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Velocity Field Calculation and Measurement at the Inlet of Centrifugal Compressor

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Abstract:

The paper is focused on the possibilities of extending the stable operation region of the centrifugal air compressor by modifying its intake system. For various compressor operation regimes and various shapes of the slots, through which the external air enters the main air in the intake system, the flow velocities at the impeller inlet were measured and calculated.

Keywords:

Centrifugal compressor, stable operation region, measurement of velocity field

1. Introduction

In the contemporary gas turbines of small and middle powers the centrifugal air compressors are used most widely. Single stage centrifugal compressors are designed with high pressure ratios up to 9:1 [1]. These compressors have a narrow range of stable operation which is done by the change of mass flow rate from the surge line to the choke at constant rotational speed [2]. For vehicle and aircraft gas turbines the compressor stable operation region is a very important factor influencing first of all their transient behaviour.

Effort to extend operation margin of the compressor characteristics is mainly realized by forming compressor impellers [3, 4]. Generally it is possible to say that the characteristics of centrifugal compressors with impellers having backswept blades have wider operational region then the ones with the straight blades impellers [5].

Other methods how to extend stable operation region of the centrifugal compressor are mentioned in [4, 6, 7]. One of the possible solutions consists in

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arrangement of the velocity field at the compressor inlet. Contemporary centrifugal compressors use specially shaped air flow by-pass channels (internal bleed systems) located at the compressor inlet channel. So the air can overflow from the inducer back into the inlet area (Fig. 1). Air flow through these channels influences the velocity field in front of the compressor.

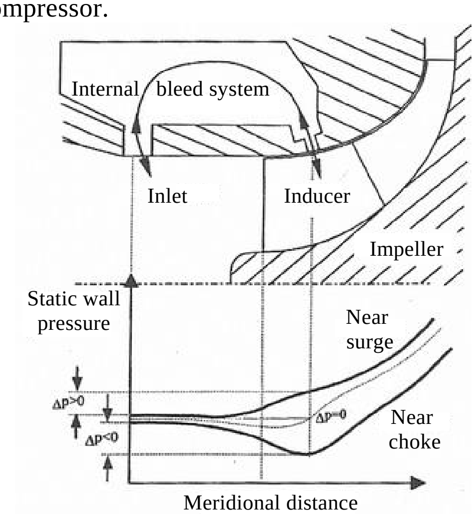

Fig. 1 Scheme of air flow in the bleed system [8]

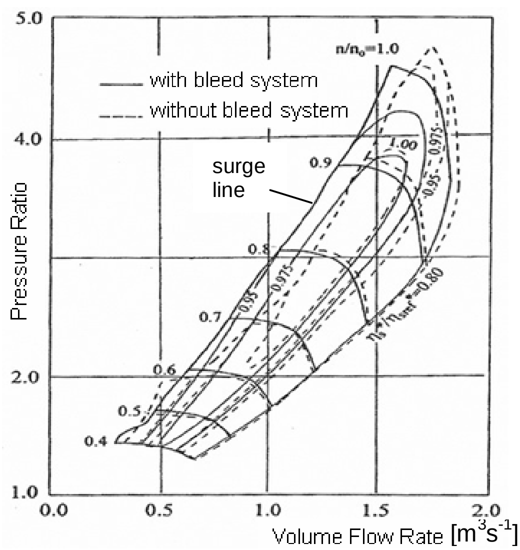

Fig. 2 Compressor characteristics with and without bleed system [8]

Compressor characteristics presented in Fig. 2 show important influence of this bleed system. The surge line was shifted towards lower flow rates by up to 18 %. However, the associated drop in efficiency was considerable and it amounts approximately to 1.5 %.

So as now not enough details are known concerning the actual air flow in this internal bleed system between compressor inducer and its intake system, it was decided to measure and calculate the distribution of the main air flow velocity field in front of the impeller at its various operation conditions.

2. Description of the Test Bench

Air flow velocity field at the compressor inlet was measured in the laboratory of the Department of Aerospace and Rocket Technologies at the University of Defence in Brno.

For the experiment, the inlet system of the small jet engine TJ 100 produced in the Czech Republic was used. So it was possible to utilize directly the obtained measurement results to optimize the design of its centrifugal air compressor intake part.

The test bench contains the following main parts: centrifugal air compressor with its intake and exhaust parts, collecting and air distributing chamber, electric motor, two-piston compressors, pressure air tank, and supply pipe (Fig. 3). Centrifugal compressor is the industrial fan SH 25/109 with backswept blades and vaneless diffuser. Electric motor with variable rotational speed and power (from 10 to 30 kW) drives the impeller. Compressor air mass flow rate was controlled by the rpm change of driving electromotor and by adjusting the throttling of the exhaust valve.

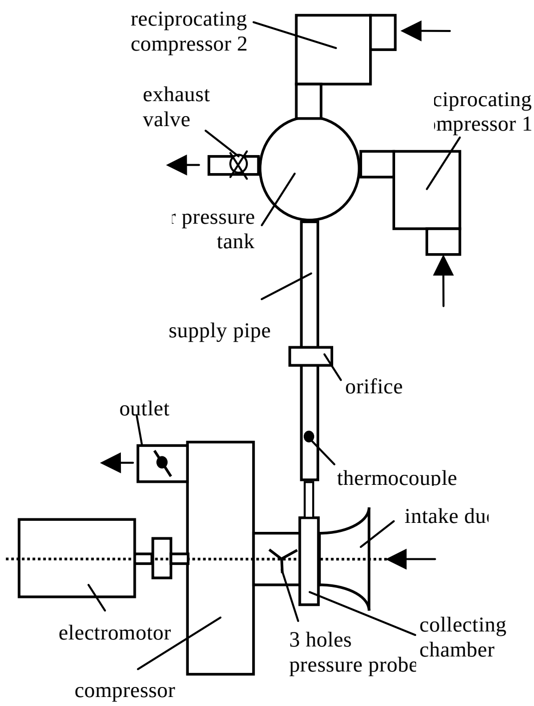

Fig. 3 Scheme of the test bench [9]

TJ 100 intake system contains two parts. The first one is the convergent part with two ribs. The second one is cylinder intake channel which continues to the impeller casing. For the measurement purposes, a collecting chamber with a slot simulating the compressor internal bleed system (Fig. 4) was inserted between these two parts. External air was supplied into the collecting chamber from the air pressure tank by the help of nine tubes uniformly distributed around the circumference [10]. Collecting chamber also serves as a space where the external intake air is stabilized. The collecting chamber consists of two parts. Slot supplying air flow from the collecting chamber into the main air stream of compressor inlet part is defined by the angles b_1 and b_2 (Fig. 5). The collecting chamber was designed in four versions with different angles b_2 from initial value of 20° to 30°, 40° and 90°. Angle b_1 was of constant value of 20°. Particular versions are presented in Fig. 5.

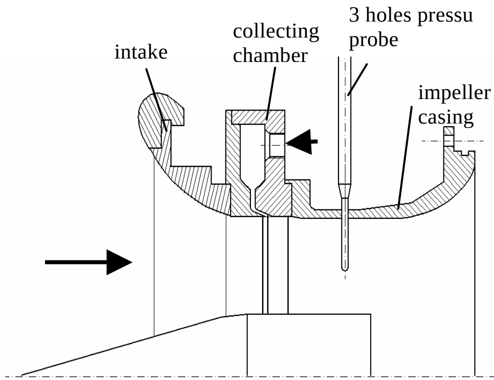

Fig. 4 Scheme of the intake system with collecting chamber [9]

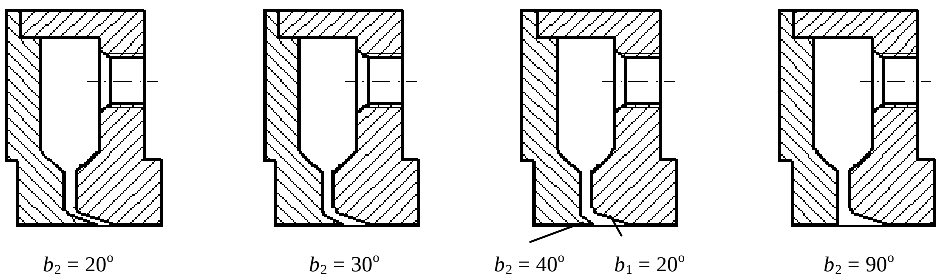

Fig. 5 Versions of collecting chamber with slot

The three holes' pressure probe (Fig. 6) was located in the compressor casing at position just in front of the TJ 100 impeller. In the same plane, at four positions and uniformly around the casing circumference, the static pressures were also measured. Motion of this probe in radial direction was arranged by the sliding equipment with scale.

3. Measurement of the Velocity Field in the Intake System

Measurements of the velocity field in the intake system of the jet engine TJ 100 were performed by setting of 3 holes' pressure probe (Fig. 6) in radial direction step by step from the casing wall to its axis (Fig. 4). At first the velocity field was measured, on radius *r*, in the intake system only, without collecting chamber. This has later on allowed evaluating the influence of external correcting air flow on the distribution of the main velocity field. After these starting measurements the collecting chamber was installed and the velocity fields for various mass flow rates of both basic and correction air flows were measured step by step [9, 10]. These measurements were then repeated also for the various design arrangement of the collecting chamber slot (its angle b_2 was changed).

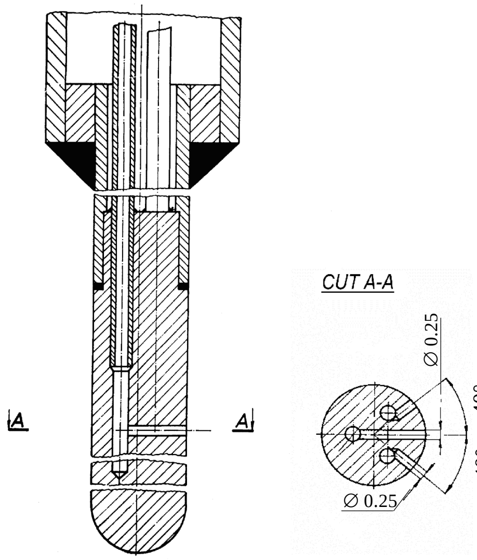

Fig. 6 Scheme of 3 holes pressure probe [9]

The three holes' pressure probe was connected with the same pressure sensors as in the case of calibration measurement (the measurement range ±1 bar and accuracy ±0.25 %) (Fig. 6). Static pressures at the wall of compressor casing and in the collecting chamber were measured by the same pressure probes. Electric signal from the probes was processed by LabVIEW software [9, 10].

By air discharge from the pressure tank the value of overpressure was set up. So, simultaneously the external (or correcting) air mass flow rate to the collecting chamber was also set up ($\dot{m}_p$). The main compressor air mass flow rates were approximately set up at 3 same values ($\dot{m}_c$). All measurements were done for the collecting chamber having gradually following four different values of slot shape angles: b_2 = 20°, 30°, 40°, 90°. The external air inlets without prewhirl in to the intake channel, it means that in inlet stream circumferential component of velocity was not created.

More than 90 measurements for various configurations of the intake system with the external air mass flow rates were performed [9]. The velocity field changes of the main air stream were observed from the point of view influence of the relative mass flow rate ($\overline{m} = \dot{m}_p/\dot{m}_c$). Simultaneously the influence of pressure drop Δp was observed which is defined as the difference between the static pressures in the collecting chamber and at the external diameter of the intake casing. In order to compare particular factors on the velocity field distribution, the respective maps are presented relatively, i.e. as the ratio of the local velocity (c) to the velocity in the middle of velocity field (c_m). At first, the measurements with setting of the collecting chamber slot at b_2 = 20° were performed. After evaluation of these ones the next measurements were performed at the following configurations of collecting chamber slot with angles b_2 = 30°, 40° and 90°. All obtained measurements are stored on the respective CD and elaborated results are presented in the research report [9].

4. Results of Measurements

The first measurements performed with the slot of collecting chamber having b_2 = 20° should have proved the influence of relative mass flow rate $\overline{m}$ and pressure drop Δp on the velocity field distribution. Velocity fields are shown in Fig. 7. For various compressor modes of run and various pressure drops Δp present relatively big changes. But it was necessary to prove the influence of pressure drop Δp on the velocity field. At approximately the same relative mass flow rates and substantially different pressure drops Δp, the measured velocity fields were compared in Fig. 8. From this map it is evident the small influence of Δp on the distribution of velocity filed. On the base of these results, concerning the significant influence of the relative mass flow rate $\overline{m}$, there were performed the measurements with another configurations of collecting chamber at the slot angles b_2 = 30°, 40°, and 90°.

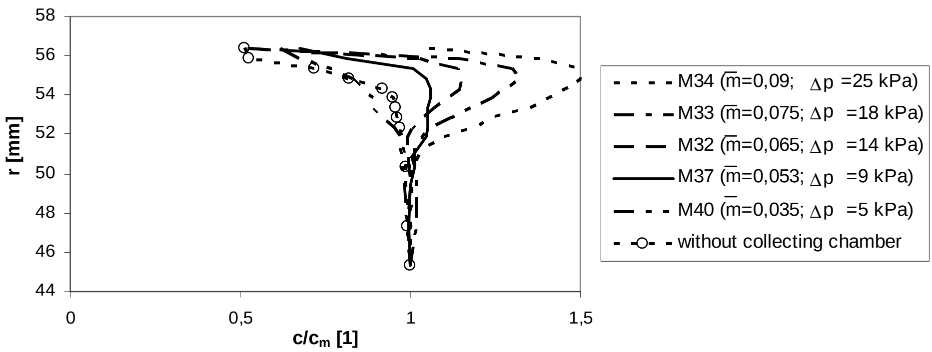

Fig. 7 Influence of relative mass flow rate on velocity field for angle b_2 = 20° [9]

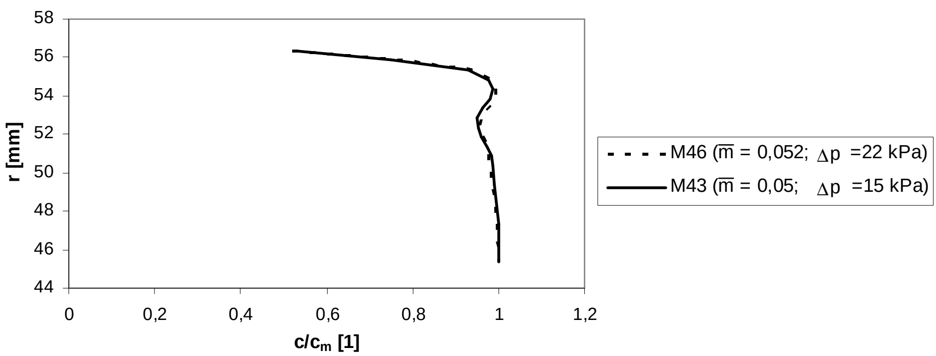

Fig. 8 Influence of pressure drop Dp on velocity field for angle b_2 = 20° [9]

Similarly, at the configuration of the collecting chamber with the slot angle b_2 = 20°, the influence of the pressure drop Δp on the velocity field at the angles b_2 = 30°, 40°, and 90° was also proved. Velocity fields presented in [9] show the small influence of Δp on the velocity field also at the angles b_2 > 20°.

In order to compare the influence of angles b_2 on the distribution of velocities at various relative mass flow rates ($\overline{m}$), some of the measurements at different levels of $\overline{m}$ were selected. The results are shown in Fig. 9 ($\overline{m}$ = 0.090 ÷ 0.103) and in Fig. 10 ($\overline{m}$ = 0.044 ÷ 0.046). From these maps it is evident that the influence of angle b_2 on the velocity field is substantially higher at greater relative mass flow rates (see Fig. 9) than at the smaller values of $\overline{m}$ (see Fig. 10).

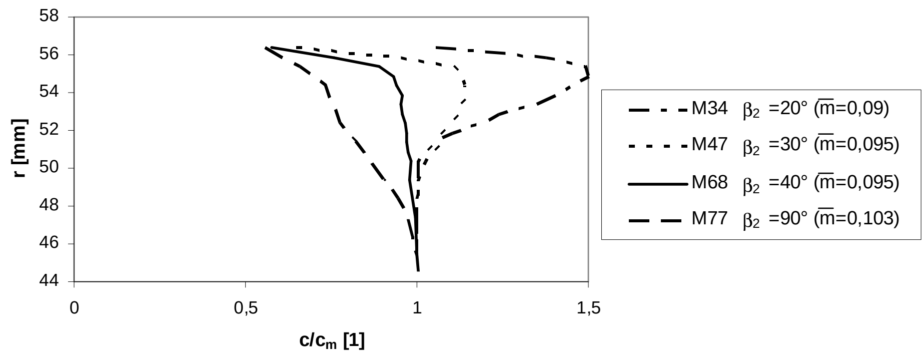

Fig. 9 Influence of angle b_2 on velocity field for higher relative mass flow rate [9]

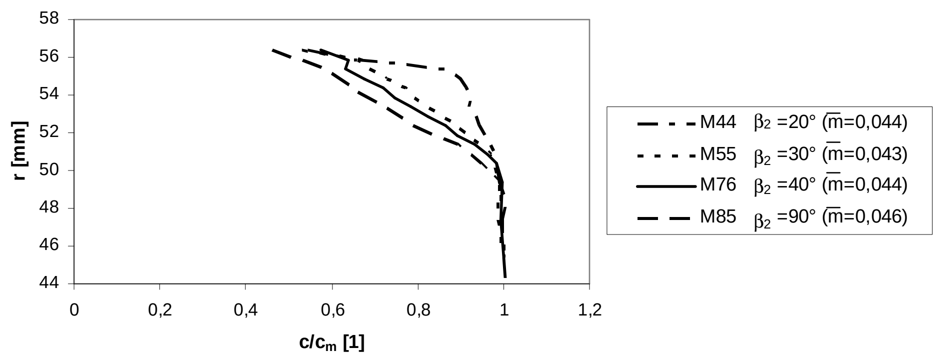

Fig.10 Influence of angle b_2 on velocity field for lower relative mass flow rate [9]

5. Calculations

5.1. Calculation of 2D Model

Model of channel geometry for CFD software FLUENT was created in postprocessor GAMBIT as well as computing mesh. Geometrical model is not complicated therefore the structure mesh was selected [11]. Operating values and boundary conditions were used as the ones obtained by measurements. Hydraulic diameter of intake channel is 0.3 m. 2% turbulence intensity was chosen. Fluid density was calculated for ideal gas. Type of calculation was defined as coupled and implicit with stable flow. Model of turbulence was used *k-e* [12].

Calculated maps of axial velocities for b_2 = 20° show large changes for different mass flow rates similarly as the measurement results (see Fig. 11). Influence of the angle b_2 changes on the velocity field shape obtained by calculations is approximately the same as these obtained by measurements (see Fig. 12).

Calculations of 2D numeric model were performed for most of the measured modes, i.e. for each one angle b_2 (20°, 30°, 40° and 90°). Presented velocity field distributions in the next figures show air flow in the whole inlet channel with slot angles b_2 = 20° (CM34) (Fig. 13) and b_2 = 90° (CM77) (Fig. 14). For the slot angle b_2 = 90° external air penetrates more deeply into the main air flow than for b_2 = 20°.

Calculations were also performed for 3D model which was solved in 1/9 channel's cut [12]. Results of these calculations are very similar to the 2D model [13].

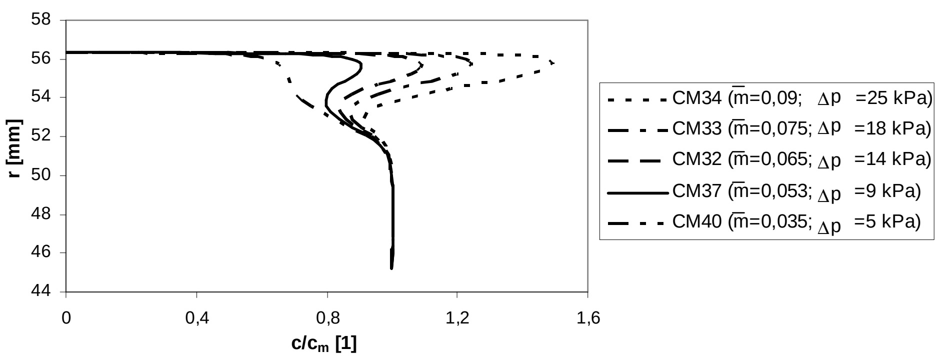

Fig. 11 Calculated velocity field distributions for slot with angle b_2 = 20° [13]

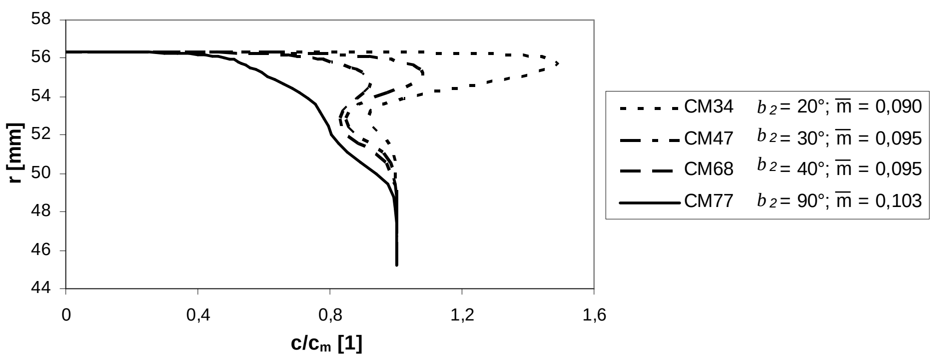

Fig. 12 Calculated velocity field distributions for different slot angles b_2 [13]

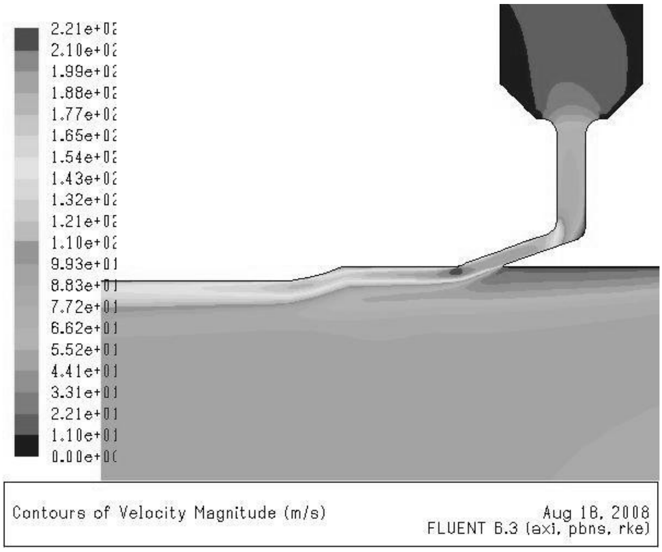

Fig. 13 Velocity field distributions with slot angles b_2 = 20° (CM34) [13]

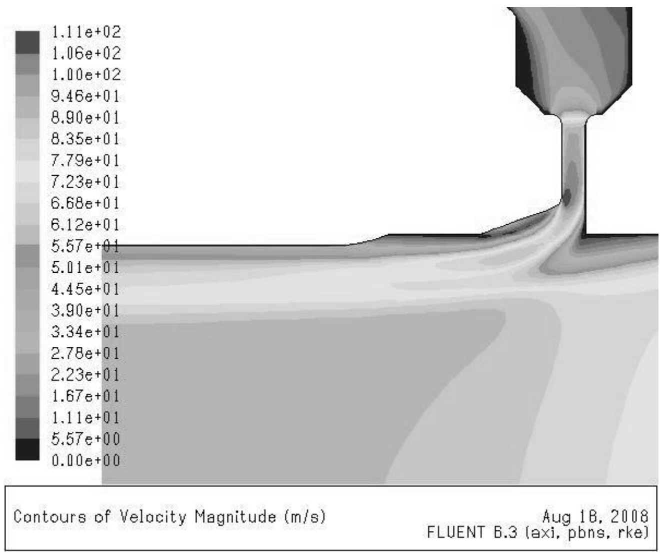

Fig. 14 Velocity field distributions with slot angles b_2 = 90° (CM77) [13]

5.2. Comparison of CFD Calculation with Experiment

If we compare the cases for b_2 = 20° (Fig. 15) and b_2 = 90° (Fig. 16) it is evident that at the calculation (sign CM) the main stream is influenced by the external air in the narrower layer near the wall than of the measurement. Similar differences are shown also for angles b_2 = 30° and 40° [9]. It is necessary to notice that the calculations for higher mass flow rates $\overline{m}$, for all angles b_2 (except b_2 = 90°) give axial velocities in narrow layer near the wall expressively grow, then suddenly fall down and subsequently grow up again. In measurements, this "wave" exists seldom, only at b_2 = 20°. At calculation this "wave" is obviously caused by worse mixing of streams.

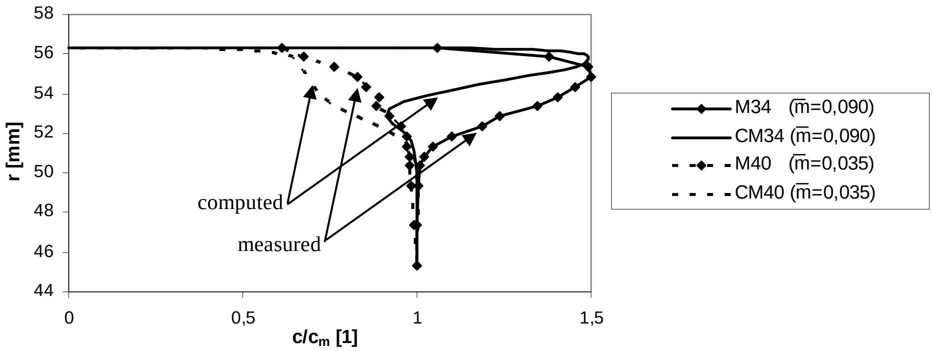

Fig. 15 Comparison of calculations and measurements for b_2 = 20° [13]

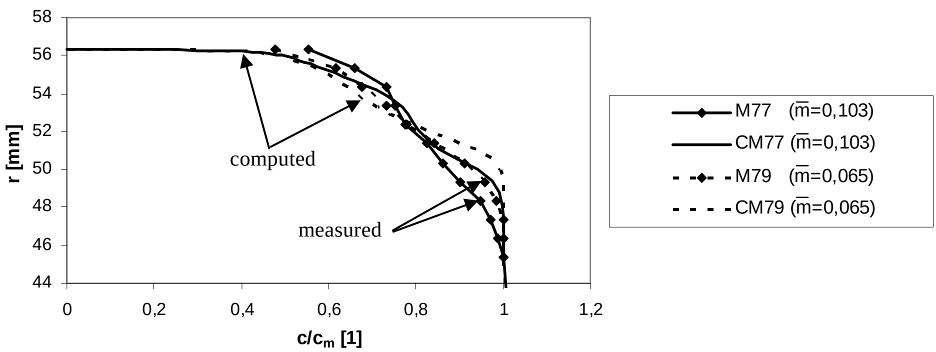

Fig. 16 Comparison of calculations and measurements for b_2 = 90° [13]

Results of CFD calculations are influenced by simplifying presumptions first of all by using of ideal channel geometry and also roughness of duct surface is not considered. Results of calculations can be also influenced by inaccurate setting of initial and border conditions and especially by using of turbulence model with appropriate coefficients.

On the other side, the results of measurements are also influenced by dimensions of probe which penetrates to the air stream from external to internal duct diameter. Intake channel is not ideally smooth and two ribs in it also disturb the stream. Nevertheless it is possible to state that calculated and measured velocity fields are at all angles b_2 very similar.

6. Conclusion

Measurements and calculations confirm fundamental influence of the channel shape defined by the angle b_2 on distribution of velocity field at the intake external diameter in front of the impeller. Significant influence of relative mass flow rate $\overline{m}$ on velocity distribution near the external diameter of compressor inducer was proved as well. It has been shown that the pressure drop (Δp) between the inlet air, measured in collecting chamber and static pressure, measured on the wall of intake channel, has not significant influence on the distribution of velocity field, when the relative mass flow rate $\overline{m}$ is a constant.

It has been proved that can not be expected substantial grow of air velocity near the inducer external diameter, when the external air flow is perpendicular on direction of the main stream (b_2 = 90°). On the contrary, by decreasing the angle b_2 (b_2 < 90°) velocity field is significantly changed and at the same time is considerably influenced by $\overline{m}$.

Our measurements and calculations prove that centrifugal compressor designed with bypass channel during its various operations the axial air velocities near the external intake diameter will be changed. Contemporary small changes of air velocities in other parts of intake system will occur.

In connection with bypass channel design significant role will be played by the location and the size of withdrawal slot in the screen above the inducer. Position of the withdrawal slot influences the pressure of air in the withdrawal place and thereby also the flow through the bypass channel. From the results of calculations and measurements follows the necessity to design the bypass channel particularly for each compressor.

One of possible solutions could be take off of air from vaneless diffuser which would be supply in to the collecting chamber situated in the intake system as it was design in our experiment (Fig. 5). By automatic control of supplying air, with considering of exhaust angle from collecting chamber b_2, it would be possible to optimize the inletting air flow in front of the inducer blades and by this to improve the air flow in the impeller. This solution is based on the presumption that an improvement of velocity field in front of the impeller is crucial for extension of stable operation region of centrifugal compressor. In this connection it is also necessary to consider energy balance of air entering into the intake system. Air during the flow through the impeller obtains energy, which is partially decreased by the losses in the bypass channel. This adversely affects on the compressor pressure ratio and efficiency.

In order to prove this extension of stable operation it will be necessary to perform calculations of whole centrifugal compressor (intake system, impeller, diffusers). Nevertheless only the experiment can prove how the improvement of velocity field in front of the impeller affects the compressor characteristics.

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