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DESIGN AND NUMERICAL AND EXPERIMENTAL INVESTIGATIONS OF THE NON-CONVENTIONAL CENTRIPETAL RADIAL BLOWER FOR SPECIAL PURPOSES

Władysław Kryłłowicz [1], Michał Kuczkowski [1], Waldemar Olczyk [1], Adam Papierski [1]
Institute of Turbomachinery Technical University of Lodz, Poland;
E-mail: adam.papierski@p.lodz.pl

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Abstract

A design and investigations of the single-stage centripetal blower to be used in a vacuum furnace for high pressure gas quenching (HPGQ) technology are presented. The blower task is to cool (a closed-loop cycle operation with nitrogen or helium as the cooling medium under pressures up to 2.5 MPa) steel elements of various shapes at high rate.

Within the project No. 1482/T02/2006 of the Ministry for Higher Education, a blower prototype (denoted as A) was built in the 1:1 scale and subject to aerodynamic investigations. On the basis of CFD calculations (ANSYS CFX code), next blading models (referred to as B, C, D, respectively) were built. A common problem for all blading systems A-D is a very narrow useful operating range and a hysteresis of the performance curve at the surge threshold. Eventually, the final blading system E, characterized by a presence of rear guide vanes of the radial-axial type, which allowed for an elimination of the hysteresis loop and widened a useful operating range, was developed.

The investigations showed a good conformity of the experiment and the CFD calculations.

1. Introduction – aim of designing the centripetal blower

The main field of application of centripetal radial blowers presented in this paper are vacuum furnaces for high pressure gas quenching (HPGQ) technology. In our country, SECO/Warwick from Świebodzin is the leader in devices for heat treatment as it is a manufacturer of vacuum furnaces based on its own technologies, see [1]. In Fig. 1, a cross-section of the HPGQ furnace cooling chamber according to the design developed at SECO-Warwick is presented. It is a two-chamber furnace. The cooling medium (it can be nitrogen, helium, hydrogen or their mixture, under pressure up of 2.5 MPa) circulates in the closed loop cycle that consists of radial blower 1, situated on drive engine shaft 2, cooler 3, system of cooling nozzles 4, and cooling chamber 5, in which batch 6 is placed.

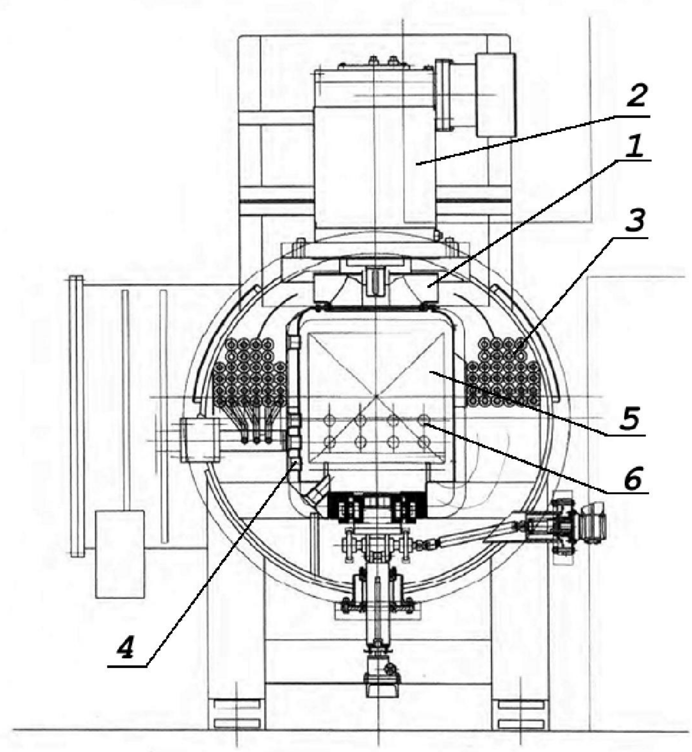


Fig.1. Cross-section of the HPGQ furnace cooling chamber according to the design by SECO-Warwick [1]

As a result of own investigations [2] and experience, a concept to reverse the direction of the working medium has been developed, which will allow for elimination of certain disadvantages from the system presented in Fig. 1. Figure 2 presents a schematic view of the proposed centripetal radial blower. The whole unit of the blower ('Power Pack") is fixed vertically to the furnace casing. The required parameters of the blower have been formulated as follows (working medium – nitrogen):

- suction parameters: pressure 2 MPa, temperature 353 K,
- pressure rise 0.035- 0.036 MPa,
- required volume flow rate 7.5 m^3/s.

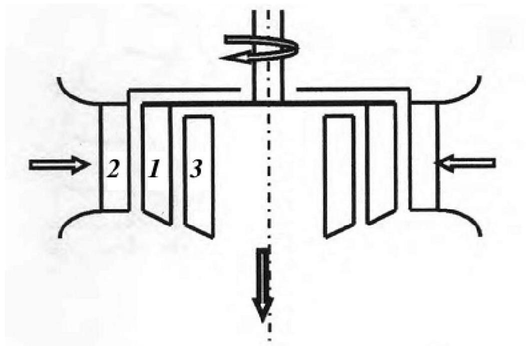


Fig. 2. Schematic view of the centripetal blower. 1- impeller, 2 - centripetal inlet guide vanes, 3 - centripetal outlet guide vanes

2. State of knowledge in the field of centripetal blowers

The theory of centripetal blowers was developed by Prof. Petermann [3], who conducted investigations of a series of geometrical variants of such machines. In Poland, Górnisiewicz dealt with these issues. His dissertation [4] includes both an analysis of Petermann's results and the results of his own experiments. In Fig. 3, a schematic view of the typical centripetal radial blower according to Górnisiewicz with individual diameters and angles marked is presented. The analogous system of notations has been used in the present paper.

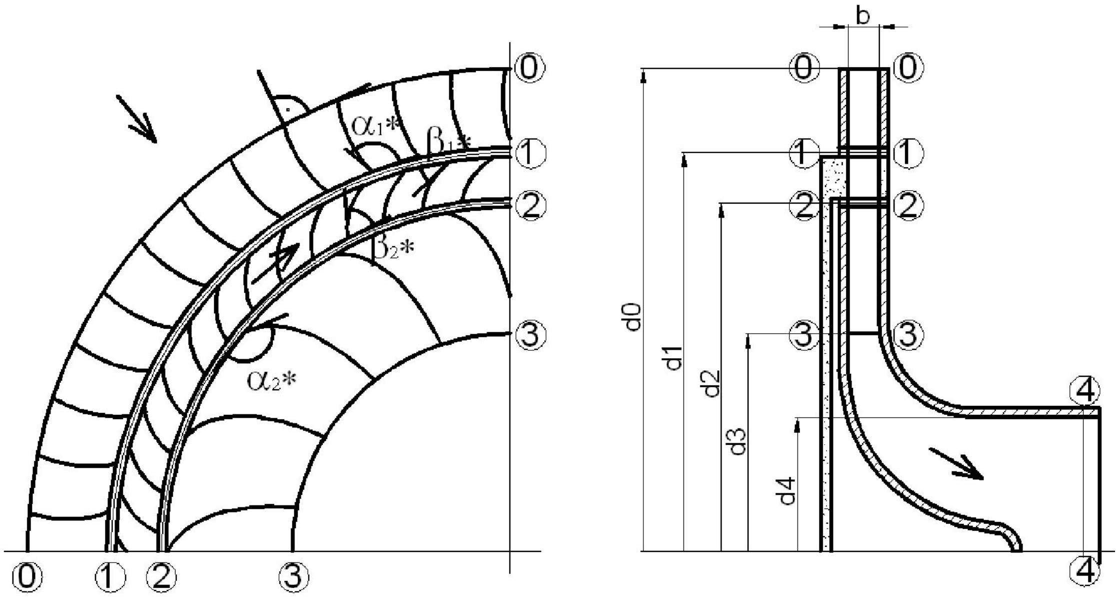


Fig.3. Schematic view of the centripetal blower according to Górnisiewicz [4].

Low popularity of applications of centripetal blowers and fans is followed by a low number of publications devoted to these machines (Schlender [5], Ejiri and Shirakura [6], etc.). All the above-mentioned publications are based on Petermann's concepts. In the light of literature, the common problems of centripetal blowers are as follows:

- low efficiency that does not exceed the threshold of 60% (see [5]),
- low steady range of operation (see [4] and [6]).

3. Results of the former investigations

Within the project No. 4T 07C 029 30 of the Ministry for Higher Education, entitled "Design method of the centripetal radial blower for special purposes", a test stand was built and measurements of four variants of blading, denoted A-D, respectively, were conducted. In Table T1 a dimensional characteristics of the variants under investigation is listed. In all four cases, the outer diameter of the impeller is d_1= 600 mm – the models were made in the actual blower scale. The model blower was driven by an engine of the rotational speed n =1500 rpm which could be adjusted.

Table T1. Geometrical characteristics of the blading variants under investigation

Blading variants		A (E)	B	C	D
d_1	mm	600	600	600	600
b_1	mm	30	30	60	60
β_1^*	°	17	15	15	17
β_2^*	°	82	60	60	82

Photograh1 shows guide vanes and rotor blade cascades of blading variant D.

Photo 1. View of the model blading blower, geometrical variant D.

A typical performance curve of the blower is depicted in Fig. 4. It is an A version blower, operating at the rotational speed of n=1500 rpm. The characteristic feature of bladings A, B and D is an existence of the clearly visible zone of hysteresis, which divides strictly two branches of operation: steady and unsteady. For variant C, steady operation has not occurred at all – the blower goes automatically from centripetal operation to the centrifugal operation state.

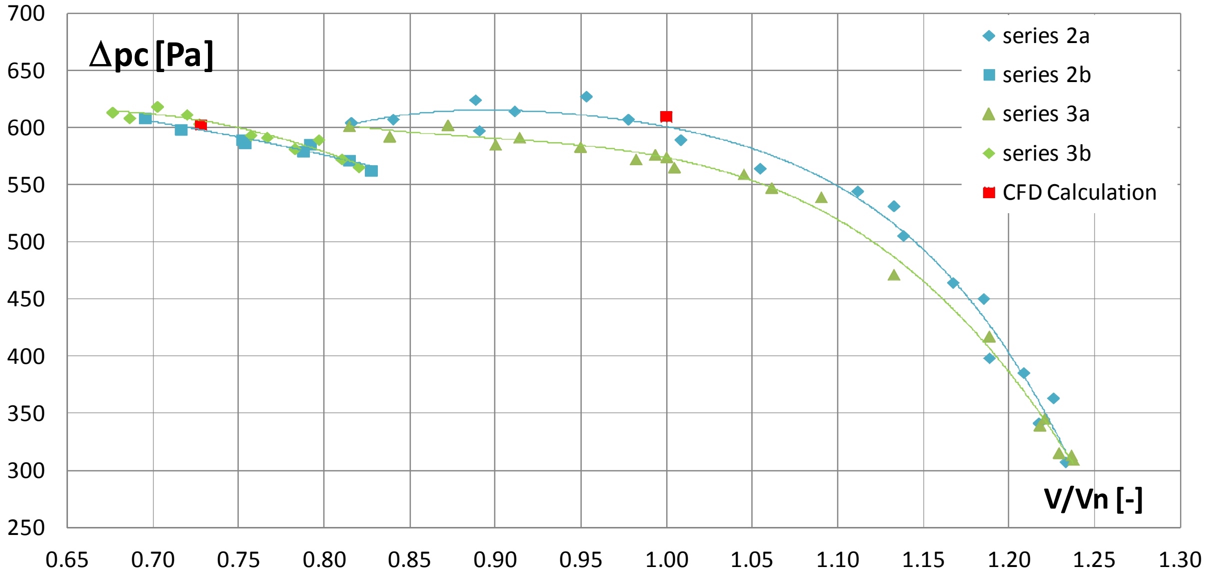


Fig. 4. Performance curve of the model blower, blading variant E

4. Numerical calculations of the target variant (variant E)

This solution has the same blading angles as in variant E. A vaned diffuser that transforms into the classical conical diffuser is mounted behind the outlet guide vanes.

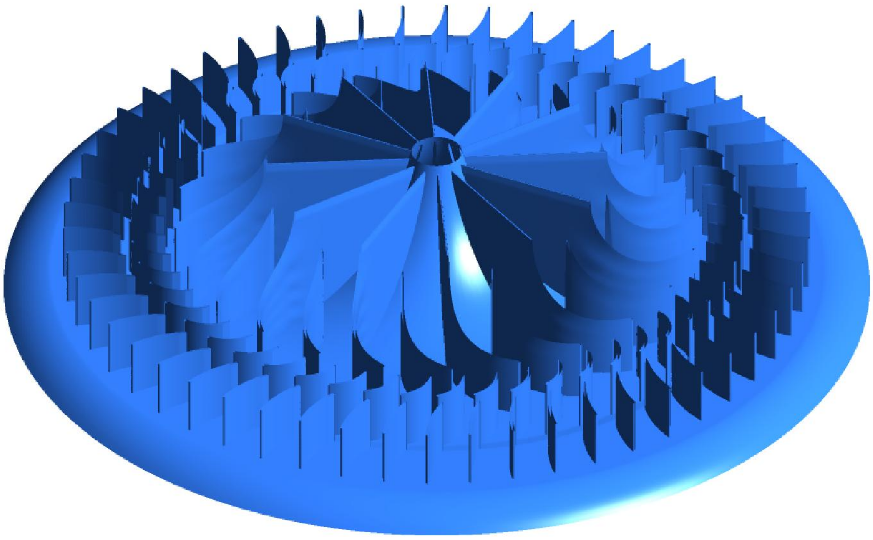


Fig. 5. Geometry of the target variant E

The steady calculations and the unsteady CFD simulations were conducted with the code ANSYS CFX using an SST (Shear Stress Transport) turbulence model.

The application of steady approaches leads to a very high divergence in the results obtained for individual approaches, and which is most important, no solution agrees with the results of the averaged unsteady calculations. It results from the fact of an occurrence of numerous unsteady stall structures in the flow (on the rotor blades and the second guide vanes), whose displacement in time has a considerable impact on the flow conditions in the subsequent guide vanes and channel elements.

4.1 Boundary conditions

The following boundary conditions were assumed in the calculations.
On the inlet interface “S0” (Inlet) shown in Fig. 6:

- total pressure 112 Pa (reference pressure 101300 Pa),
- turbulence low intensity 1%,
- static temperature 20°C.

On the outlet interface “S2” (Outlet):

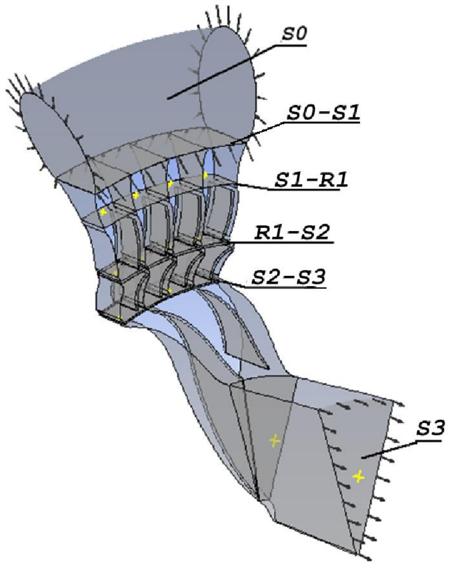
- mass flow rate 2.157 kg/s.

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 an insufficient level of convergence and high oscillations of pressure values on the outlet interface from the blower (S2).

The unsteady calculations covered the relative motion of the rotor blades with respect to the stationary guide vanes in the following configuration:

- 4 channels of the first guide vanes (54 channels for all guide vanes),
- 4 channels of the rotor (52 channels for the whole rotor),
- 2 channels for the second guide vanes and the outlet channel (24 channels for all guide vanes with every second vane elongated towards the axial diffuser).

As a result, besides an insignificant difference in the rotor pitch, a complete agreement of the remaining pitches occurred. The dimension of the mesh for the task formulated in such a way was 2 353 652 nodes.



S0÷S3 – stationary domain
R1- rotating domain

S0 – Inlet,
S0-S1 - Interface
S1-R1 - Transient Rotor Stator interface,
R1-S2 - Transient Rotor Stator interface,
S1-S3 – Interface,
S3 – Outlet

Fig. 6. Boundary surface and the interface

The calculations were conducted with the time step corresponding to the rotor rotation by 1/20 of the pitch (time steps 7.6924 ^{x} 10^{-5} s). This time step corresponds to values of the mean Courant number Cu ≈ 3.1. A reduction of this time step to achieve the recommended values of Cu = 1 is restricted by the time consuming calculations, which is not justified at the present stage of the investigations. The calculations were finished after two full rotations of the rotor t=0.08 s. The criterion for finishing the calculation was determined by establishing the amplitude in alternations of static and total pressures at a constant value in the monitored points P1-P12 shown in Fig. 7.

5. Results of the calculations

The values of total pressures were averaged in time and, on this basis, a static pressure increment in the blower was determined. The results of these calculations are presented on the performance curve of the blower (Fig. 4).

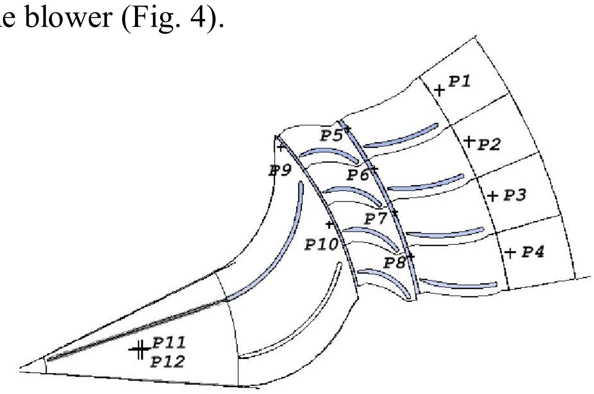


Fig 7. Points in which static pressure alternations were monitored

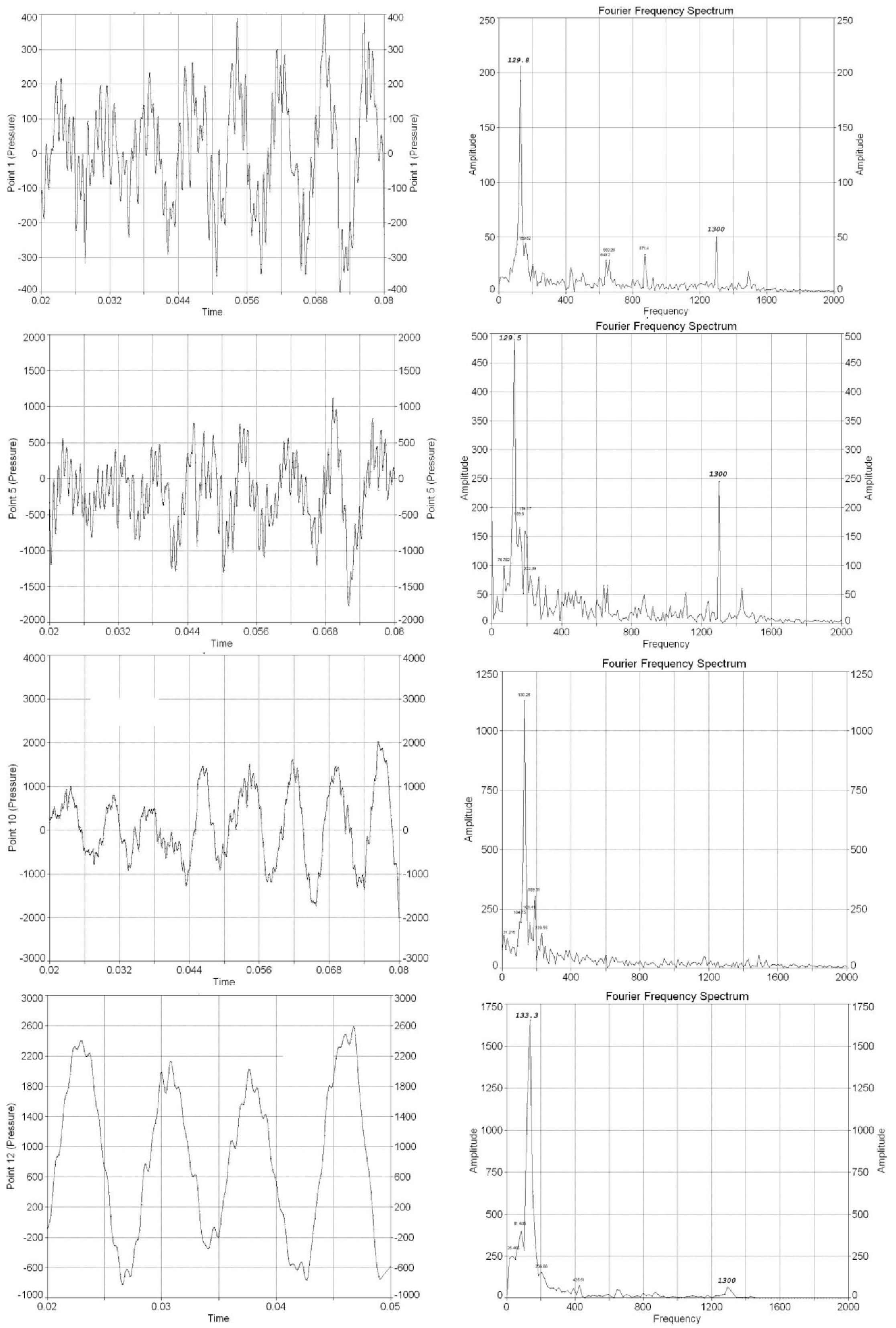


Fig. 8. Pressures in selected points of the blower as a function of time and frequency

Two characteristic frequencies can be observed on the diagrams, namely:

- ~130 Hz – related to the so-called rotating stall,
- 1300 Hz – related to the rotor blade frequency (52x50Hz).

Characteristic stall cells descending from the rotor blade trailing edges with the frequncy of about 130 Hz can be seen seen in Fig. 9.

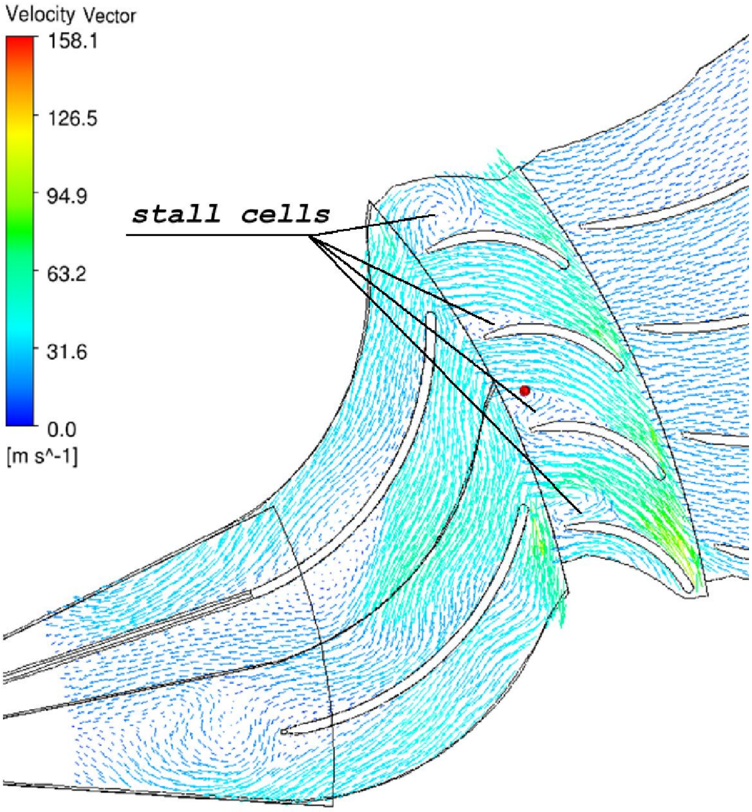


Fig. 9. Velocity field in the central plane of the rotor

6. Conclusions

The variant E (target) blading is characterized by a slightly wider steady range of operation than the previous geometrical variants. The inner efficiency of the variant E blower does not exceed 53%, which is caused mostly by the blade design (bent and welded metal sheet). However, such blades are used in furnaces.

A comparison of the experimental results of the target E blading (Fig. 4) and the CFD results shows a good conformity.

To determine the phenomena occurring in the flow in centripetal blower channels, unsteady calculations should be conducted due to the stall cells that appear in the rotor channels. Despite of the mean value of the total pressure increment in the nominal point of 600 Pa, the blower operation is characterized by high pressure oscillations at the outlet with the amplitude of 1600 Pa and the frequency of approx. 130 Hz.

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