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VELOCITY MEASUREMENTS DOWNSTREAM OF THE IMPELLERS IN A MULTISTAGE CENTRIFUGAL BLOWER

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ABSTRACT

The paper presents the results of an experimental investigation on a four-stage centrifugal blower, having the aim of obtaining an accurate description of the flow field behind the impellers in several operative conditions and for different geometrical configurations. Actually, the test plant allows to change the turbomachinery characteristics assembling one, two, three or four stages and three different types of diffusers.

In this first research step, the blower has been tested in the four-stage vaneless diffuser configuration.

The unsteady flow field behind the impellers and in the diffusers has been measured by means of a hot-wire anemometer. A Phase Locked Ensemble Averaging Technique has been utilised to obtain the relative flow field from the instantaneous signals of the stationary hot-wire probes.

Several detailed measurements sets have been performed using both single and crossed hot-wire probe, to obtain the velocity vectors and turbulence trends, just behind the blower impellers and in several radial positions of the vaneless diffusers. These measurements have been done at different flow rate conditions, covering unsteady flow rate phenomena (rotating stall) too.

The results obtained allowed to get a detailed flow field analysis in the multistage centrifugal blower, in relation to the geometrical configuration and to the differing operating conditions.

NOMENCLATURE

b	diffuser axial span
C	absolute velocity
c	fluctuation of the absolute velocity
D	diameter
$\dot{m}$	mass flow rate
N	number of signals acquired in each point
P	shaft power
PS	pressure side
R	radius
SS	suction side
T	non-dimensional velocity fluctuation intensity $= [\Sigma_N c^2/(N-1)]^{0.5}/C$
Tu	non-dimensional tangential Reynolds stress component $= [\Sigma_N u^2/(N-1)]^{0.5}/C$
Tv	non-dimensional normal Reynolds stress component $= [\Sigma_N v^2/(N-1)]^{0.5}/C$
Tuv	non-dimensional shear Reynolds stress component $= [\Sigma_N uv/(N-1)]/C^2$
t	circumferential blade pitch at the impeller exit
U	impeller peripheral velocity
u	velocity fluctuation component tangential to average velocity
v	velocity fluctuation component normal to average velocity
W	relative velocity
W_s	specific work
x	axial coordinate
y	circumferential coordinate
α	absolute flow angle related to tangential direction
β	relative flow angle related to tangential direction
η	efficiency= $\dot{m} W_s P^{-1}$
λ	power coefficient = $P\rho^{-1}\omega^{-3}D_2^{-5}$
ρ	air density (inlet conditions)
φ	flow coefficient = $\dot{m}\rho^{-1}\omega^{-1}D_2^{-3}$
ψ	pressure coefficient = $W_s\omega^{-2}D_2^{-2}$
ω	impeller angular velocity [rad/s]

Subscripts

2	impeller exit
r	radial

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INTRODUCTION

The detailed study of the flow field in centrifugal turbomachines is essential to an exact understanding of the links existing among the geometrical characteristics of the machine, the flow within the same and its performances in order to be able to enhance the machine's project.

Alongside with the application of improved Computational Fluid Dynamics Codes for the solution of flow in turbomachines, a great importance is still to be attached to the development of experimental measurement techniques, both with the aim of testing the results of CFD codes and to describe the functioning of these machines in detail.

The dramatic developments in the field of computers in the recent years have brought significant improvements in the experimental measurement techniques used to this aim, both in terms of measurement instrumentation and of data acquisition and data processing systems; at present large quantities of data can be collected and processed to an extent that was previously unimaginable.

Apart from direct internal flow measurement techniques within the rotating blade channels, other conventional techniques are based on stationary probes which are positioned just at the impellers' in- or outlet and in the turbomachines diffusers.

The equipment extensively used to measure the instantaneous velocity field include Laser Anemometer (Fradin and Janssens 1990, Rohne and Banzhaf 1990, Flack et al. 1987, Eckardt 1976, Krain 1981) and Hot-Wire Anemometer (HWA), in particular to study the flow in pumps (Flack et al. 1987), often tested with air (Ubaldi et al. 1992, Jaberg and Hergt 1989), fans (Cau et al. 1987, Ray and Swim 1981), blowers (Höfler et al. 1988, Jansen 1964, Kinoshita and Senoo 1985, Jiang et al. 1972) and compressors with measurements limited to the flow at the impellers outlet (Olivari and Salaspini 1975, Ishida et al. 1989, Ligrani et al. 1983, Dickmann 1972) or in vaned diffusers (Fradin and Janssens 1990, Inoue and Cumpsty 1984) or in vaneless diffusers also (Eckardt 1975, Maksoud and Johnson 1989, Frigne and Van Den Braembussche 1984, Kämmer and Rautenberg 1986).

Different techniques have been applied using HWA with stationary probes in centrifugal turbomachines:

— single-wire probes with fixed positioning in order to obtain the velocity's intensity (Jansen 1964, Dickmann 1972, Frigne and Van Den Braembussche 1984, Kinoshita and Senoo 1985, Jiang et al. 1972, Ubaldi and Zunino 1990);
— single-wire probes with several subsequent positioning in order to assess two velocity components (Olivari and Salaspini 1975, Eckardt 1975, Höfler et al. 1988, Jaberg and Hergt 1989, Ishida et al. 1989, Ray and Swim 1981, Inoue and Cumpsty 1984) or three velocity components (Ubaldi et al. 1992) and associated turbulence components;
— two-wire X probes with fixed positioning for the assessment of a bi-dimensional velocity field (Ligrani et al. 1983);
— two-wire X probes with several subsequent positioning in order to assess a tri-dimensional velocity field (Cau et al. 1987);
— three-wire probes in order to assess the three velocity components and the six turbulence components simultaneously (Maksoud and Johnson 1989).

Clearly, each of above-mentioned techniques presents some advantages and some disadvantages (Lakshminarayana 1981), that have to be evaluated to find the most suitable solution.

This paper deals with the experimental measurements performed on a very interesting centrifugal turbomachine model as it offers the possibility of varying its geometrical configuration (number of stages, varying from one to four, and three different configurations of the diffusers).

Since this is a multistage machine it offers the possibilty of getting a detailed insight of the flow field in the different stages and of comparing these data in different operating conditions.

The aim of this research is to describe the flow within the machine in great detail and possibly to link its characteristic to the particular geometrical configuration and performance.

EXPERIMENTAL PROCEDURE

Test Rig and Operating Conditions

Experiments were carried out at the Laboratory of the Dipartimento di Energetica - University of Trieste on the modular centrifugal blower. In this investigation the four-stage set was used; all the stages are identical geometrically, with 16-bladed shrouded impellers and vaneless diffusers with parallel straight walls (figure 1). More geometrical features are shown in table 1.

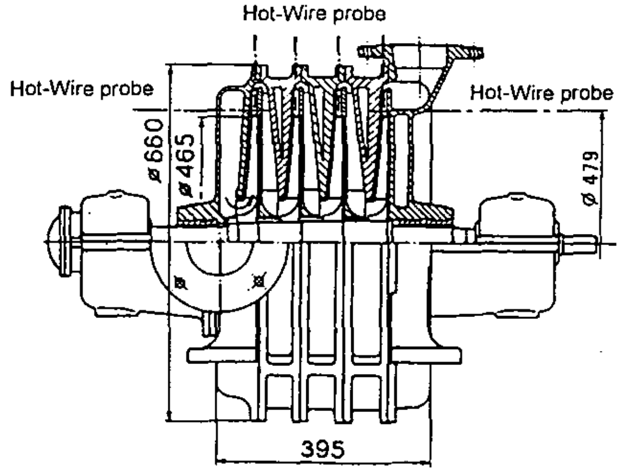

Fig. 1 - Test blower and measuring sections.

IMPELLER		
inlet diameter	160	mm
outlet diameter	465	mm
inlet axial span	23	mm
outlet axial span	8	mm
inlet blade angle	35	°
outlet blade angle	67	°
number of blades	16	
DIFFUSER		
inlet diameter	467	mm
outlet diameter	570	mm
axial span	10	mm

Tab. 1 - Geometrical data.

The impeller was driven by a DC motor coupled to a speed increasing gear. The blower operated in open circuit: the air flow entered through a radial inlet pipe.
The blower characteristics for the design rotational speed of 3000 rpm are shown in figure 2 and the test conditions are pointed out. The design condition is: flow coefficient φ =0.0065, pressure coefficient ψ=0.44, power coefficient λ =0.0085, efficiency η=0.33 (gear mechanical losses and lubricating power included). The flow is stalled for flow coefficient less than 0.0040. The measurements were performed at four stable and four unstable conditions (table 2).

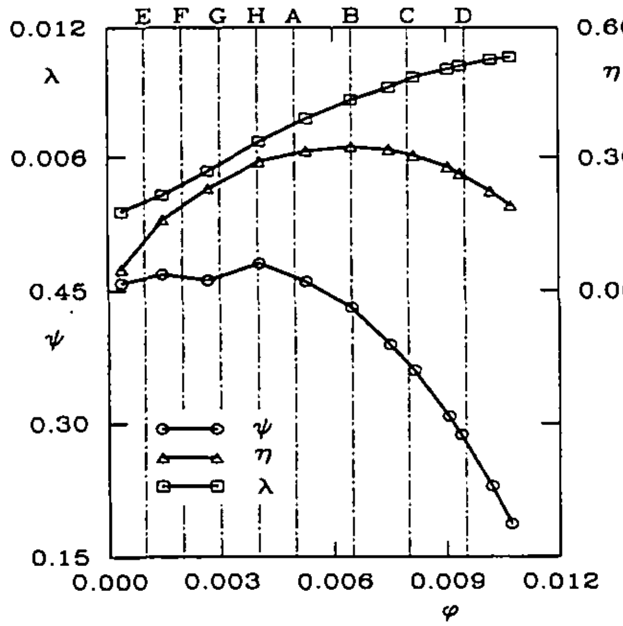

Fig. 2 - Performance map of the test blower.

	E	F	G	H	A	B	C	D
φ	0.0010	0.0020	0.0030	0.0040	0.0050	0.0065	0.0080	0.0095

Tab. 2 - Investigated operating conditions.

Instrumentation and Measuring Technique

Mass flow rate was measured by an orifice flow meter and controlled by a throttling valve in the discharge pipe. Pressures were measured by variable inductance transducers, temperature by type K thermocouples, torque by a load cell and rotational speed by a magnetic pick-up indicator.

Flow measurement was performed by a constant temperature hot-wire Dantec anemometer, both with single (55P11) and X-wires (55P62) straight miniature probes. The sensors were tungsten wires, of 5 µm diameter and 1.25 mm length. Two hot-wire units (55M01) with standard bridge (55M10) were used and a low-pass filter at 10 kHz was applied to cut high frequency noise, the base frequency being 800 Hz (blade passage). The hot-wire instantaneous voltage signals were sent to an analog/digital converter board. The phase reference was taken off by the pick-up device through a time base that provided the trigger signal. The data acquisition system (HP6900) was controlled by a HP9000 computer (figure 3).

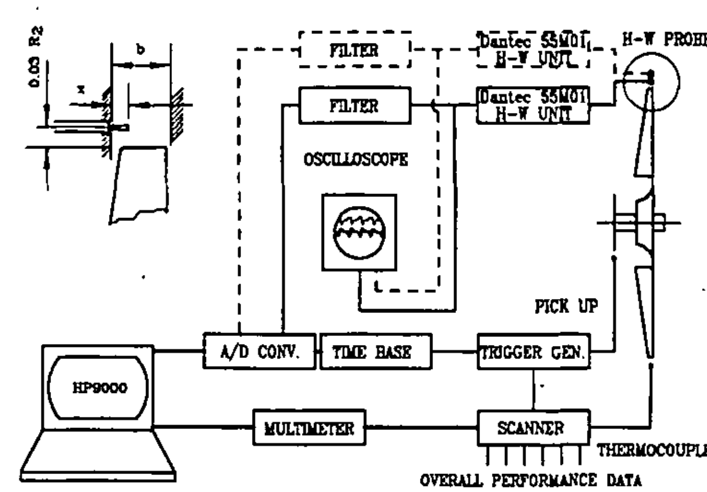

Fig. 3 - Schematic of experimental rig.

Every probe was calibrated in a Disa 55D45 unit using a technique that was described in detail by the authors in a previous paper (1993). By composing King law with a parabolic directional sensitivity law a calibration surface (voltage in function of velocity intensity and flow angle) was obtained for each wire.

The coefficients of the calibration surface were corrected in order to take into account the temperature effect, according to Collis and Williams model (1959), with the fluid properties evaluated according to Morrison (1974).

After calibrating probes a check was done by sample stationary flows (velocity from 60 to 80 m/s, flow angle within the 40° allowable calibration zone): mean errors equal to 0.75% on velocity and 1° on angle approximately were found with maxima (1.9% and 2° respectively) located at the borders of the angular range.

A stationary hot-wire technique for rotor exit flow measurement was used. The Phase Locked Ensemble Averaging Technique (Lakshminarayana and Poncet 1974, Lakshminarayana 1981) was adopted for stable operating conditions and 120 records (40 circumferential location for each of three blade-to-blade channel) were taken on N=300 consecutive revolutions.

Probes were located within the diffusers both radially, at several different distances from the machine axis, and axially at a distance of 3% of the outlet radius from the impeller exit.

When the probe was radially placed, a fixed orientation was adopted and the single wire was set parallel to the machine axis in order to be normal to absolute velocity (in fact negligible axial component was supposed, because of the aspect ratio of the channel). This radial set was used to measure the intensity of the velocity vector and the flow decay downstream of the impeller in the diffusers of all the stages.

On the contrary with probes axially placed, at the outlet of the first and fourth impeller, bi-dimensional flow measurements were done, adopting two different techniques to obtain the velocity vector and the Reynolds stress tensor on the blade-to-blade plane; a triple orientation technique with single-wire probe (Olivari & Salaspini 1975), and a fixed orientation technique with X-wire probe.

To have a significant description of the flow field, in the first stage the X-wire probe were used, in order to obtain a more accurate measurement of Reynolds stress.

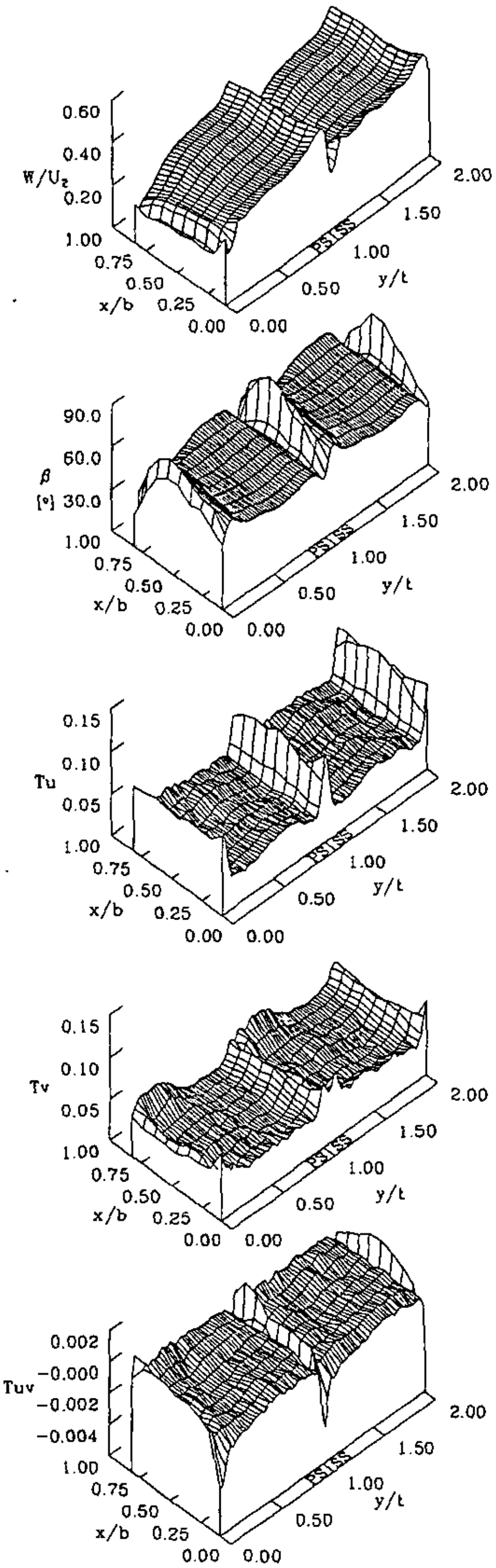

Fig. 4 - Flow pattern in the first stage at nominal conditions (φ=0.0065).

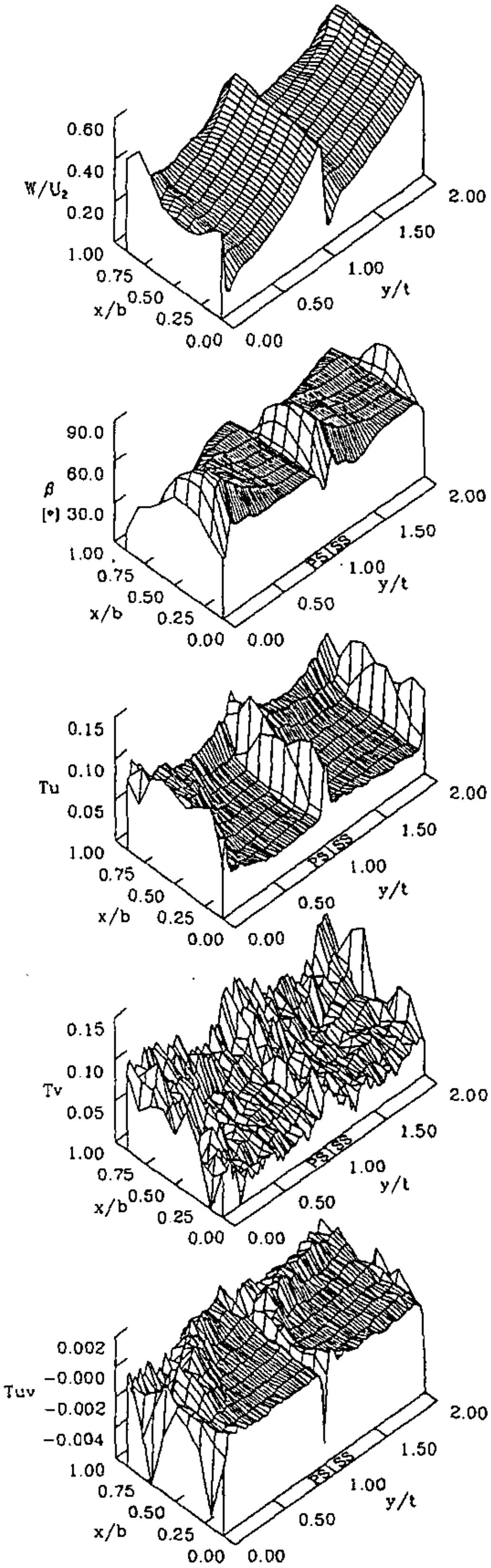

Fig. 5 - Flow pattern in the fourth stage at nominal conditions (φ=0.0065).

Since in the fourth stage the flow field was found not to be bi-dimensional, but to show zones with considerable axial gradients (above all near the hub), the triple orientation single wire probe technique was considered more correct in order to obtain the complete velocity vector field.

In fact, the two wires of the X-wire probes should not be too close in order to avoid reciprocal disturbance (Nagano and Tsuji 1993), but on the other hand they have to perform essentially a point measurement; that is possible only in bi-dimensional flows.

During X-wire probe measurements, the probe orientation was accurately set to minimise the aerodynamic disturbances caused by the tips of wire supports.

EXPERIMENTAL RESULTS

Stable flow conditions

Following is a description of the results of measurements carried out with blower in stable operating conditions (A, B, C, D), at the design rotational speed of 3000 rpm, by ensemble averaged velocity measurements.

Figures 4 and 5 show the conditions of the flow at the outlet of the first and fourth stage respectively, at the radial position R/R_2=1.03, at nominal flow rate *B (φ=0.0065)*.

Relative velocity W/U_2, relative flow angle β and turbulent Reynolds stress tensor components Tu, Tv, Tuv are represented as a function of the axial position x/b (0=shroud, 1=hub) and of the circumference coordinate y/t for two consecutive blade-to-blade channels (first channel 0=SS, 1=PS).

From e detailed analysis of the outlet flow at the first stage (figure 4) very uniform hub-to-shroud trends can be noted both in terms of velocity and turbulence in accordance with Gyarmathy et al. (1991), whereas no gradual variation between hub and shroud as reported by Olivari and Salaspini (1975) and Ishida et al. (1989), nor a wake area near the shroud as reported by Rohne and Banzhaf (1990) could be noted.

Instead relative velocity increases rather regularly in circumference direction from SS to PS, with a clear discontinuity in correspondence with blade passages; similar trends were reported by Inoue and Cumpsty (1984) and Flack et al. (1987).

The relative flow angle sharply decreases after the SS remaining almost constant in the blade channel and decreasing again at the PS.

The turbulence components remain approximately constant and show rather low values for most of the blade channel, with high peaks or remarkable discontinuities in correspondence to blade passages. Therefore, a clear area of higher turbulence could not be noted in proximity of the PS as reported by Maksoud and Johnson (1989).

Flow conditions are markedly different at the outlet of the fourth stage (figure 5). In hub-to-shroud direction there are marked differences in the area near the hub showing a higher relative velocity, a lower relative angle, and generally quite higher turbulence components, in accordance with data reported by Maksoud and Johnson (1989). However, relative velocity gradually increases in the blade-to-blade plane from SS to PS as in the first stage, although a lower relative velocity area may be pointed out near SS - shroud as reported by Ubaldi et al. (1992) and Eckardt (1975).

The relative angle in the fourth stage presents lesser variations in the entire channel, while the turbulence components highlight an area with higher turbulence near the SS - hub end. As far as turbulence is concerned it should be kept in mind that, according to the experience of authors, the different measurement technique (the single-wire probe instead of the two-wire one), can lead to greater errors and represents an obstacle to a really homogenous comparison with the first stage results.

A comparison among the different flow rates (A, B, C, D) is summarised in figure 6 showing the conditions of the outflow from the fourth stage at a mean axial position (x/b=0.5) and at the radial position R/R_2=1.03 again.

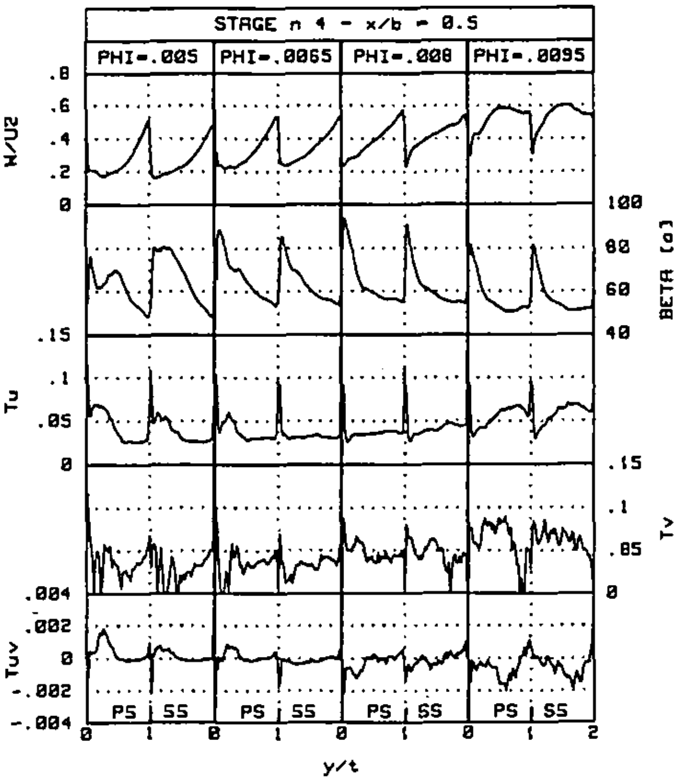

Fig. 6 - Blade-to-blade profiles of flow parameters: comparison among different flow rates (stage No 4 - x/b=0.5).

The trends of the relative velocity show a regular increase from SS to PS for low and medium flow rates, while a peak area is to be pointed out at the channel centre at φ=0.0095. A similar situation was reported for a centrifugal compressor by Inoue and Cumpsty (1984).

As regards the relative angle high values can be noted at φ=0.0050 right after the SS, while with high flow rates β rapidly decreases after the SS and remains around low values in the central part of the blade channel.

The flow turbulence is characterised by higher values of the Tu component near the PS for high flow rates, as measured by Olivari and Salaspini (1975), while at low flow rates an area of higher turbulence can be noted near the SS. In these conditions the occurrence of a wake area near the SS can by hypothesised, which is characterised by low velocity and high turbulence (Ubaldi et al. 1992, Eckardt 1975).

The different flow conditions at the outlet of the four impellers are compared in figure 7 with reference to the nominal flow rate (B).

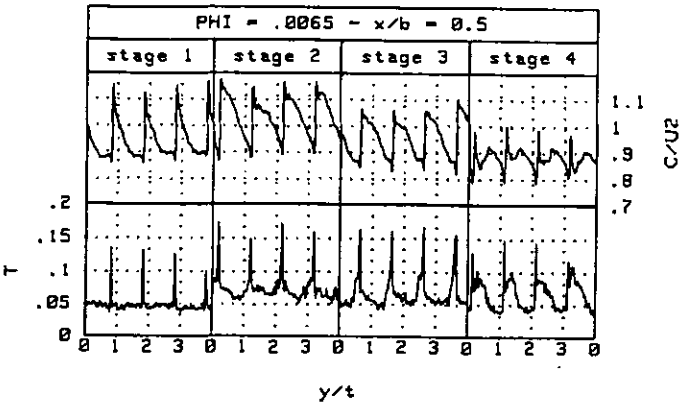

Fig. 7 - Blade to blade profiles of flow parameters at x/b=0.5: comparison among the four stages at nominal conditions (φ=0.0065).

These results refer to measurements carried out with the single-wire probe radially located and report the absolute velocity C/U_2 and its fluctuation intensity T, in the mean x/b=0.5 section at a 5 mm radial distance from the trailing edge (R/R_2=1.022). Remarkably different trends can be reported:

— in the first stage velocity rapidly decreases at the SS and the values of velocity fluctuation are low in the entire blade-to-blade channel;

— in the second and third stages velocity decreases gradually from SS to PS, with more discontinuous and in average higher fluctuation values;

— in the fourth stage the absolute velocity decreases sharply after the SS showing its relative maximum at the centre of the channel, with high fluctuation values near the SS, as already shown in figure 6 with the turbulence components.

In order to have a picture of the flow trend in the diffusers, figure 8 represents the trends of C/U_2 and T measured in correspondence with different radial positions in the first stage diffuser.

These measurements referring to nominal conditions were carried out with the single-wire probe radially located to the mean hub-to-shroud position (x/b=0.5), at a distance from the impeller exit varying from a minimum of 5 mm (R/R_2=1.022) to a maximum of 45 mm (R/R_2=1.194). The diagram points out the following progressive changes with the increasing radius: decrease of the velocity mean value, decrease in the velocity variation due to the blade passage, circumferential shift of these velocity variations, increase of the fluctuation mean value, decrease of the fluctuation peaks in correspondence with the blade passage.

Similar trends referring to the fluctuation with varying radius are reported by Maksoud and Johnson (1989). In conclusion it can be noted that at a max. 45 mm distance (R/R_2=1.129) from the impeller exit, the trend of velocity is only slightly influenced by the blade passage (velocity differences less than 6%), with an almost constant fluctuation value around 10%.

The flow field can be alternately shown by a representation of the radial iso-velocity lines in the section considered; figure 9 takes into consideration the conditions at the first and fourth impeller exit.

An analysis of the situation at the first stage indicates still a regular outflow both with varying y/t and x/b coordinates. Only a restricted low radial velocity area can be pointed out near the hub - SS, which can be clearly noted at high flow rates; thus there is not a clear wake area in proximity of SS at low flow rates as normally happens with compressors (Inoue and Cumpsty 1984, Gyarmathy et al. 1991, Johnson and Moore 1983), but greater velocity gradients at high flow are to be noted, as reported by Maksoud and Johnson (1989), and in any case in the area near the hub (Jaberg and Hergt 1989).

On the contrary, the situation is substantially different in the fourth stage where there are considerable velocity gradients passing from the hub to the shroud; an extended low velocity area clearly appears near the hub at high flow rates (Jaberg and Hergt 1989), which is present however also at low flow rates near SS. Higher turbulence values correspond to these low radial velocity areas (figures 5 and 6).

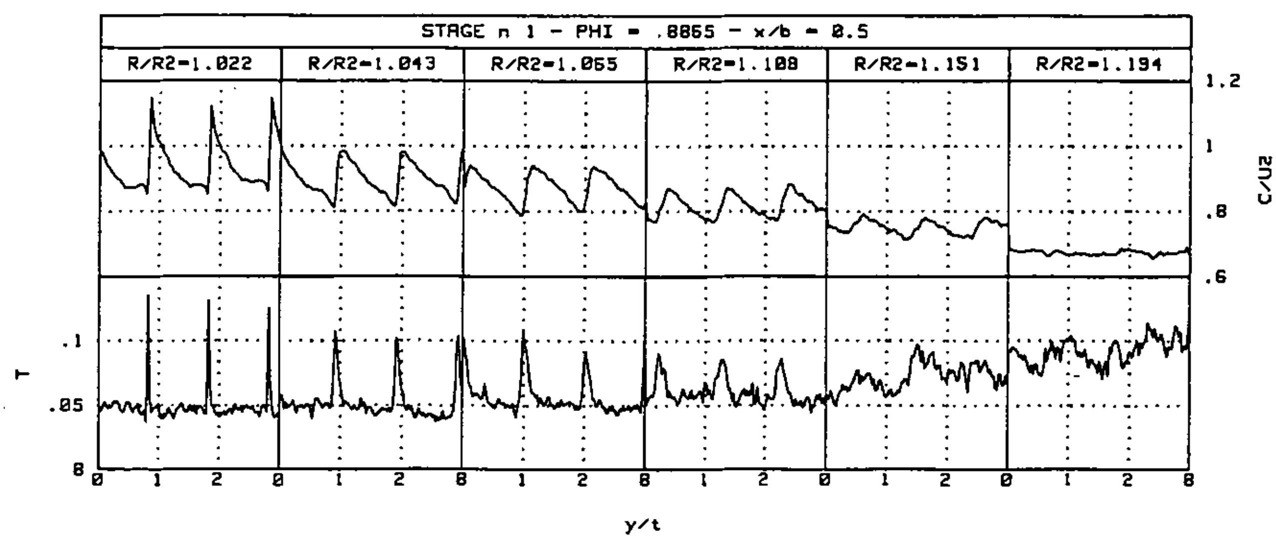

Fig. 8 - Flow parameters traces in the first stage diffuser at x/b=0.5: comparison among differing radial position, at nominal conditions (φ=0.0065).

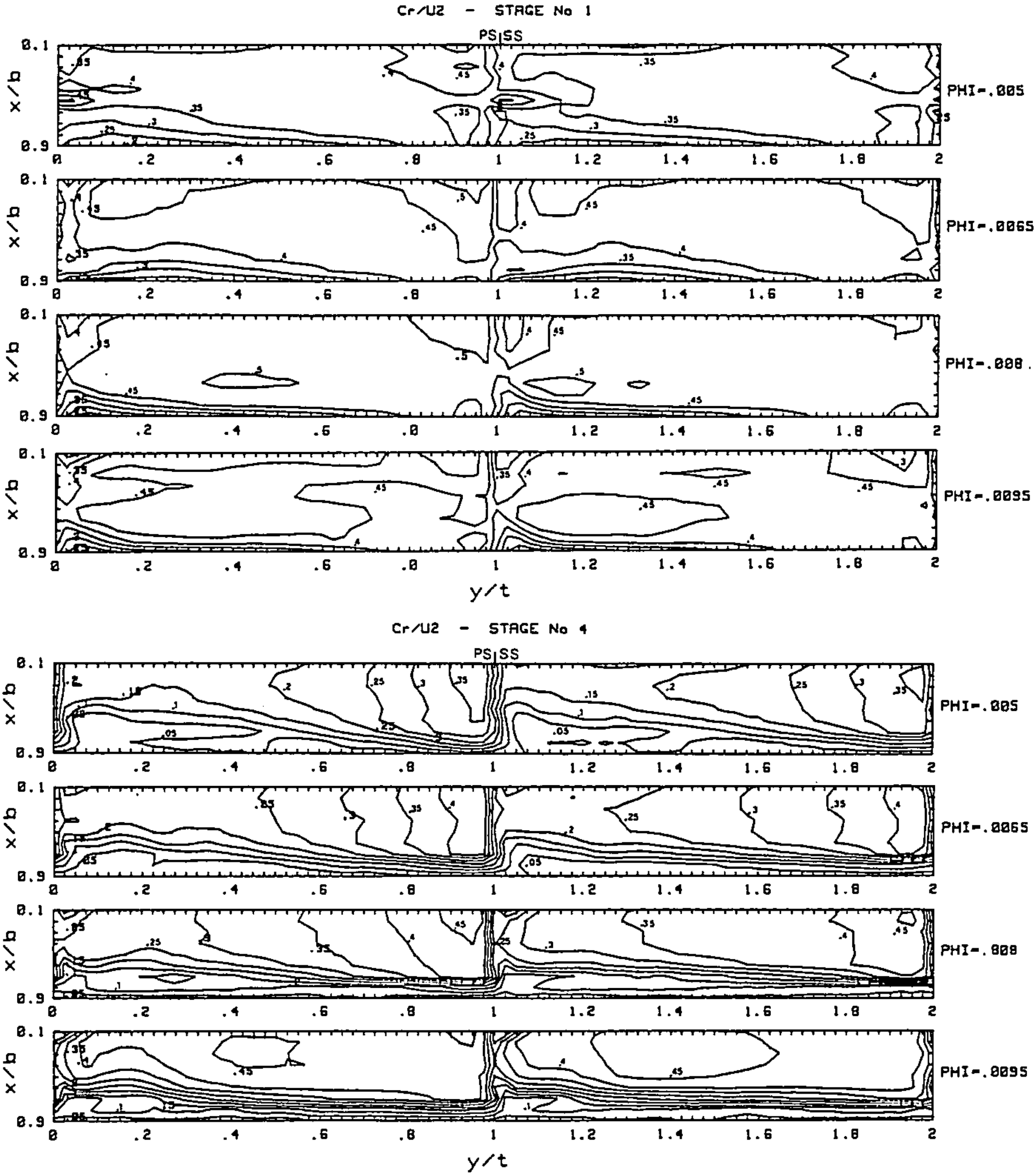

Fig. 9 - Isolines of the radial velocity at the first and fourth impellers exit: comparison among different flow rates.

Unstable flow conditions

The subsequent phase has been a study of the trends of instantaneous velocity at the impellers' outlet in unstable conditions in order to observe the modifications of the flow with a diminishing flow rate. All the tests were caeeied out at the nominal rotational speed (3000 rpm). The X hot-wire probe was placed axially at the outlet of the first and of the fourth impeller at an intermediate position (x/b=0.5), obtaining directly the value of the instantaneous velocity vector in the blade-to-blade plane along all blade channels (40 measure points for each channel) for N=32 consecutive revolutions.

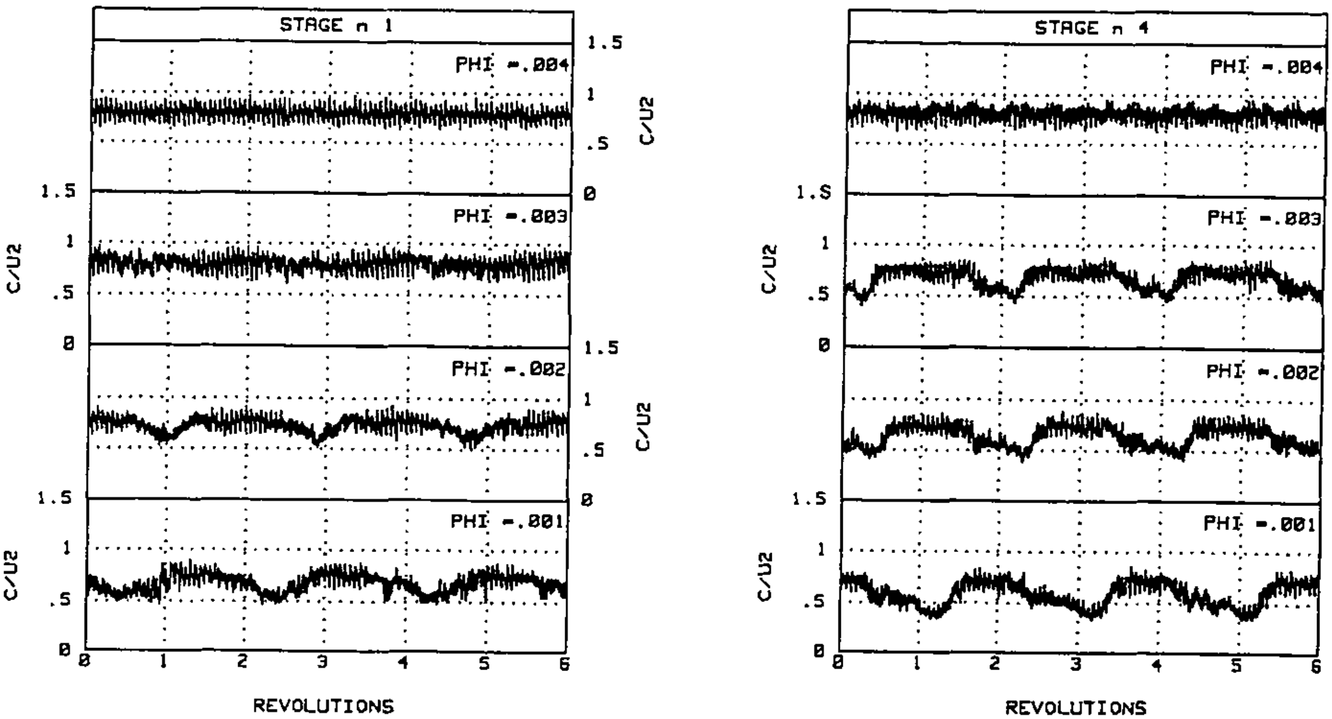

Fig. 10 - Velocity traces at the impellers exit in unsteady flow conditions (x/b=0.5).

Spectral analysis of the velocity signals was performed to gain quantitative information on the frequency characteristics of the observed flow instabilities, using the FFT technique.

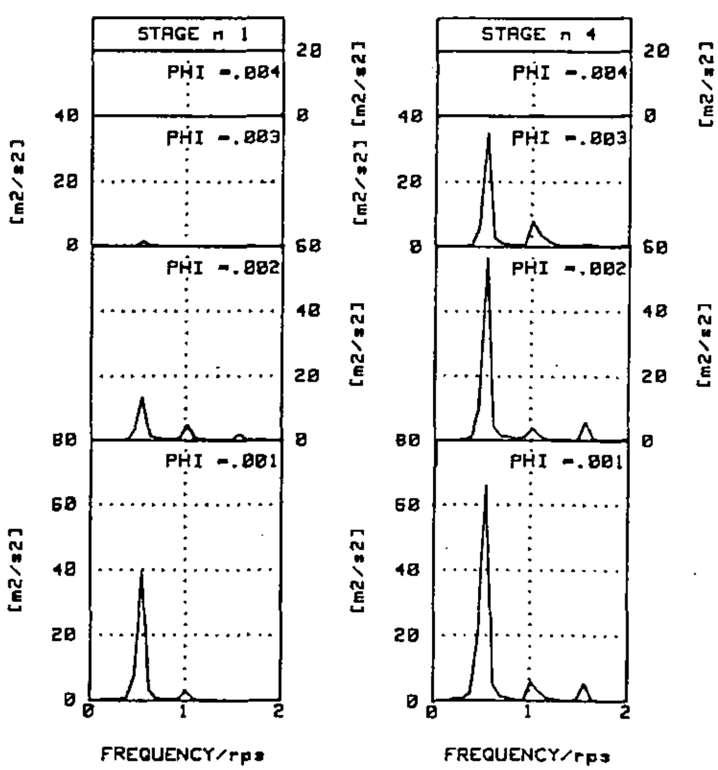

Fig. 11 - Power spectra of the velocity at the impellers exit (x/b=0.5).

Figure 10 shows the trend of instantaneous velocity at the outlet of both the stages for the four flow coefficient values considered (E, F, G, H) starting φ=0.0040, that had been already selected as transition value in unstable conditions (figure 2).

An analysis of figure 10 points out the occurrence of a low frequency oscillation starting to φ=0.0030 in the first stage, but just perceptible from φ=0.0040 in the fourth stage, in which the wave amplitude is greater than in the first one for all the considered flow rates. In both cases this oscillation persists and becomes higher with decreasing flow rate, showing a periodic pattern - corresponding to two impeller rotations approximately - characterising the presence of the rotating stall (Kinoshita and Senoo 1985, Ubaldi and Zunino 1990, Kämmer and Rautenberg 1986).

Figure 11 shows the corresponding results obtained by means of FFT analysis and reports the power spectra as a function of the frequency made dimensionless in respect to the base rotation frequency of the impeller. Data reported for φ=0.0040 present a very low uniform background noise. The complete spectra, not reported in the figure, show only the harmonics of the 16 frequency ratio corresponding to the number of blades. From φ=0.0030 a peak can be observed which is to be attributed to the stall at a frequency equal to one half of the base frequency approximately. With decreasing flow rate this frequency does not change, while the amplitude of the local disturbance is slightly increasing. In the fourth stage amplitudes are always greater than in the first stage, and a harmonic of a higher order can be observed.

In order to analyse the flow characteristics in greater detail in these unstable conditions, figure 12 shows the absolute velocity intensity trends and its angle for two revolutions only referring to the lower flow rate (φ=0.0010). Substantial differences can be noted according to the position on the instability wave:

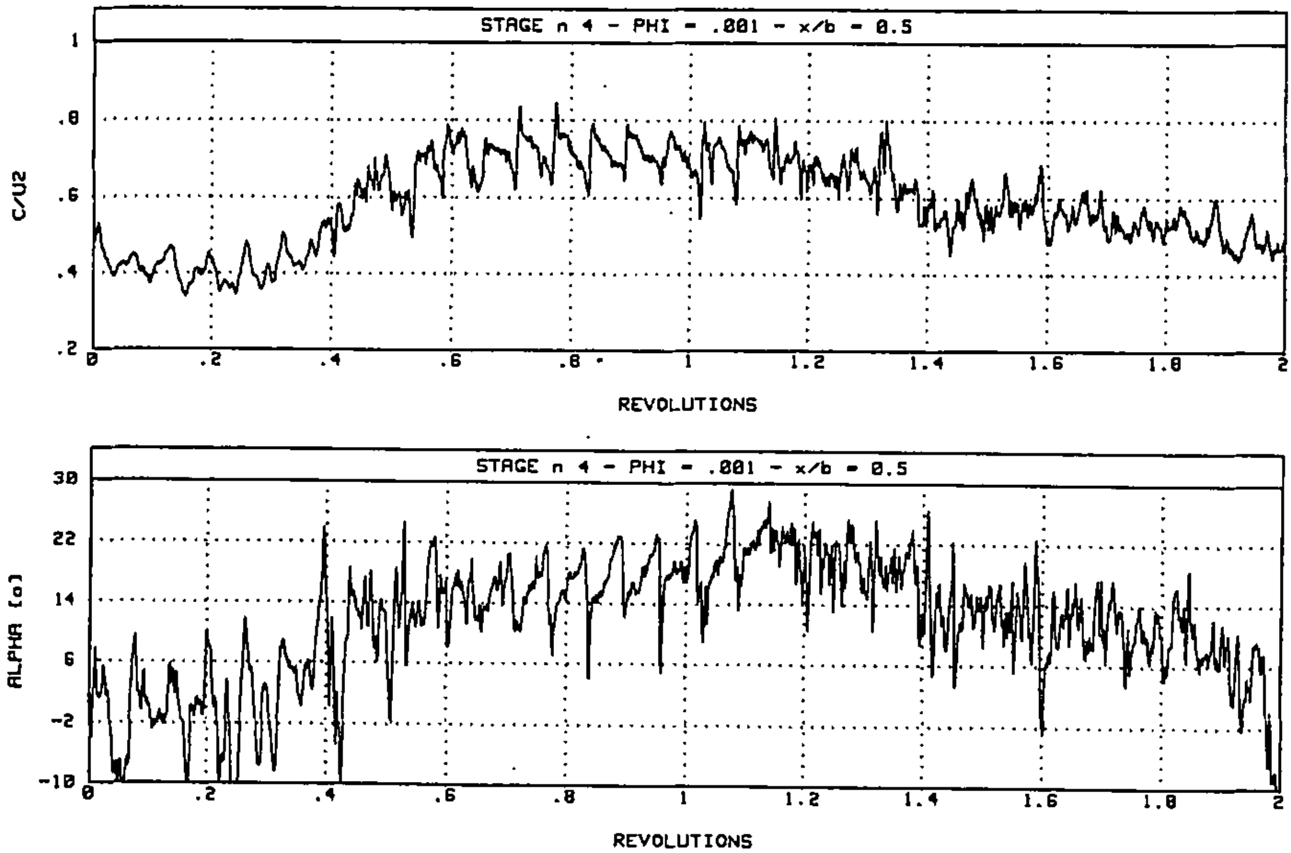

Fig. 12 - Absolute velocity and flow angle traces at the fourth impeller exit at x/b=0.5 at unsteady flow condition (φ=0.0010).

— in the area of the minimum mean velocity (abscissa = 0-0.3) instantaneous velocity tends to increase passing from SS to PS with a corresponding marked decrease of the angle which reaches negative values (reverse flow);

— in the area of increasing mean velocity (abscissa = 0.3 - 0.65) irregular trends can be noted both for the module and for the angle, with marked oscillations within the blade channels;

— in the area of mean maximum velocity (abscissa = 0.65 - 1.15) trends similar to those observed in stable conditions can be noted, with a decrease of C from SS to PS and a corresponding increase of α;

— in the area of decreasing mean velocity (abscissa = 1.15 - 2) irregular trends with strong oscillations occur again.

CONCLUSIONS

The instantaneous flow at the outlet of the impellers of a multistage centrifugal blower has been measured and analysed in detail at different operating conditions, using a hot-wire anemometer with stationary probes.

With varying flow rates remarkable differences in the trends of velocity and of turbulence in the blade-to-blade plane could be observed.

Significant differences have been observed for the different stages, particularly the first, intermediate, and fourth one; the latter presenting a flow separation zone in the channel near SS.

Moving away from the impeller, the blade passage influence rapidly decreases reaching almost uniform velocity at the diffuser outlet, but with strong turbulence.

The flow at the outlet of the first impeller is sensibly bi-dimensional and presents gradual variations from SS to PS with perceptible gradients only in correspondence with the blade passages.

The flow at the outlet of the fourth impeller is less uniform and presents high values from the hub to the centre of the channel in progressively extended areas with the decreasing of the flow rate.

The flow does not present the configuration of the potential theory in the blade-to-blade plane. It does not equally show the jet and wake flow typical of centrifugal compressors. It has to be noted that the test blower falls right in the transition zone indicated by Adler (1980).

Finally, data were collected also with the flow in unstable conditions measuring remarkable velocity oscillations due to the presence of the rotating stall.

In these conditions the detailed analysis of the velocity trends in the blade-to-blade plane pointed out opposite trends in the different oscillation zones, with a flow inversion in the area of minimum velocity.

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