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ATTEMPT OF ASSIGNMENT OF SINGLE-STAGE RADIAL BLOWER PERFORMANCE CURVE BY CFD METHOD

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The article includes comparison of experimental and numerical data of single-stage centrifugal blower performance curve.
The blower characterizing by pressure ratio Π ~ 1.2, has been modernized on the basis of 1D method. A special measurement system in accordance with EN rules was build for experimental measurements. The ANSYS CFX code was used for numerical simulations. Computations were limited to part of a flow path and their results were not agreeable with experimental measurements. Only results of computations of the flow through the complete flow system were more agreeable with the experimental results.

Keywords: centrifugal blower, CFD method

Introduction

Introduced blower belongs to group of low-speed blowers(n = 2950 obr/min), which were produced in Poland in the 80‘ of the XX century. These blowers were characterized a huge diameter of rotor disk (D_{ex} = 650 mm till D_{ex} = 1500 mm) and they were equipped with vaned diffuser. Structural diagram of the typical single-stage radial blower is presented in Fig.1.

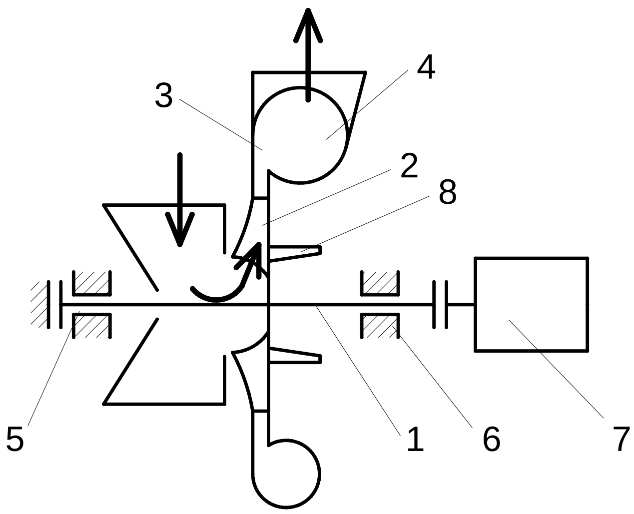

Fig. 1: . Structural diagram of a typical single-stage radial blower.
1 - shaft, 2 - impeller, 3 - diffuser, 4 - spiral aggregate, 5.6 - bearings, 7 - the electric motor, 8 - piston unloader

Relatively simple construction of the blower and relatively low mechanical loading of the low-speed blower allow to get long life of the machine, exceeding even 120,000-,000 hours of work (12-15 years of continuous motion). However, over time the blower flow systems, especially rotor disk are subject to strong corrosion and erosion phenomena what causes the need for their exchange, which in practice means the modernisation of equipment.

2. Radial blower modernization

Blower parameters supplied by the manufacturer are the following:

- Volume flow rate at the suction $\dot{V}$ = 9,3 - 10 Nm^3/s (33500 - 36000 Nm^3/h);
- Static pressure rise Δp = 15,62 kPa (1600 mmH_2O).

The aim of machinery modernisation was to achieve the following parameters of work:

- Volume flow rate at the suction $\dot{V}$ = 11,66 Nm^3/s (42000 Nm^3/h);
- Static pressure rise Δp= 25,506 kPa (2600 mmH_2O).

Conceptual calculations of modernised flow system of the blower were made based on the one-dimensional method, for ambient parameters equal to normal (p_{atm} = 1,01325 bar, T_{atm} = 293,15 K), and assumes that the relative humidity of the suctioned air is φ = 70% which corresponds to the average local conditions, thus assuming a constant gas of air equal to R = 288,84 J/kgK.

Experimental measurements were carried out on the modernized machine. Their results allow to perform the performance curve of the blower. Measurement results confirmed that the aim of the blower modernisation was achieved.

3. Numerical computations

It was decided to perform numerical simulations of the flow through the upgraded machine in order to verify the correctness of modernization.

In order to reduce the size of the numerical task the computation domain was restricted to impeller and diffuser channel. A numerical geometry was developed on the basis of the documentation provided by the manufacturer and documentation developed for the needs of modernization.

The steady calculations were conducted with the code ANSYS CFX using an SST (Shear Stress Transport) turbulence model with an automatic wall function.

The following boundary conditions were assumed in the calculations.

On the inlet:

- total pressure 0 Pa (reference pressure 1 atm),
- turbulence intensity 5%,
- static temperature 20°C.

On the outlet:

- mass flow rate 0,69236 kg/s.

The first step of calculation was to investigate the influence of the mesh size to the obtained results. To this end the calculations for several computational mesh sizes were carried out.

The following mesh sizes were used for the analysis

mesh name	s	m	h	hh	hhh
nodes number	250125	427975	703521	1453680	2053923
elements number	224708	390984	653258	1371084	1950200

The results of this investigation are presented in Figure 2. The measured value of static pressure on the measurement surface located 5 mm behind the diffuser blade is represented by a blue line. As you can see the best convergence was obtained for the mesh named "hh", which contains nearly 1.5 million nodes. Further calculations have continued using this mesh.

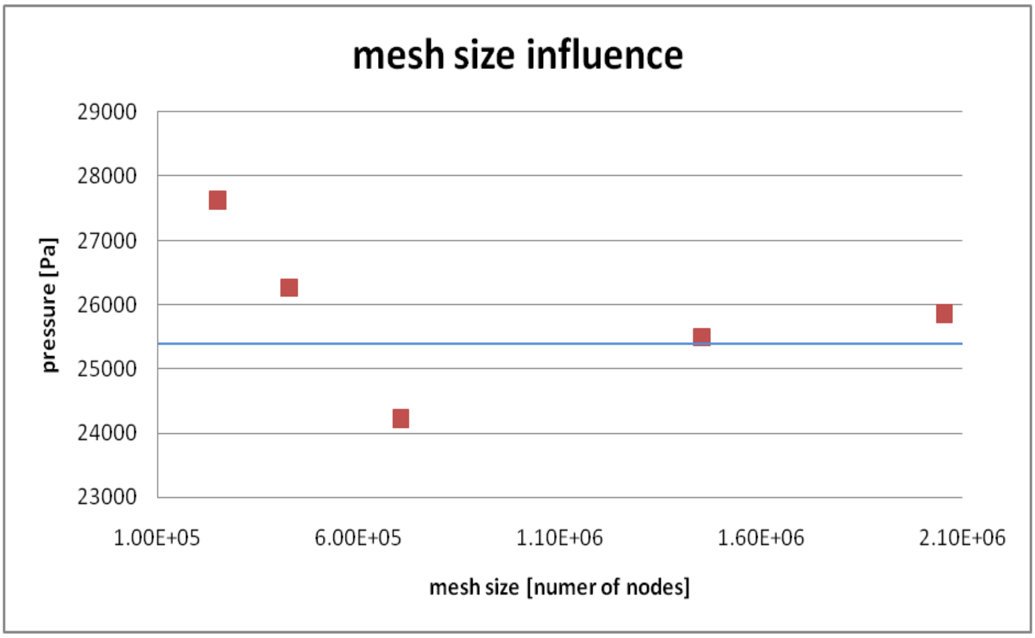

Fig. 2: Mesh size influence.

The steady calculations with two types of transition between the stationary guide vanes and the rotating blades were conducted. The calculations were carried out both for the transition of the ”stage” and “frozen rotor” type. Both the calculations were characterized by high oscillations of pressure values on the control surface behind the diffuser blade.

Due to the high oscillations of pressure values it decided to create the computational domain includes inlet chamber, rotor and diffuser (all passages). The obtained mesh was very large – it consists about 13.7 millions of nodes.

The computational domain of the whole flow system is presented in Figure 3. Black arrows show the inlet surface, yellow arrows show the outlet surface. The control surface behind the diffuser blade is green.

In this case of numerical simulation the following boundary conditions were assumed:

On the inlet:

- total pressure 0 Pa (reference pressure 1 atm),
- turbulence intensity 5%,
- static temperature 20°C.

On the outlet:

- mass flow rate 13.992 kg/s.

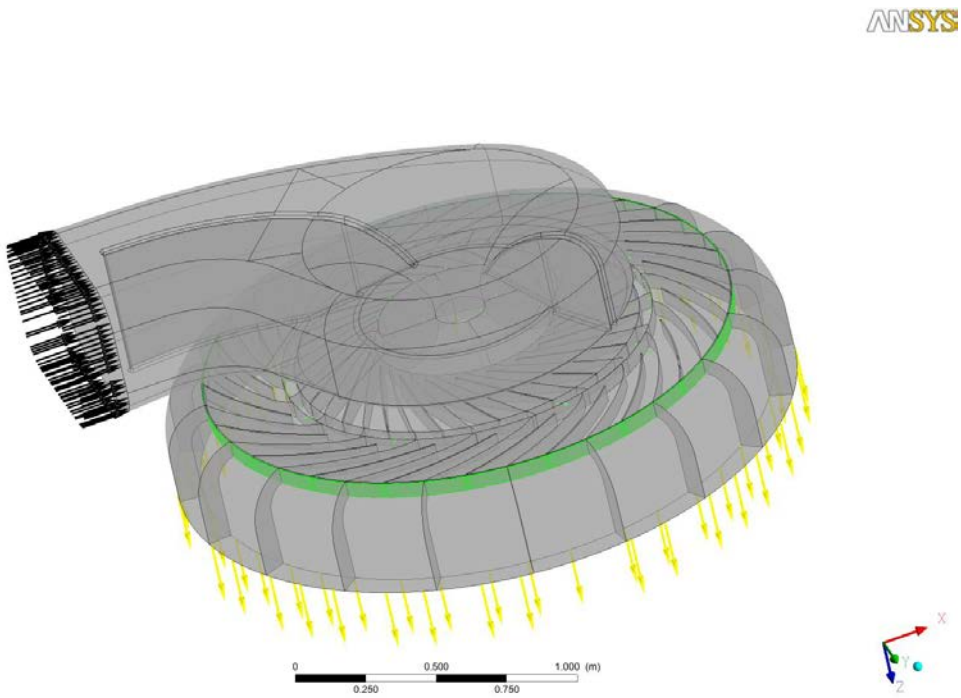

Fig. 3: The computational domain of whole flow system.

4. The results

Figure 4 shows the results of all numerical simulations. They are presented as performance curves. The static pressure is averaged on the control surface located 5 mm behind the trailing edge of the diffuser blade. The measurements were made at the same place. The results of the measurements are shown in Figure 4 also.

As mentions earlier both the calculations of flow through the one passage were characterized by high oscillations of pressure values. However the results of the transition “frozen rotor” are more agreeable with the experimental results then the results of the transition “stage”.

In both cases it is difficult named the results as “performance curve”. The points represented pressure values for following relative mass flow rates not create smooth curve. The performance curve quite similar to the one based on measurements was obtained by simulation of the flow through the whole flow system.

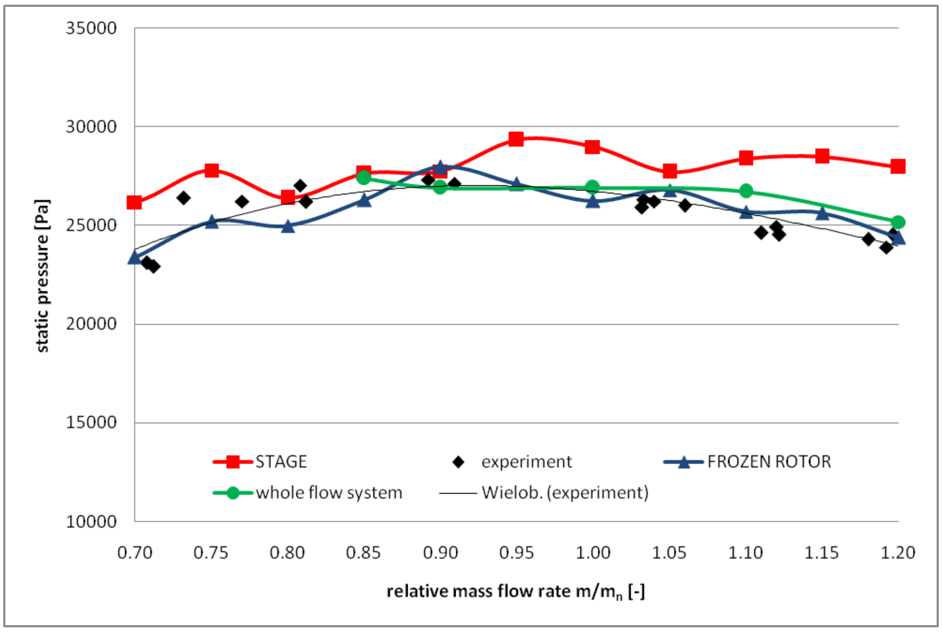

Fig. 4: The results of all computations.

The numerical results of the flow through the inlet chamber reveal a non-uniform velocity distribution at the outlet from this chamber (at the inlet to rotor) (see Figure 5). The velocity value on side closer to the blower inlet was higher than on the opposite side. A region of the higher values takes less then half of the circumference of the chamber outlet. Probably it was the reason of high oscillation of the pressure values obtained from the calculation of the flow through the single passage.

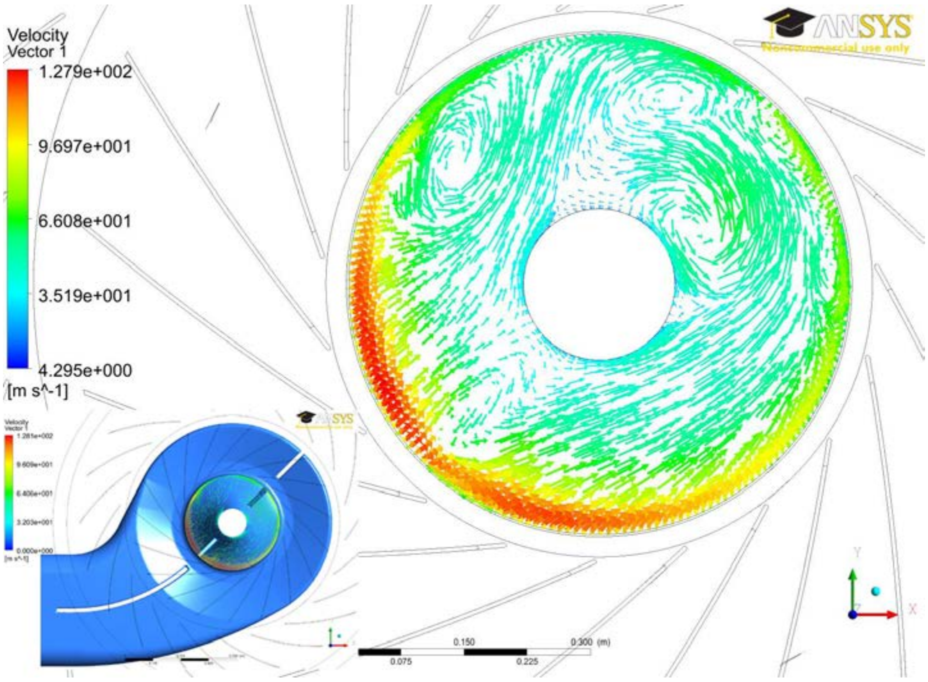

Fig. 4: The velocity distribution at the outlet from the inlet chamber.

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Conclusion

The results of computations of the flow through the single passage of the blower reveal high oscillations of parameter values. It decided to add the inlet chamber to the computational domain to avoid mentioned problem. Of course it extended the numerical task but results of these computations were worth it.

The expansion of the computational domain allows to obtain the velocity distribution at the rotor inlet and it has an influence to results of the flow simulation.

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