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Application of a compliant foil bearing for the thrust force estimation in the single stage radial blower

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Abstract: The paper presents the application of a compliant foil bearing for estimation of the thrust force in a single stage radial blower under operational conditions. The bump foil of the thrust bearing behaves as a nonlinear spring. The knowledge of the spring deflection curve allows estimation of the actual thrust force for a measured bump deflection at the given rotational speed. To acquire the deflection curve, static calibration of the axial shaft displacement sensor was performed. During the calibration, the information about voltage signals of the sensor for the given loading force was collected. The measured voltage values at different speeds and loads were then converted into the thrust force. The results were verified by comparison to the thrust force resulting from the pressure distribution on the impeller.

Keywords: compliant foil bearings; thrust force estimation; single stage radial blower

1 Introduction

Foil bearings are aerodynamic gas bearings that use air as the working fluid. It is well known that the application of foil bearings has a beneficial effect on machine properties. This solution keeps the working fluid absolutely oil-free, decreases friction and simplifies the machine design. Air as the working fluid has very good properties: it is clean, ensures rotordynamic stability and behaves predictably [1]. However, there are some disadvantages as well – foil bearings have the minimum speed of operation, generate high friction during start-ups and shut-downs, and thrust foil bearings have relatively low load carrying capacity [2–4].

A unique feature of the thrust foil bearing (Figure 1) is a noticeable deflection that appears on the machine impeller under normal operating conditions in response to the thrust force [5, 6]. This behaviour is the most undesirable feature in particular. It can cause some extensive damage during machine operation, since the relatively soft bearings cannot prevent rotor axial displacement, causing blade tip clearances to disappear. During investigations of this misalignment, however, it has been discovered that this disadvantage may be transformed into a positive feature.

The axial displacement can be measured with a non-contact eddy current probe. The spring deflection curve of the thrust foil bearing allows one to correlate the shaft displacement to the thrust force that appears on the impeller. This system can function in real-time during normal compressor operation [7].

For this purpose, a previously built test bench was used in this study (Figure 2). The experiment was carried out on a single stage radial compressor supported on journal and thrust aerodynamic foil bearings. The compressor was driven by a synchronous motor controlled by a power inverter. With this type of drive, it was possible to change the operating speed smoothly.

During the operation of the compressor, an axial thrust force appears. The thrust force results from the pressure distribution on the impeller. As shown in Figure 1, the thrust foil bearing behaves as a nonlinear spring. When the axial force occurs, the bearing deflects slightly. This deflection is measured with a preinstalled shaft displacement sensor. The system used in this study utilized the existing probe and no additional instrumentation was needed.

2 Compressor stage reconstruction calculations

In order to confirm the results of the experiment, the compressor stage was computationally reconstructed. First, a theoretical model of the compressor was created. For this purpose, the impeller was dismantled and measured. To achieve the most accurate results, the rotor itself was mea-

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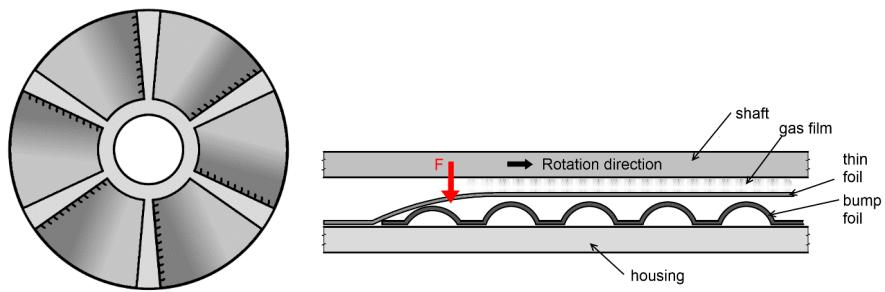


Figure 1: Design of the thrust foil bearing.

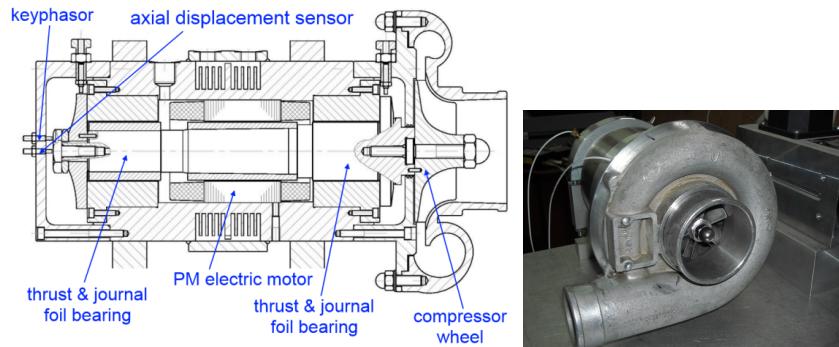


Figure 2: Cross-sectional view of the compressor supported on foil bearings [10].

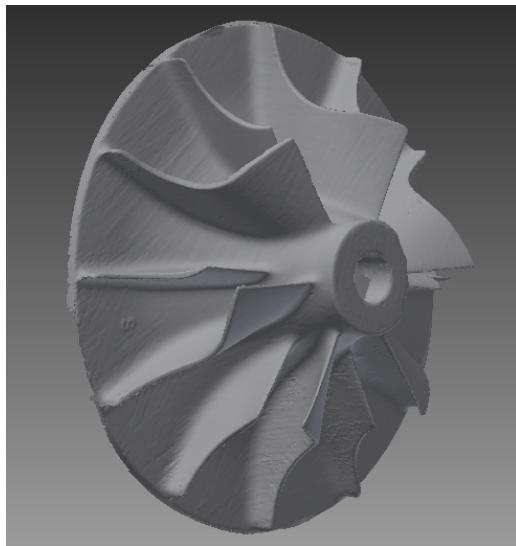


Figure 3: 3D scan of the compressor impeller.

sured with a 3D scanner. The scanner created a cloud point model that represented the dimensions of the rotor accurately (Figure 3).

The calculations were based on previous experience in the compressor and turbine design. The temperatures, densities and pressures upstream and downstream the impeller were determined for the maximum achievable speed of the compressor. The results are shown in Figure 4.

3 Experimental results obtained from the foil bearing blower operation

Figure 5 presents a blower test rig with a mounted flow measurement tube that was designed in order to obtain the stage operational parameters. The mounted measurement device indirectly provided information not only about the axial force, but also about the compressor power, mass flow, etc.

The measurement device consisted of a long, relatively thin tube with a flow straightener right behind the compressor scroll. This design straightened the flow of the compressed air and guaranteed an undisturbed laminar flow profile [8].

About 450 mm from the beginning of the tube, three holes were made circumferentially to measure the static wall pressure. A Prandtl probe was installed 50 mm behind these holes. Its purpose was to measure the dynamic pressure in the tube, to find out the flow velocity. The tube was fitted with a special design choke valve at the end (Figure 6). This construction allowed the fluid to flow circumferentially, thus it did not cause any turbulence even when choking. Temperature was measured centrally at the beginning of the tube with a thermocouple. At the non-

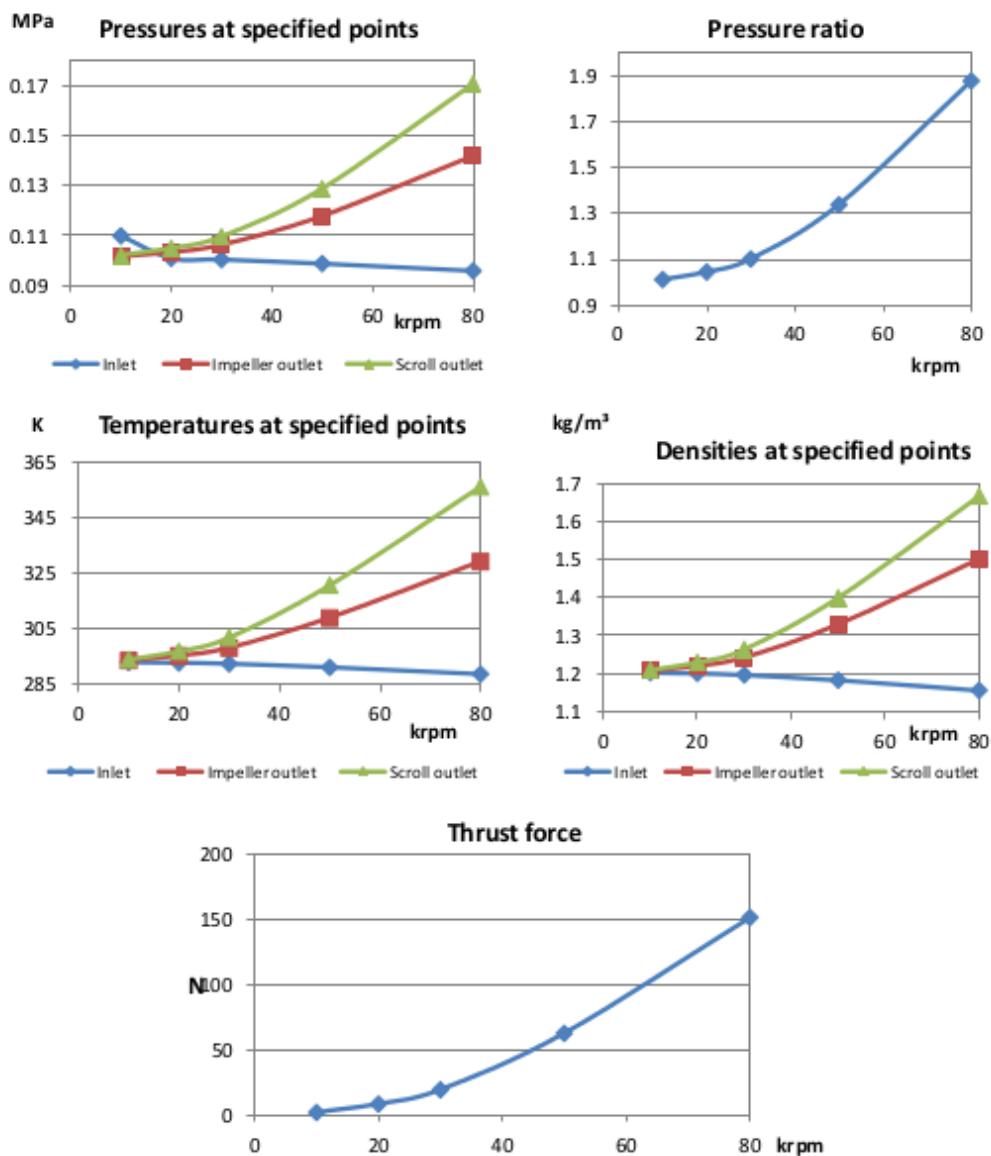


Figure 4: Calculated parameters of the reconstructed compressor stage.

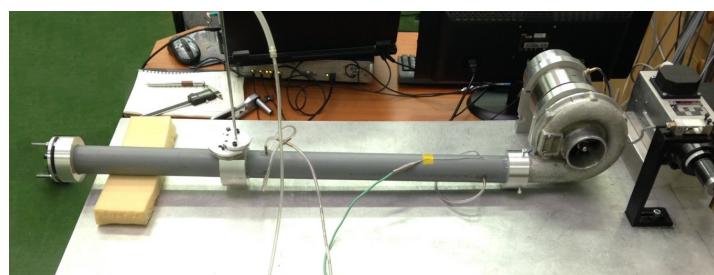


Figure 5: Test rig-blower mounted with a measurement tube.

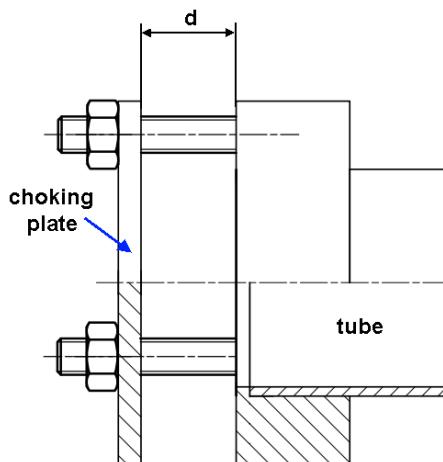


Figure 6: Choke valve design.

drive end of the compressor shaft, a displacement sensor was mounted to measure the axial displacement occurring while the blower was running.

The test was carried out with different loads at different speeds. The speeds varied from 24000 to 30000 rpm. After setting the valve slot width (dimension “d” in Figure 6), the blower was started and quickly run-up to set the number of revolutions per minute. Then, at a constant rotational speed, the data were collected and transformed to voltage signals by proper transducers. The data was processed in the LMS Test Express software.

4 Axial force measurement

Since the thrust bearing behaved as a nonlinear spring and the mounted displacement sensor had unknown characteristics, the axial force measurement equipment had to be calibrated. The calibration setup is presented in Figure 7. A force gauge was attached to the shaft impeller hub. Axial load (simulating the axial force appearing during the operation of the compressor) was generated with a screw to obtain a smooth increase in the axial force.

The chart in Figure 9 was generated by collecting the data simultaneously from the displacement sensor and the dynamometer while the force was applied.

After constricting the chart to the operational conditions (1.2–1.5 V), it became possible to fit its linear characteristics to obtain a formula describing it (Figure 10).

With this data, it was possible to correlate the voltage signals from the displacement sensor to the force values.

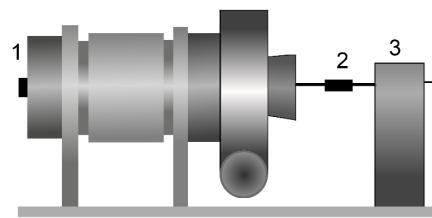


Figure 7: Test rig for the displacement sensor calibration 1- displacement sensor, 2- force gauge, 3- load screw.



Figure 8: NDE thrust foil bearing with nonlinear stiffness characteristics that was loaded during the blower operation.

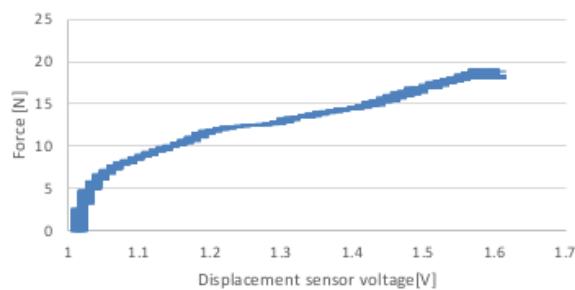


Figure 9: Characteristics of the displacement sensor.

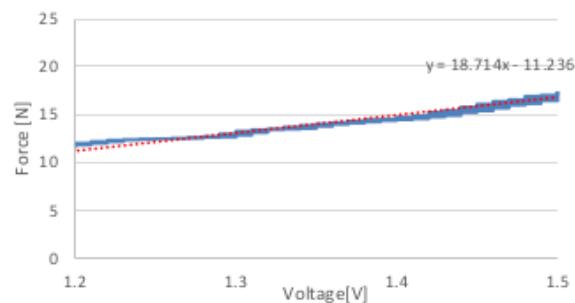


Figure 10: Linear region of the displacement sensor.

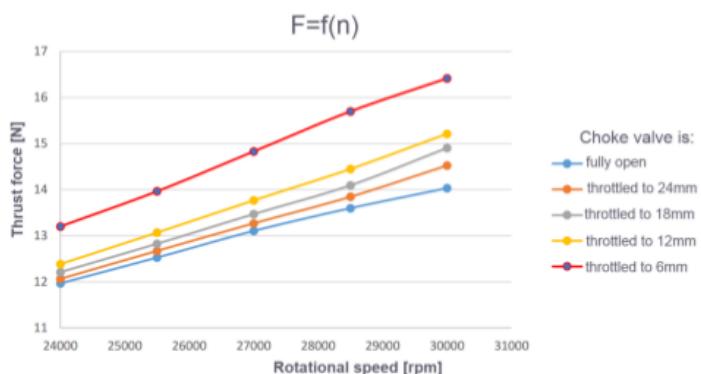


Figure 11: Force values measured during the test at different rotational speeds and different choke valve slot widths.

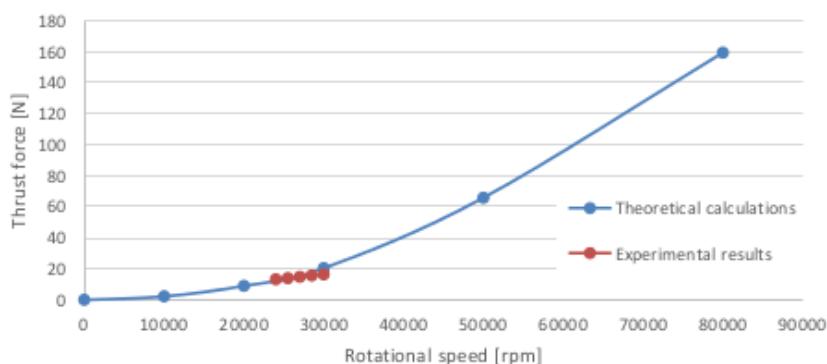


Figure 12: Theoretical and experimental axial force.

Figure 11 presents force values for the speed range from 24 to 30 krpm at different setups of the choke valve.

5 Comparison of the results

The experiment was carried out in a very narrow range. The minimum speed of 24 000 rpm was limited by foil bearings that can perform correctly only at a high rotational speed. At lower speeds, a continuous gas film would not appear and high friction would destroy the bearings. Therefore, the shafts supported in aerodynamic foil bearings should be run-up as quickly as possible to attain the desired rotational speed in order to avoid destructive friction. On the other hand, the maximum speed of 30 000 rpm was limited by the available frequency of the inverter. Hence, the only comparable speed is 30 krpm. For this speed, a comparison between the results of the calculations and the experimental data are shown in Table 1. Differences between the theoretical and experimental results were negligible.

The measured values of the thrust force were plotted alongside the thrust force characteristics obtained from the

calculations. The experimental values for the most throttled valve were taken for the comparison (red line in Figure 11, 6 mm slot width). Despite a very narrow range of experiments, it is easily noticeable on the chart in Figure 12 that the axial forces resulting from the calculations and the experiments agree with each other.

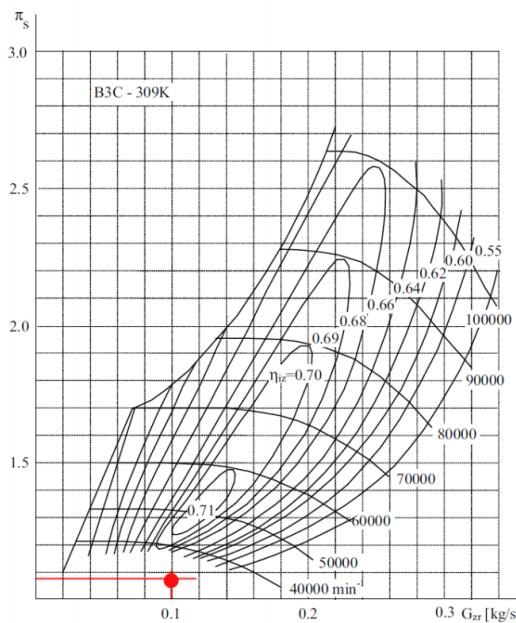
The limitations of the available motor inverter did not allow for testing of the compressor in a full range of operational conditions. Figure 13 presents a performance map for this type of impeller. As one can observe, the current operation point barely falls within the designed ranges of pressure ratios and airflows. In future investigations, it is planned to use an inverter able to drive a PM electric motor up to 60 000 rpm.

6 Conclusions

Since the experimental data are similar to the theoretical results, it can be concluded that the test rig was constructed correctly, and that the results of the experiment are reliable. It means that foil thrust bearings can be successfully used for real-time axial force measurements.

Table 1: Comparison of the theoretical and experimental results at 30 000 rpm.

	Theory	Experiment
Pressure at the scroll outlet	109.6 kPa	107.8 Pa
Temperature at the scroll outlet	302 K	307.5 K
Air density at the scroll outlet	1.264 kg/m ²	1.228 kg/m ²
Pressure ratio	1.101	1.0635
Mass flow	0.084 kg/s	0.0897 kg/s

**Figure 13:** Performance map of the tested compressor with the operating point marked in red [9].

Thus, complex instrumentation is no longer needed. Several pressure sensors can be replaced by one axial displacement sensor. The displacement sensor also delivers real-time information about blade tip clearances between the impeller and the scroll. The only disadvantage is that the displacement sensor needs to be calibrated, which involves a troublesome process.

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