The basic clearance is set by adjustable stop by measuring the rotor lift in vertical plane.

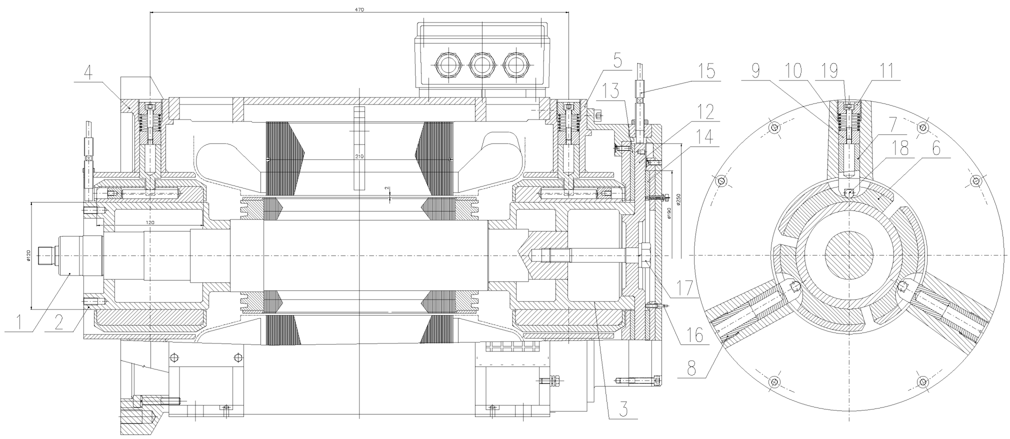

Fig.1: Turboblower with aerodynamic bearing support

Axial forces are taken up by the main thrust bearing 13, axial position of the rotor is secured by auxiliary thrust bearing 14, with diameter of only 190 mm. The titanium thrust runner 12 is centred in the right rotor sleeve and fastened by screw 17. The runner surface is equipped with shallow spiral grooves (Fig. 2), which pump the air from outer diameter to bearing centre. At the inner diameter of the thrust bearing there is an annulus without grooves, which prevents the leak of generated pressure to ambient atmosphere. The groove depth is about 50 μm and the assumed gas film thickness is from 10 to 15 μm. Therefore the alignment of the runner and of the main thrust bearing should be very accurate.

For reliable operation of aerodynamic support it is necessary to use suitable materials for the sliding surfaces, because accidental contact of the bearing surfaces with the rotor cannot be excluded. The best sliding properties so far proved surfaces made of electrographite impregnated by antimony. Therefore the sliding surfaces of all pads and those of both thrust bearings were manufactured from this material supplied by Kompositum Topolčany. The rotor sleeves are hardened, so that their surface would not damage in case of contact with the electrographite lining.

To observe the rotor behaviour the turboblower is equipped with sensors of relative vibration 15. The sensors are located as near the journal bearings as possible. At the left bearing the sensors observe directly the surface of bearing journal, while at the right bearing, because of insufficient distance between the journal and thrust bearing, the sensors

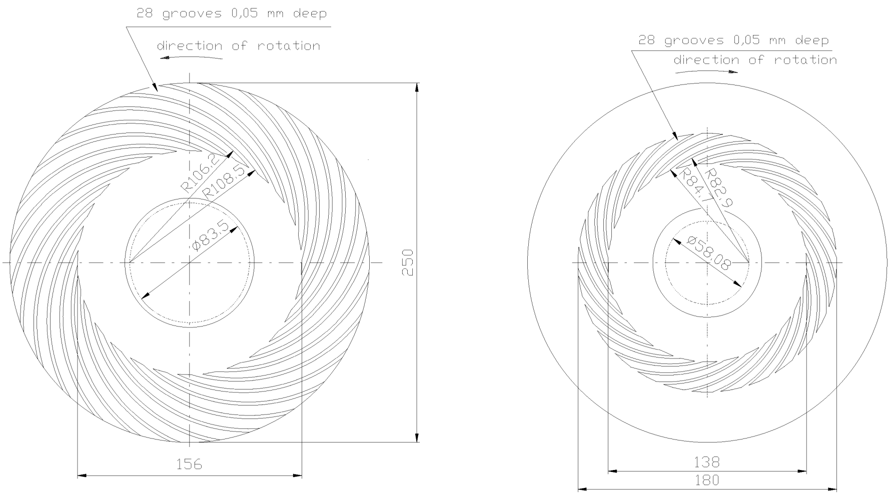

Fig.2: Spiral groove thrust runner (main bearing – left, auxiliary bearing – right)

are located above the cylindrical surface of thrust runner. Schenck IN-085 transducers, working on eddy current principle, were used for relative vibration measurement of the rotor. The IN-085 transducers have measuring range of 2 mm and sensitivity 8 mV/μm or 13 mV/μm on magnetic or non-magnetic surface. The sensors have the oscillator integrated into the transducer body, so that their resistance to external disturbances is very high. At each bearing location two sensors are used.

Calculation of the rotor dynamic, with the bearing stiffness and damping coefficients included, indicated the 1st bending critical speed in the neighbourhood of 34 000 rpm. There are four 'bearing' critical speeds, i.e. critical speeds of the rigid rotor on gas bearing film. These critical speeds, ranging from about 6 000 to about 15 000 rpm, are relatively well damped. More detailed dynamic analysis of the ATUR turboblower rotor, the pad dynamic included, can be found in [2].

3. The test results

Due to insufficient distance of relative sensors from pad side the measured signals on the impeller side (IS) were not correct. Therefore only relative vibration at the thrust bearing side (TBS) could be measured during the 1st test. That is why problems with reduced bearing clearance at IS were diagnosed too late and the journal and pad sliding surfaces were damaged. In Figs. 3 to 5 are samples of relative vibrations measured in the 1st test.

Character of vibration signals in Fig. 3, which includes speeds from about 1 400 to about 1 600 rpm, indicates the journal lift off and separation of sliding surfaces by the gas film. At 10 000 rpm relatively significant subharmonic vibration components appeared, with the frequency equal to roughly 1/3 and 1/2 of rotational frequency. At 11 000 rpm the subharmonic component with 1/3 frequency in vertical direction is almost comparable to component with the frequency of rotation. With further increasing of speed the subharmonic vibration component amplitude somewhat decreased, but from about 13 000 rpm (Fig. 4) it grew again, later accompanied by the change of vibration character. The difference in vibration character in time domain is evident from comparison of Fig. 4 – 13 500 rpm, and Fig. 5 – 15 200 rpm in which also marked extension of subharmonic vibration region is apparent. The presence

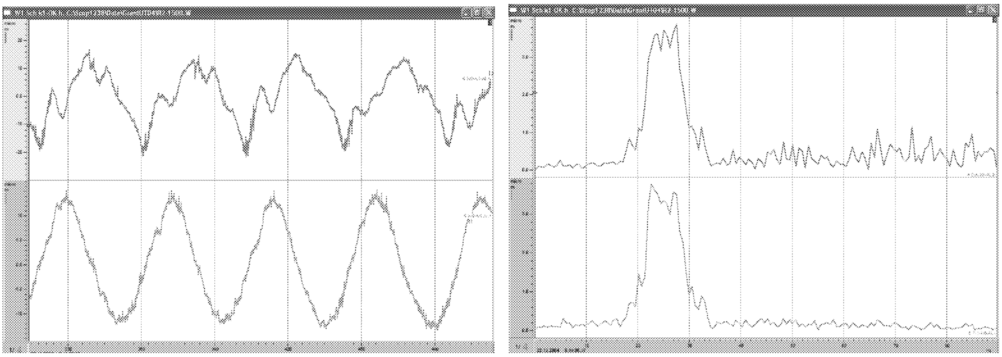

Fig.3: Vibration signals and frequency spectra at about 1 500 rpm

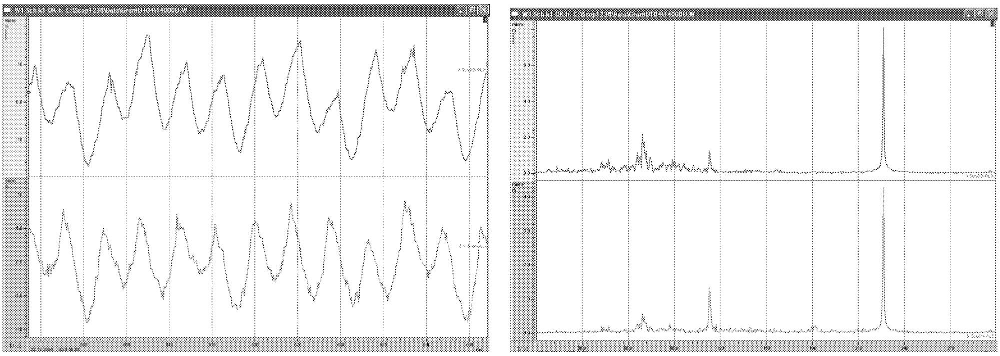

Fig.4: Vibration signals and frequency spectra at 13 500 rpm

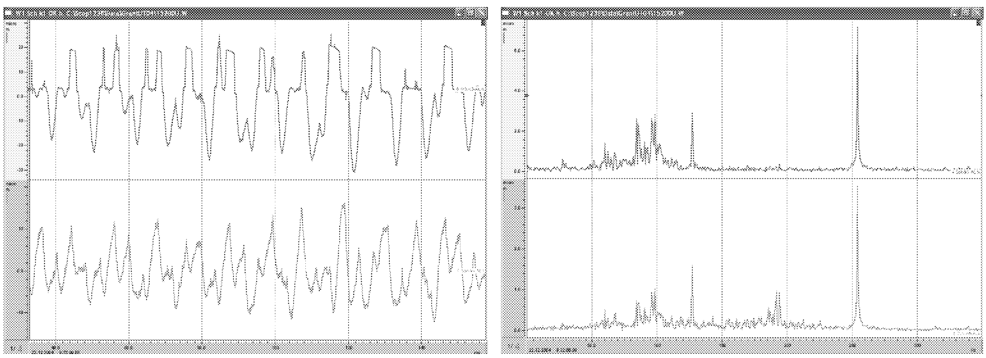

Fig.5: Vibration signals and frequency spectra at 15 200 rpm

of subharmonic vibration component obviously indicated the contact of sliding surfaces in journal IS bearing, which was confirmed by the machine disassembly.

It was ascertained, that the screw 19 (see Fig. 1), which secures preload of the spring 10 in the IS bearing, was loosened. Subsequent shift of the socket 9 resulted in elimination of basic clearance and the spring started to press the pad 6 to the journal. Although the pad load carrying capacity should be greater than the spring force, intermittent contact of all the pads with journal surface took place. The bearing temperature increased rapidly and the pad and journal surfaces were damaged (see Fig. 6). Luckily the damage was only slight and the sliding surfaces could be repaired relatively easily.

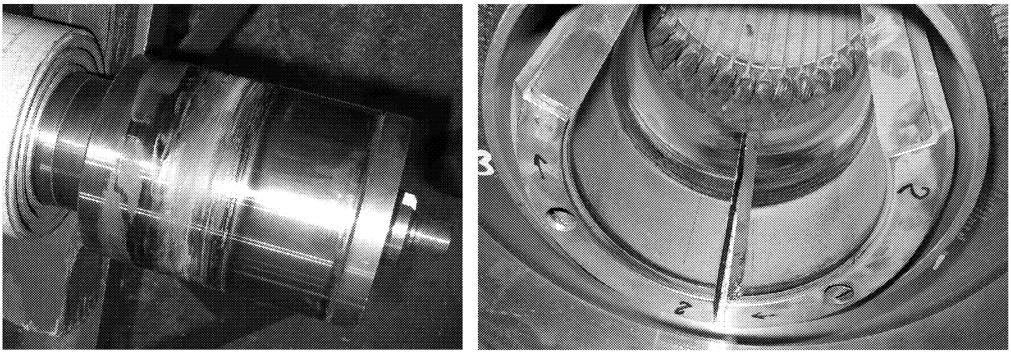

Fig.6: Damaged surfaces of the front bearing journal and pads

Rear journal bearing and its pads (Fig. 7), as well as both thrust bearings and the surfaces of the thrust runner were practically intact with slight traces of mutual contact during start-up and run-down. After the IS journal bearing repair the tests could be resumed.

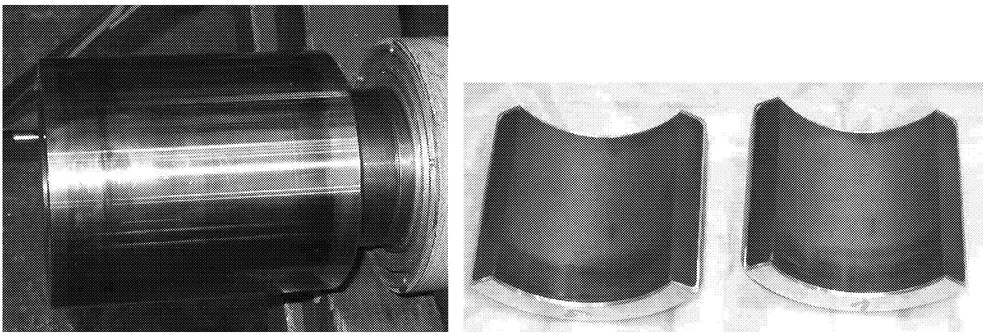

Fig.7: Undamaged surfaces of the rear bearing journal and pads

During the 2nd phase of test the relative vibrations of rotor without impeller were measured in both locations near the rotor ends in two perpendicular directions. Due to failure of one sensor only three vibration signals could be measured simultaneously. Therefore two test runs were needed – the 1st one with two sensors in location IS, the 2nd one with two sensors in location TBS. The amplitude-frequency characteristics of both runs in Fig. 8 show favourable RMS values of rotor vibrations at both ends of the rotor. The resonance peek RMS value of amplitude 12 μm corresponds to peak-to-peak amplitude of 33 μm, i.e. only 25 % of bearing clearance.

Fig. 9 illustrates difficulties with measuring the static displacement of journal centre. Even the most sophisticated systems, such as ADRE for Windows, are not able to measure realistic displacement due to temperature shift. Though the load in both journal bearings is basically the same, on IS bearing system ADRE indicates displacement of about 150 μm, while at the TBS the measured displacement is roughly double that value. The realistic value of journal lift, based on numerical calculation, is about 30 μm.

Due to excessive heating of main thrust bearing the clearance of journal bearing on the thrust bearing side was reduced as a consequence of different speed of thermal expansion of the journal and bearing body. It can be demonstrated by relative vibration signals at the TBS shown in Fig. 10. The vibration signal amplitude in vertical direction (middle

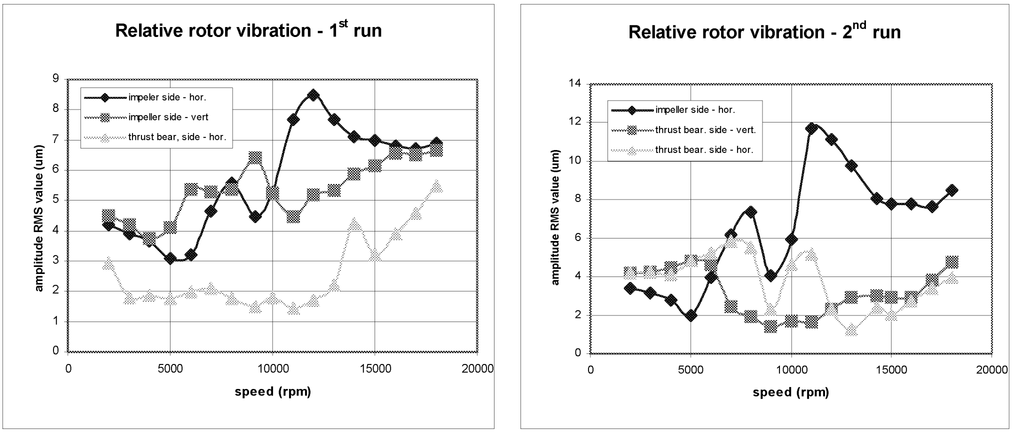

Fig.8: Rotor relative vibration with fixed thrust bearing

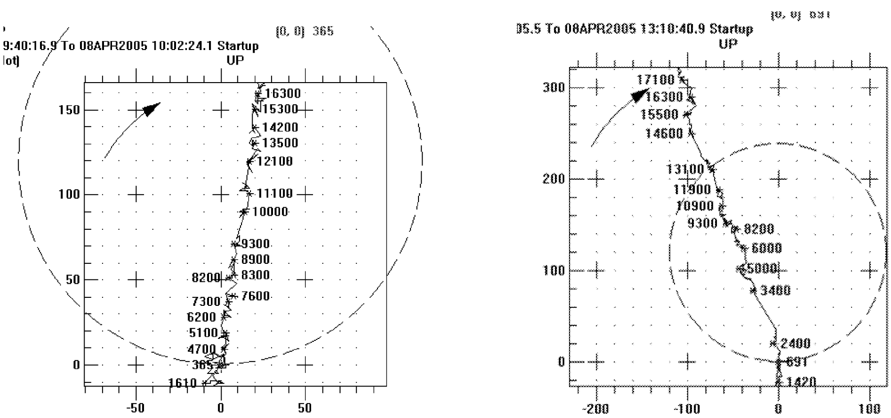

Fig.9: Static trajectory of journal centre (IS left, TBS right)

curve) is remarkably smaller than the horizontal one (lower curve) and it shows symptoms of intermittent contact of pad with the shaft surface. Within the relatively short time interval the vertical amplitude is further reduced and the symptoms of pad-journal contacts are more distinct (Fig. 10 right). During rotor run-down (Fig. 11) the bearing clearance was reduced even more, which was accompanied by distinct subharmonic component with 1/2 rotational frequency (Fig. 11 right). In a journal bearing with all the pads fixed this situation would lead to complete elimination of bearing clearance and consequently to heavy damage of sliding surfaces. The system with spring preloaded upper pad (see Fig. 1) secured gas film of minimum thickness, thus preventing any surface damage, as appeared after the machine disassembly.

4. Design modifications

Originally the thrust bearings were fastened to the machine body, but due to manufacturing imperfections it was impossible to achieve necessary parallelism of the bearing and runner sliding surfaces and the load carrying capacity of thrust bearing was therefore minimal. This is demonstrated by traces of contact with the runner at the main thrust bearing sliding surface (Fig. 12). For substantial improvement of the situation it was necessary to

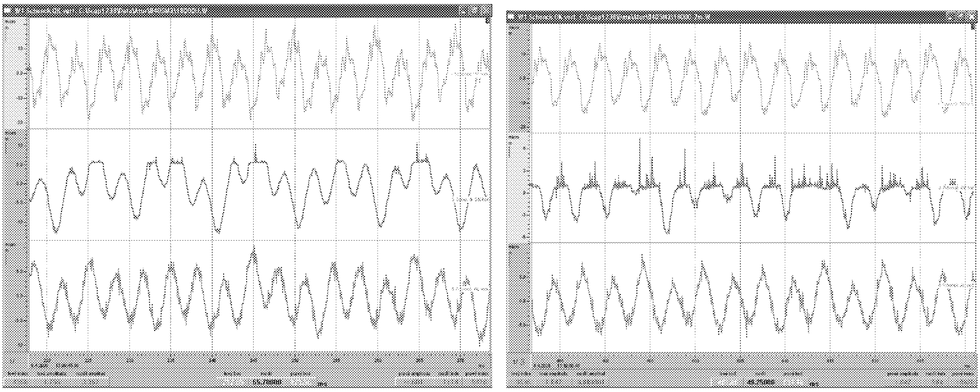

Fig.10: Relative rotor vibration at 18 000 rpm (right record ≈ 1 minute after the left one)

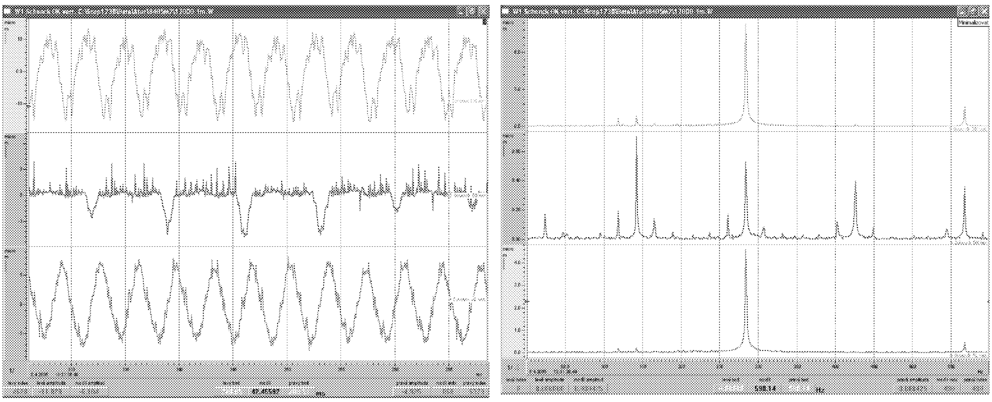

Fig.11: Relative rotor vibration and frequency spectrum during run-down at 17 000 rpm

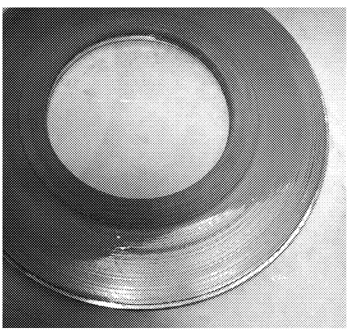

Fig.12: Main thrust bearing after test

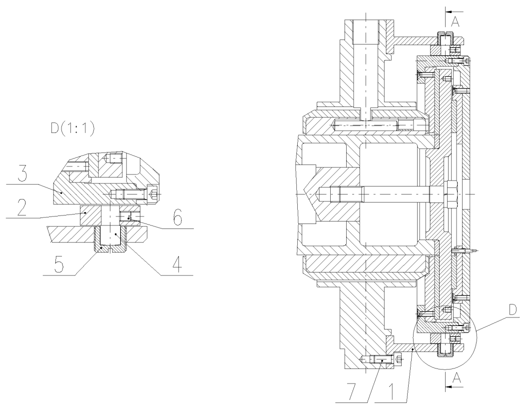

Fig.13: Thrust bearing block gimbal suspension

reconstruct thrust bearing support to gimbal suspension, enabling the bearing sliding surface to align according to thrust runner surface. The details of gimbal mounting are shown in Fig. 13.

The thrust bearing block 3, consisting of the main and auxiliary thrust bearings, is supported by means of two pivots in the ring 2. The ring 2 is connected to the bearing carrier 1 through another pair of pivots 4. The threaded bushings 5, in which the pivots swivel, enable to set up position of the thrust bearings in radial direction. The gimbal allows the main thrust bearing block to turn about two perpendicular axes and thus to align with the runner. After reconstruction of thrust bearing suspension the turboblower tests were resumed.

5. Further test of the turboblower

Also in this phase of test the relative vibrations of rotor without impeller were measured in both locations near the rotor ends in two perpendicular directions. With regard to expected gimbal vibration, due to the runner sliding surface pitching, also the thrust bearing block vibration measurement by means of an accelerometer was prepared.

The rotor operation was smooth up 12 000 rpm as can be seen from relative vibration signals and their frequency spectra in Fig. 14 (upper curve – impeller side, lower curve – thrust bearing side). The vibration amplitudes are relatively big because the rotor operates near rigid rotor critical speed. Very small subharmonic vibration can be observed in both directions of measurement.

When the speed was further increased, the rotor vibration suddenly increased and it was accompanied by substantially intensified sound. It was supposed, that the increased vibration is caused by oscillation of thrust bearing block and by contacts of bearing sliding surface with thrust runner. Dismantling of thrust bearing confirmed the assumption by traces of contact with the runner at opposite sides of the main and auxiliary bearing.

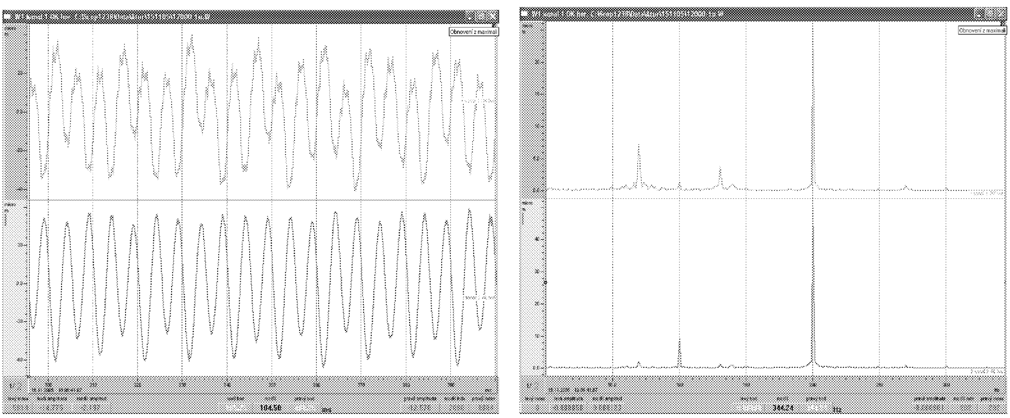

Fig.14: Relative rotor vibration at 12 000 rpm

To identify the speed in which the sliding surfaces would come into contact the axial clearance was somewhat increased and accelerometer was fastened to thrust bearing block. The onset of increased vibration was apparently due to greater axial clearance somewhat shifted – to about 14 000 rpm. The moment of onset of thrust bearing contacts with the runner is evident from the records of rotor relative vibration (lower curve) and vibration of thrust bearing block measured by accelerometer (upper curve) in Fig. 15. It is evident, that the rotor relative vibration level was doubled and its almost ideal harmonic course was heavily deformed. The acceleration signal shows rise of RMS value from $6.5\,\mathrm{m\,s^{-2}}$ to $9.7\,\mathrm{m\,s^{-2}}$. As the RMS value is determined statistically from relatively long recording, the rate of increase is highly reliable.

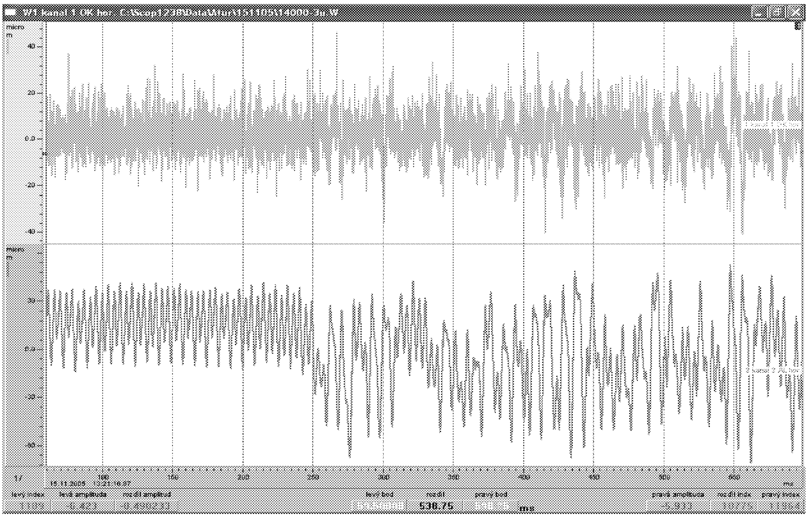

Fig.15: Onset of increased vibration due to the contact of thrust bearing with rotor

After the machine disassembly the above suspicions were confirmed. The thrust bearings and thrust runner showed traces of mutual contact in direction perpendicular to axis of inner gimbal pivots and also the journal and pads at the thrust bearing side bear marks of contact (see Fig. 16)

Both the thrust and journal bearing damage was slight and so after cleaning of sliding surfaces the machine could be reassembled, this time with the impeller and spiral casing.

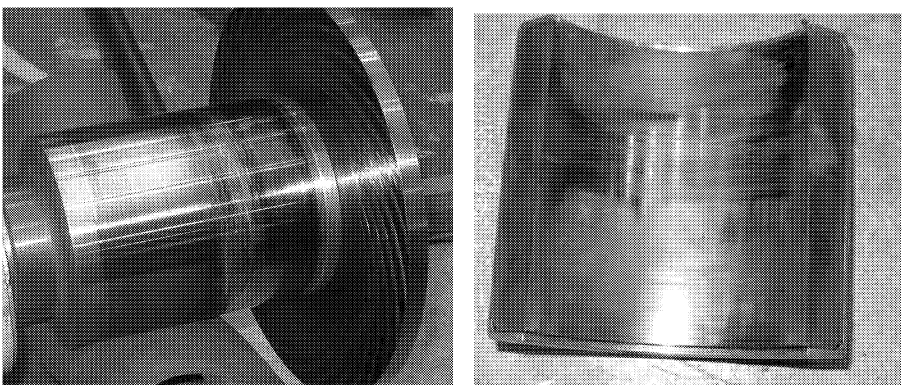

Fig.16: Thrust bearing side journal and pad after the 2nd phase of tests

6. Test of the completed turboblower

The layout of the turboblower completed with the impeller and spiral casing can be seen in Fig. 17, photographs of the complete rotor and assembled machine are in Fig. 18. The assembly of the impeller and spiral casing excluded the mounting of relative vibration

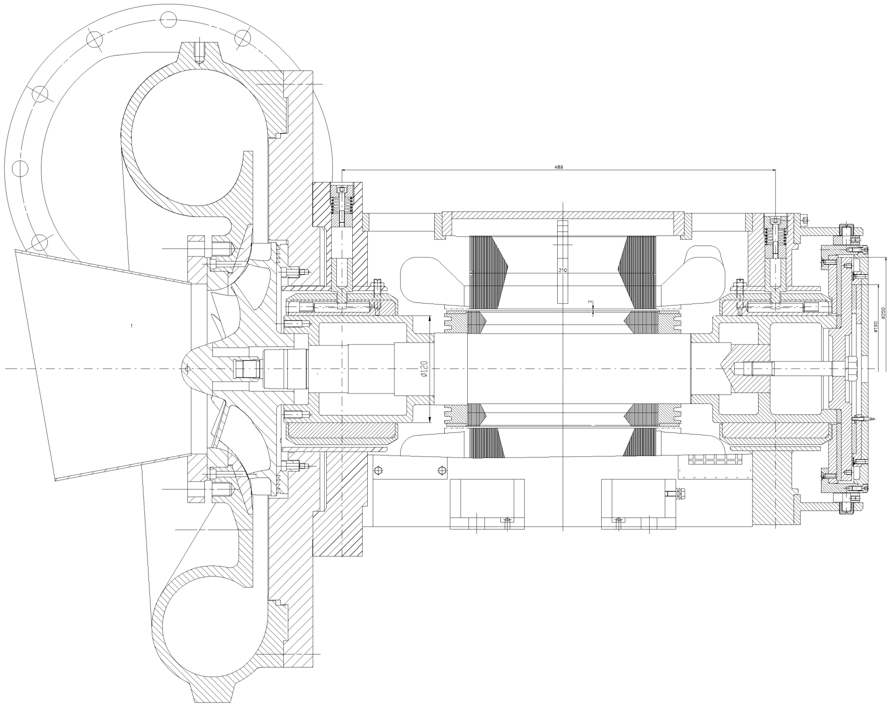

Fig.17: Complete turboblower with the input of 100 kW

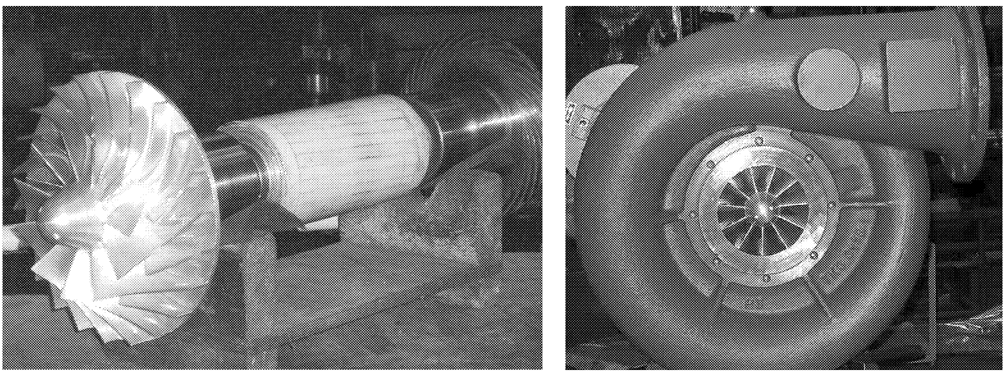

Fig.18: Assembled rotor and complete machine

sensors at the impeller location. Therefore in completed machine relative rotor vibration could be measured at the thrust bearing side only.

Movement of gimbal suspension in the final test was damped by inserting elastic glue in the gaps between thrust bearing block 3, ring 2 and bearing carrier 1 (see Fig. 13). This operation was carried out with the thrust bearing in contact with the thrust runner, so that the sliding surfaces remain aligned. This modification proved effective, because the RMS value of gimbal suspension vibration at 9 000 rpm decreased from original value of $6.5\,\mathrm{m\,s^{-2}}$ to $1.1\,\mathrm{m\,s^{-2}}$. The maximum speed was restricted to 9 000 rpm because of imperfect seating of the impeller in the rotor bushing. Up to this speed the rotor run quite smoothly with small vibration amplitudes, as is evidenced by amplitude-frequency characteristics of run-down from the system ADRE in Fig. 19. Maximum peak-to-peak amplitude of about 45 μm corresponds to amplitudes measured in the 1st phase of the test, considering changes in residual unbalance due to mounting of the impeller.

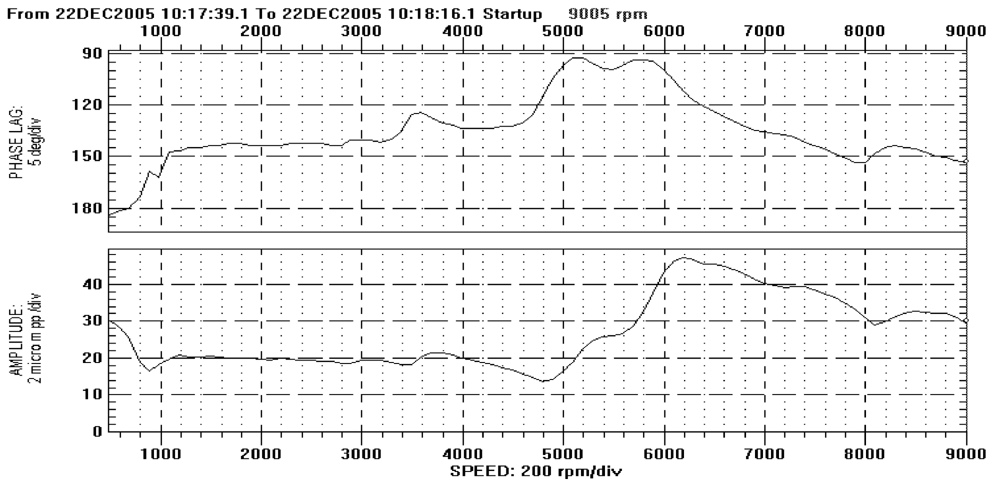

Fig.19: Amplitude/phase – frequency characteristic of run-down, thrust bearing side

Correct function of aerodynamic bearing support was confirmed also by journal centre orbits on thrust bearing side of the rotor. The journal centre trajectories generated during run-down by system ADRE are presented in Fig. 20 and 21. It can be seen, that at 2 000 rpm the rotor is still 'airborne'. The orbits at 9 000, 7 000 and 6 000 rpm show considerable anisotropy of the aerodynamic tilting-pad journal bearing, which is in agreement with computed results.

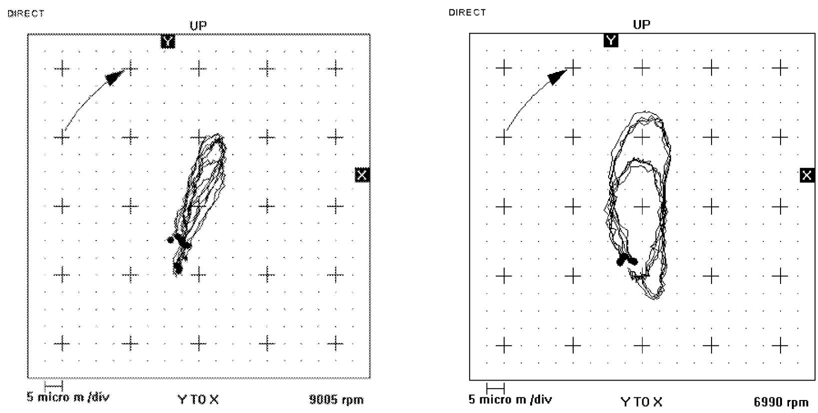

Fig. 20: Journal centre trajectory at 9 000 rpm (left) and 7 000 rpm (right)

7. Conclusions

Aerodynamic bearing support of a big rotor with the mass of about 50 kg was designed, manufactured a tested up to maximum operating speed of 18 000 rpm. In spite of the complications with delivery of modified electric motor stator, which required among others increasing the bearing distance, the machine with aerodynamic bearings proved to be oper-

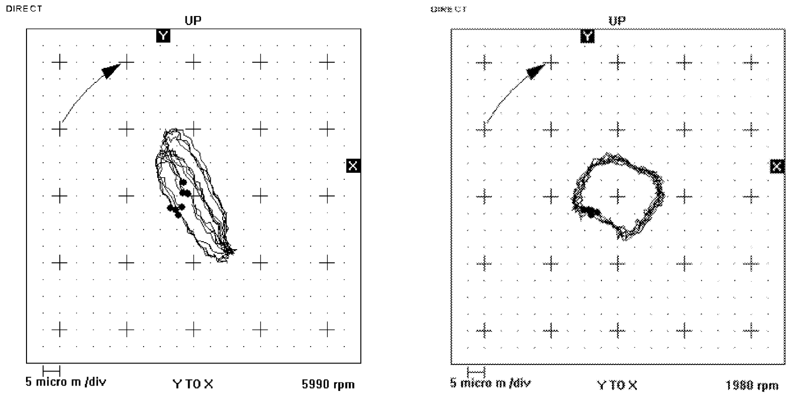

Fig.21: Journal centre trajectory at 6 000 rpm (left) and 2 000 rpm (right)

ational. The maximum double amplitude of relative vibration in both journal bearings was lower than 30 % of bearing clearance and no symptoms of its increasing due to 1^{st} bending critical speed were noted.

Based on the test results it is realistic to expect the ability of aerodynamic support to ensure long-term reliable operation of the ATUR turbocharger rotor. It also opens the perspective of designing even bigger machines with input exceeding 100 kW, which will be necessary for operation of very high temperature gas cooled nuclear reactors.

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