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SECONDARY FLOW MEASURING AT THE RADIAL FAN ENTRANCE

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Abstract – Analyze is given of different measuring methods relating to the prerotation flow in the entrance pipe of the radial pump. The appearance of the prerotation flow is a result of complicated fluid flow model, which appears as a consequence of the pump operating out of design point and reduce the pump efficiency. The goal of this contribution is in estimating the adequate measuring method, taking into account the inconvenience of conventional Laser – Doppler anemometry. Therefore, two measuring methods – multiblade (ASB) and single blade anemometer (ASSB) are introduced, analyzed and compared with well known 2 channel hot wire anemometry (HWA). The advantages of the introduced measuring system - ASB are in its simple construction and simple use and its low price. Direction and the swirl flow intensity in the entrance pipe of radial pumps and fans, using this method, could be measured.

Keywords: prerotation, radial fan, CFD

Nomenclature

b	relative velocity component angle
D	multiblade anemometer diameter
D_e	entrance pipe diameter
j_{GV}	angular orientation of beam axes
f_{nat}	blade's natural frequency
$f_{sampl.}$	sampling frequency
N	number of samples
d	logarithm decrement
A	instantaneous amplitude
x	damping coefficient
x_c	critical damping coefficient
f	frequency
d	diameter
U	flow velocity
Str	Strouhal number
p_{dyn}	dynamic pressure
A_{blade}	measuring blade surface
r	fluid density
F_{dyn}	dynamic force
$F_{st,cal}$	static force, during calibration
g	gravity
m	mass

1. INTRODUCTION

Conventional measuring technique using laser-Doppler anemometry (LDA) is unsuitable when measuring a tangentional flow velocity component which is relatively small compared to larger ones, in our case, axial flow velocity due to problematic flow direction determination. In the case of LDA, the user measures the flow velocity at a given point and doesn't know if the measured value belongs to the main tangentional flow velocity component or is part of some local eddy flow. This last fact led to the idea of developing a new measuring method, which gives correctly measured values but does not greatly disturb the pump/fan entrance flow. This measuring problem occurs when the prerotation flow in the entrance pipe of the radial pump/fan needs to be measured.

When considering the potential flow, prerotation can be described using Euler´s entrance velocity triangles (Fig. 1) at blade leading edge. The real flow through the turbo machines is turbulent since most fluids that cross turbo machines are viscous. The existence of the prerotational flow phenomena in real flows can therefore be proved using a system of Navier – Stokes equations, which are non-linear and must be solved numerically. Therefore the numerical flow simulation results for the under- and over-optimal flow rates for different turbo machine geometries are presented and analyzed in the paper.

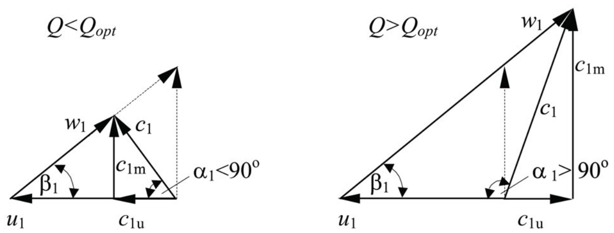

Figure 1: Euler´s entrance velocity triangles

Tangentional velocity component occurs as a consequence of the relative secondary flows acting at the pump impeller eye and in the individual impeller channels [1] and it is the main cause to drive the prerotation flow at the entrance pipe. The magnitude and direction of the prerotation flow depends on the pump/fan operating regime. The prerotation flow propagates up to five entrance pipe diameters (5 De) in an upstream direction of the entrance flow. This phenomenon is also closely associated with cavitation swirl by pump part load operating [2] and with water column separation [3] when considering pumps.

Since the axial flow velocity component (in direction of pipe axis) is much (about 100 times) bigger than the tangentional flow velocity component in practice the anemometer with straight blades (ASB) (Fig. 2a and Fig. 2b) as well as the swirlmeter [4], [1] is often used for tangentional velocity determination. The problem, when using ASB, is flow disturbance at the ASB location that affects the pump operating characteristics in the sense that it decreases the exit pressure and impeller efficiency. To avoid this problem – entrance flow disturbance – an anemometer with a single straight blade (ASSB) can be used (Fig. 3a and Fig. 3b). In this way the area of disturbed flow is decreased to one small section covering less than 1% of the whole entrance flow area in the axial direction. This area is small enough to be neglected.

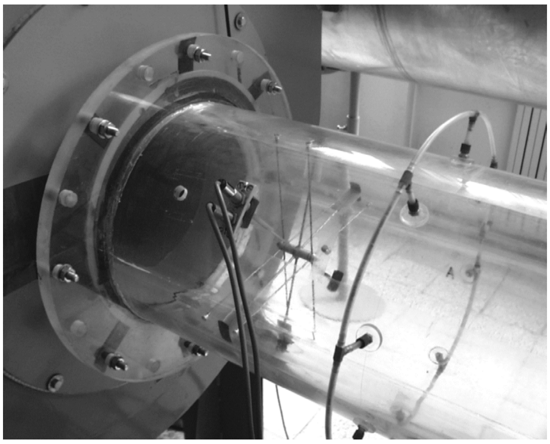

Figure 2a: Anemometer with straight blades

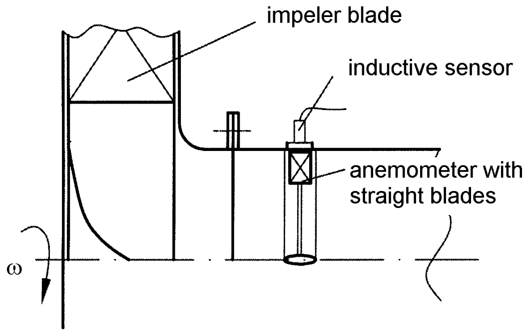

Figure 2b: Anemometer with straight blades

2. MEASURING SYSTEM

The single blade is fixed on a special housing as a cantilever beam (Fig. 3b). The length of blade beam can be changed in the radial direction by up to one half of the intake pipe diameter, and can be twisted around the beam axes in the range of φ_{GV}=0 to φ_{GV}=180 degrees.

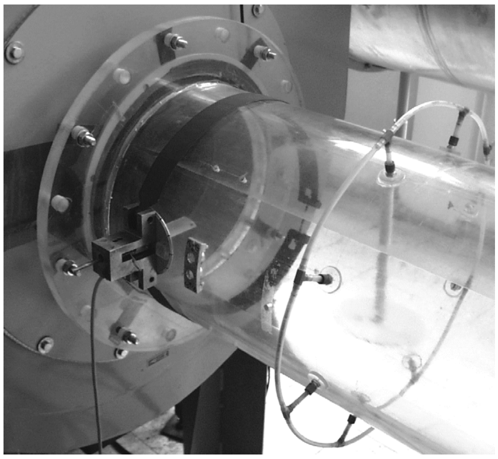

Figure 3a: Anemometer with single straight blade

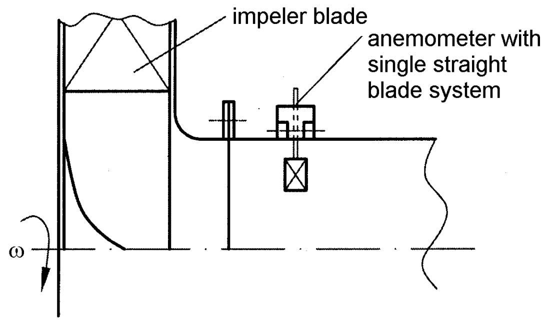

Figure 3b: Anemometer with single straight blade

Four measuring HBM LY13/360 strain-gauge strips are put on the blade beam holder and electrically connected into the full Wheatstone bridge, well known [5]. Temperature equilibrium between the individual strain gauge strips is achieved using this electrically connection. Measuring data is taken by A/D DASPORT PCI-20450P-25 converter, using VISUAL DESIGNER software. The sampling rate is 5 kHz and each measuring point has 2048 measured values.

Measurements were done on a radial fan with a tip diameter of D = 0.6 m. The entrance pipe, made from transparent plexiglas, has a diameter 0.3 m (Fig. 2a and Fig. 3a). ASSB is placed at a distance of one entrance pipe diameter in an upstream direction in front of the impeller eye. The tested fan operated at 2000 rpm with a flow-rate in the range of 0.1 up to 1 m^3/s. Measurements were taken parallel to both measurement systems, using an anemometer with straight blades and with proposed ASSB. Several samples are taken at each measuring point from which the average value is determined.

3. MEASURING SYSTEM CALIBRATION

Measuring system (ASSB) is calibrated statically, using weights, and dynamically, when the measuring blade is put into the fluid flow.

During statically calibration the weights are put in the center of the measuring blade edge when the cantilever beam is in the horizontal position (Fig.4). Both loading directions (as the fluid action is expected) are tested. The measuring system response is almost linear (Fig. 5) and the achieved measuring repetition is very good. Correlation coefficient is 0.999.

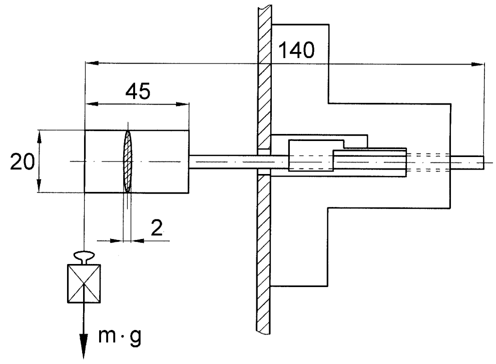

Figure 4: Statically calibration set up

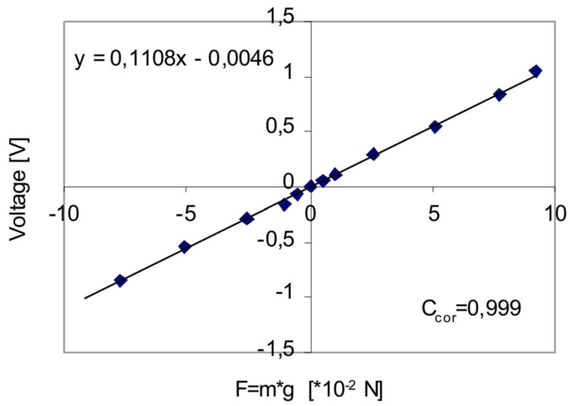

Figure 5: System response by statically calibration

The ASSB blade's natural frequency (NBF) is determined by pulse excitation (Fig. 6a) and calculated from the power spectra where the peak is evident at the 16th sample. From following equation

$$f_{nat} = 16\frac{f_{sampl}}{N} \qquad (1)$$

the natural frequency of ASSB can be calculated. When considering the number of samples (N=2048) and sampling frequency (f_{sampl}=5 kHz) is ASSB natural frequency 39.06 Hz. Practically the same natural frequency value is determined from the power spectra record shown (Fig. 6b), where only one frequency peak is evident and that belonging to the ASSB natural frequency. To check if the ASSB will vibrate during the measuring, the ASSB damping, using logarithmic decrement definition [6], was determined as

$$d = \frac{A_n - A_{n+1}}{A_n} = \frac{2px}{x_c} \qquad (2)$$

where A is temporary (instantaneous) amplitude of natural oscillation and ξ, ξ_c are damping and critical damping coefficient, respectively. The logarithmic decrement, according to [6], for our case is δ=0.075, where ξ=0.00879 and ξ_c=0.736, are damping and critical damping coefficients, respectively. It is evident that the ASSB damping coefficient is very small, when compared to critical damping that means that the blade will oscillate with its natural frequency during measuring. The Strouhal frequency for inclined plate (approximation for the measuring blade of ASSB) can be used for defining flow-induced vibrations. The frequency of flow-induced vibrations can be, according to [7], determined as

$$\frac{f\,d}{U_0} = \mathrm{Str} \rightarrow f = \frac{\mathrm{Str}\cdot U}{d}, \qquad (3)$$

where f is a frequency, d is diameter of plate, U is flow velocity, and Str is Strouhal number. The Strouhal number is in the range of 0.15 up to 0.8, [8], for the inclined plate, where d become the plate length and for plate thickness 0.1d. In our case, flow-induced vibration frequencies could be expected in the range between 60 to 1600 Hz at flow velocities from 1 up to 5 m/s. From this fact could be concluded that all vibrations, caused by flow, will be superposed on the natural vibration of ASSB. This means that in measured signal is dominant natural frequency, what is confirmed during dynamical calibration and during measurements.

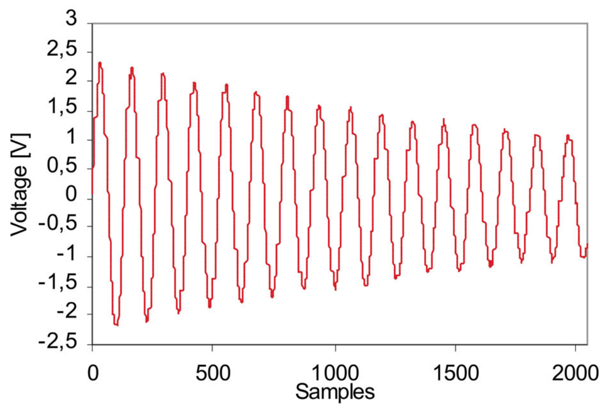

Figure 6a: ASSB system natural oscillation

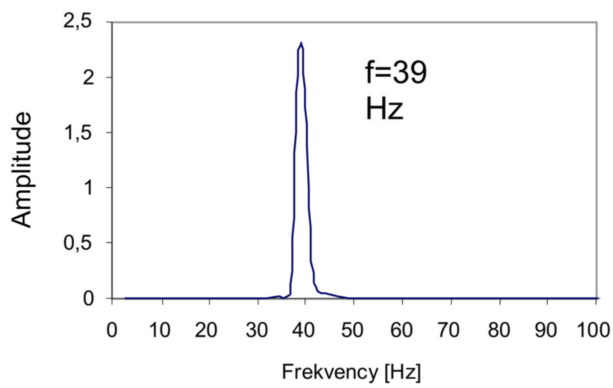

Figure 6b: ASSB system natural oscillation power spectra record

A radial fan with a square – shaped channel at the exit where the measuring blade is placed (Fig. 7) is used for ASSB dynamical calibration. Guiding blades are positioned in the rectangular shape channel. The flow is therefore guided to be theoretically almost perpendicular to the measuring blade which is placed into the centre of the channel facing perpendicular to flow.
The angle of the measuring blade is changed during calibration to determine the sensitivity angle regarding to flow direction. The ASSB response at changed flow velocities and angles is evident from the calibration results (Fig. 8). Almost linear response can be concluded in the area from 70 up to 110 degrees of blade position to the flow. This means, that linear calibration characteristic can be considered by measuring at a small flow attack angle.

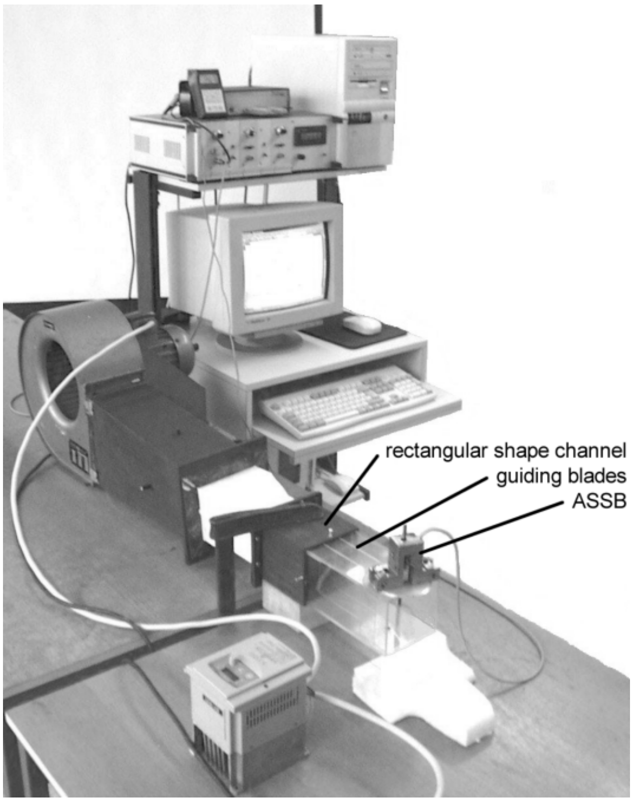

Figure 7: Measuring set up during dynamical calibration

When calibrating the comparative results some non-linear effect between static and dynamic calibration is evident, especially at larger flow attack angles to the ASSB blade. The reason for this can be found in the flow nature and in the flow conditions near the measuring blade. The force is related to squared flow velocity in the sense of determined dynamic pressure at the measuring blade surface

$$p_{dyn} \propto \rho \frac{U^2}{2} \propto \frac{F_{dyn}}{A_{blade}}, \quad F_{dyn} \propto A_{blade} \cdot \rho \frac{U^2}{2} \qquad (4)$$

where ρ is flow density, F_{dyn} is dynamic force, caused by air flow, and A_{blade} is the measuring blade surface in a normal direction to acting flow.

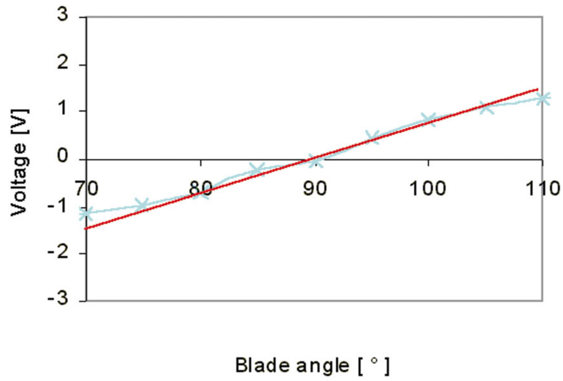

Figure 8: System response at dynamical calibration

In this way the connection between calibrated static and calibrated dynamic force can be found. However, dynamic flow force acting to blade surface is a square function of flow velocity and this fact must be considered when measuring with ASSB. The connection between the static calibration force and dynamic calibration force is

$$F_{st,cal} = m\,g \propto F_{dyn,cal} \propto K \cdot U_{cal}^2, \qquad (5)$$

where K is a constant that depends on blade geometry, air density and flow effects. This constant can be determined from both calibration data.

4. MEASURING RESULTS

The averaged results are given in the diagram forming Fig. 9 at flow-rates in the range of 0.2 up to 0.8 m^3/s. Linear course is evident in the range between 0.3 up to 0.8 m^3/s when using an anemometer with straight blades (multiblade anemometer), and non-linear using ASSB. The cause of this effect is probably in the squared functional dependency of the flow velocity at ASSB as mentioned previously.
With ASB, ASSB and HWA comparison, can be concluded, that all methods could be used for prerotation estimation. It is evident, that point of zero prerotation coincide (ASB and ASSB) with the minimal measured tangentional velocity with HWA method.

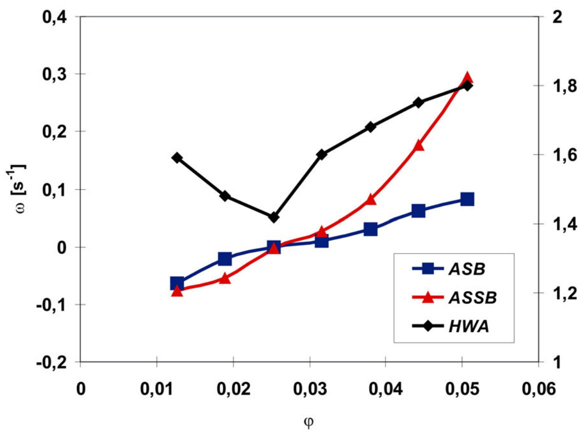

Figure 9: Measuring results

The main problem when using HWA is in the additional cooling of the probe wire (parallel to the wire axes) which results in a bigger measured tangentional flow velocity value comparing to the actual. From the course of HWA in the diagram on Fig. 9, it is obvious, that both presented methods (ASB and ASSB) are suitable for prerotation direction definition also. The unique advantage of 2 channel HWA measuring system (Fig. 10) is in simple connection to inlet pipe (taking to pieces is not necessary).

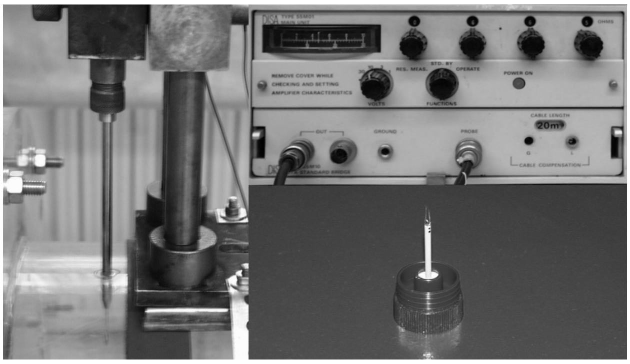

Figure 10: HWA measuring system

5. NUMERICAL SIMULATION

Using CFX numerical system we tried to model the prerotation flow at the impeller entrance and analyze the prerotation flow phenomenon. For impeller geometry definition and mesh generation we used BladeGen CFX module. BladeGen+ module, included in CFX BladeGen gave us the opportunity to calculate flow conditions in the single impeller channel with simple (zero equation) turbulent model. Since BladeGen+ do not support connection of motionless intake pipe with moving (rotating) pump impeller, we used the CFX 5.5 numerical code, as solver. We started with single impeller channel and with the intake pipe sector (45 degrees pie), which suits to the 8 blade pump impeller.

The vector plot of the circumferential component velocity at the distance of 50 mm in front of the impeller eye is given on Fig. 11. Both graphs (Fig. 11.a and b) show the flow velocity distribution calculated using two equation turbulent model (k-e) for minimal (Fig. 11.a) and for maximal (Fig. 11.b) flow rate through the impeller. It is evident that the prerotation flow direction is equal to the measuring results at minimal flow rate. At maximal flow rate the deviation from experimental values is evident, since the direction of prerotation flow in the intake pipe does not change its direction to the direction that is opposite to direction of the impeller rotation. From this reason we recalculated same pump impeller geometry with SST (Shear Stress Transport) turbulent model, based on (k-w) equation system for fluctuating quantities of flow field description. From the numerical simulation results, given in Fig. 12, it is evident, that turbulent model change does not influence the flow direction significantly. This is the reason why we continued our simulations with full intake pipe cross section (up to 360 degrees). Calculation results are given at Fig. 13. From these plots the deviation of results is evident again, because the prerotation flow direction doesn't change its direction at higher, over-optimal flow rates. From both records shown on figures 13.a and 13.b is evident that the flow field in pipe is not uniform. The reason for this could be found in the fact that in the simulation was made only for one impeller channel. To solve this problem, we added seven (7) impeller channels, and so constructed complete impeller. With this change the calculation time was increased for over 15 times with respect to the calculation time needed for the calculation with single channel.

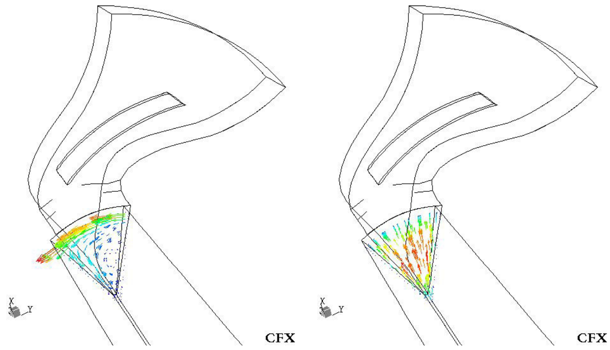

Figure 11: Vector plot of tangentional velocity component in intake pipe sector of radial fan (k-ε)

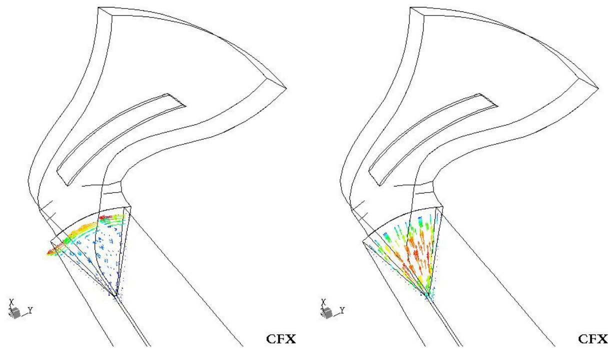

Figure 12: Vector plot of tangentional velocity component in intake pipe sector of radial fan (SST)

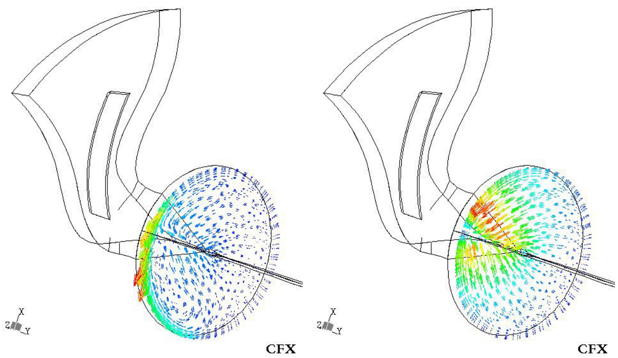

Figure 13: Vector plot of tangentional velocity component in intake pipe of radial fan (k-ε)

The results of complete impeller simulation for minimal and maximal flow rates are given at Fig. 14.a and 14.b. The good agreement with measured results is evident only at small that means minimal flow rates. At larger flow rates bigger disagreement with measured results is obvious. From this fact we can conclude that this numerical approach is not suitable for prerotation flow modeling.

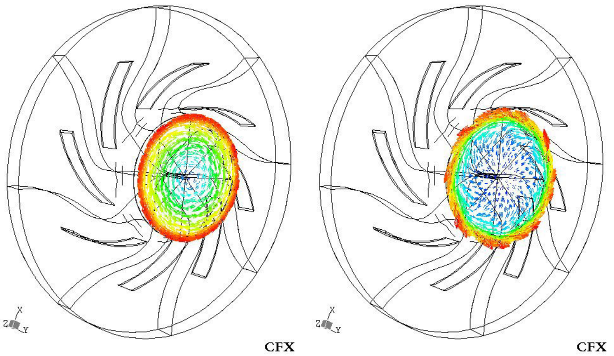

Figure 14: Tangentional velocity component distribution in front of radial impeller (k-ε)

Alternative approach could be found in different interface model. In presented examples the simplest and therefore the fastest "Frozen rotor" interface was used. The second, "Stage" interface, is unsuitable also, since it is necessary to use ring instead of full inlet pipe. From this reason it is impossible to get insight into flow conditions at small pipe diameters, where cavitation swirl occurs [9] when pump operates in unstable cavitating [10] operating regime. "Transient Rotor – Stator" interface is, as follows from the name, meant for transient phenomena analysis which was not the goal of this contribution

6. CONCLUSIONS

The presented ASSB can be used as a simple measuring device for the first approximation measurement of the prerotation flow intensity measuring. It gives good results in the sense of the flow prerotation direction-course measuring and less in the sense of the exact measuring of the absolute flow velocity values of the prerotation flow intensity measuring. The ASSB advantage is in its simplicity, small flow disturbance during measuring, and in the fact that is "low-cost" measuring device in comparison with the multiblade anemometer measuring system or other similar measuring systems. The calibration of ASSB is unpretentious and its priority is in simple use while only the weights are needed. The test results prove that the linearity of the measuring method is not so well as by using ASB measuring system. The reason of this non-linearity could be the squared functional dependency of measured force that depends of the flow velocity (flow dynamic force). The repetition of the measuring results are good with small disperse of the measured values.

Different CFD turbulence models have no serious influence on the velocity profile in intake pipe. Regarding to presented impeller – intake pipe configurations, used numerical simulation approach does not give applicable results.

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