MEASUREMENT OF VELOCITY DISTRIBUTION AT
THE IMPELLER EXIT OF A RADIAL COMPRESSOR

D. OLIVARI and A. SALASPINI

MARCH 1975
MEASUREMENT OF VELOCITY DISTRIBUTION AT THE IMPELLER EXIT OF A RADIAL COMPRESSOR

D. OLIVARI and A. SALASPINI

MARCH 1975
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>SUMMARY</td>
<td>iii</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>v</td>
</tr>
<tr>
<td>LIST OF SYMBOLS</td>
<td>vi</td>
</tr>
<tr>
<td>1. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>2. TEST SET-UP</td>
<td>4</td>
</tr>
<tr>
<td>2.1 The radial compressor</td>
<td>4</td>
</tr>
<tr>
<td>2.2 The measurement chain</td>
<td>5</td>
</tr>
<tr>
<td>2.2.1 Measurement of the velocity components with the hot wire probe</td>
<td>5</td>
</tr>
<tr>
<td>2.2.2 Measurement of the velocity fluctuation</td>
<td>8</td>
</tr>
<tr>
<td>2.2.3 The synchronous sampling technique</td>
<td>9</td>
</tr>
<tr>
<td>3. RESULTS</td>
<td>13</td>
</tr>
<tr>
<td>3.1 Introduction</td>
<td>13</td>
</tr>
<tr>
<td>3.2 Radial velocity</td>
<td>13</td>
</tr>
<tr>
<td>3.3 Slip factor</td>
<td>16</td>
</tr>
<tr>
<td>3.4 Blade-to-blade turbulent distribution</td>
<td>17</td>
</tr>
<tr>
<td>4. CONCLUSIONS</td>
<td>19</td>
</tr>
<tr>
<td>4.1 Detailed test results</td>
<td>19</td>
</tr>
<tr>
<td>4.2 Mass averaged flow characteristics</td>
<td>19</td>
</tr>
<tr>
<td>REFERENCES</td>
<td>20</td>
</tr>
<tr>
<td>FIGURES</td>
<td>21</td>
</tr>
</tbody>
</table>
SUMMARY

The flow field at the exit of the impeller of a low speed radial compressor was investigated in detail to analyse the velocity distribution over one blade pitch.

The main goal of the research was the mapping of the "jet and wake" configurations at different flow rates, and at different rotational speeds. For this purpose, spanwise traverses were carried out at two circumferential angles. Furthermore, an attempt was made to measure the two components of the turbulent fluctuation parallel and normal to the absolute velocity in the jet and in the wake.

All the measurements were performed by means of a hot wire anemometer. A particular calibration technique was used to resolve the different velocity components. The data analysis was performed by means of an analog conditional sampler developed by one of the authors which made possible the actual measurement of the blade-to-blade velocity distribution.

The results confirm the theoretical prediction that if a jet and wake configuration occurs, a low value of the slip factor is obtained on the blade pressure side where the jet is developing.

The velocity map led to an estimation of the secondary flows which do not seem to be predominant on the blade height.

A new data reduction technique was used to measure the turbulence parameters in the blade-to-blade passage. The intensity of the velocity fluctuations parallel to the mean absolute velocity is maximum corresponding to the blade trailing edge. The velocity fluctuations perpendicular to the mean velocity have a maximum on the pressure side and a minimum on the suction side.
Some data were also obtained for the variation of the turbulence correlation factor (uv), which represents the turbulent shear stresses over a blade pitch.
LIST OF FIGURES

1  Schematic of the separated flow model
2  Back flow model at centrifugal impeller exit
3  Testrig plan
4  Overall performance
5  Lighting system
6  Hot wire calibration
7, 8  Symbolism
9  Blade-to-blade flow field example
10 a, b, c  Hot wire signal of a single passage flow field
11 a, b, c  Envelope of hot wire signals
12 a, b, c  Sampling procedure
13  Sampler output
14 a-d  Radial velocity distribution for tests 1, 2, 3, 4
15 a, b, c  Radial velocity distribution for tests 5, 6, 7
16  Mean averaged radial velocity for RPM = 3000
17 a-d  Radial velocity distribution for tests 8, 9, 10, 11
18 a-d  Slip factor distribution for tests 1, 2, 3, 4
19 a, b, c  Slip factor distribution for tests 5, 6, 7
20 a-d  Slip factor distribution for tests 8, 9, 10, 11
21  Mass averaged slip factors
22 a, b, c  Velocity fluctuation component and shear stress
23  Mean absolute velocity
24  Radial velocity component
**LIST OF SYMBOLS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area</td>
</tr>
<tr>
<td>b</td>
<td>blade length</td>
</tr>
<tr>
<td>D</td>
<td>diameter</td>
</tr>
<tr>
<td>E</td>
<td>mean hot wire output</td>
</tr>
<tr>
<td>e</td>
<td>fluctuation of the hot wire output</td>
</tr>
<tr>
<td>l</td>
<td>traverse length</td>
</tr>
<tr>
<td>Q</td>
<td>flow rate</td>
</tr>
<tr>
<td>R</td>
<td>radius</td>
</tr>
<tr>
<td>T</td>
<td>period</td>
</tr>
<tr>
<td>t</td>
<td>time</td>
</tr>
<tr>
<td>u</td>
<td>fluctuation component parallel to the mean velocity</td>
</tr>
<tr>
<td>U</td>
<td>peripheral velocity</td>
</tr>
<tr>
<td>v</td>
<td>fluctuation component perpendicular to the mean velocity</td>
</tr>
<tr>
<td>V</td>
<td>absolute mean velocity</td>
</tr>
<tr>
<td>V'</td>
<td>actual absolute velocity</td>
</tr>
<tr>
<td>x</td>
<td>transversal position</td>
</tr>
<tr>
<td>z</td>
<td>blade number</td>
</tr>
<tr>
<td>α</td>
<td>angle between the flow and the normal to the hot wire sensor</td>
</tr>
<tr>
<td>α_i(i=1,2,3)</td>
<td>probe positions</td>
</tr>
<tr>
<td>β</td>
<td>relative angle (referred to tangential velocity)</td>
</tr>
<tr>
<td>δ</td>
<td>angular velocity fluctuation</td>
</tr>
<tr>
<td>γ</td>
<td>circumferential angle</td>
</tr>
<tr>
<td>θ</td>
<td>phase angle</td>
</tr>
<tr>
<td>μ</td>
<td>slip factor</td>
</tr>
<tr>
<td>φ</td>
<td>flow coefficient</td>
</tr>
<tr>
<td>ψ</td>
<td>pressure coefficient</td>
</tr>
<tr>
<td>τ</td>
<td>delay time</td>
</tr>
</tbody>
</table>

**Subscripts**

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>true</td>
</tr>
<tr>
<td>1</td>
<td>inlet</td>
</tr>
<tr>
<td>2</td>
<td>outlet</td>
</tr>
<tr>
<td>b</td>
<td>blade</td>
</tr>
<tr>
<td>j</td>
<td>jet</td>
</tr>
<tr>
<td>p</td>
<td>passage</td>
</tr>
<tr>
<td>R</td>
<td>radial</td>
</tr>
<tr>
<td>w</td>
<td>wake</td>
</tr>
<tr>
<td>θ</td>
<td>tangential</td>
</tr>
</tbody>
</table>
1. INTRODUCTION

The flow field inside the impeller of a radial machine is a very complex one: Coriolis and centrifugal forces, unsteadiness, secondary flows, asymmetric flows, boundary layer separation are present simultaneously to make it a fully three dimensional problem.

Quite a large amount of research has already been carried out on this subject, nevertheless, there is still an increasing need for experimentation and for more developed theories to continuously improve the characteristics of the machines.

The most important phenomenon to be analyzed in a centrifugal impeller is the so called "jet and wake" flow field, which consists in a large region of separated flow on the suction side of the blade and in the concentration of the bulk of the flow rate close to the pressure side. Such a complexity of the flow field explains the failure of all the simplest quasi three dimensional potential flow calculations. An approach to the theoretical evaluation of such a flow was developed first by Dean and Senoo (Ref. 1), then by Johnston and Dean (Ref. 2).

The basic assumptions were: constant relative angle, no delivery in the wake region. Such a treatment was strongly criticized by Baade (Ref. 3), who proposed a theory based on the time dependent equations. Unfortunately, he could not produce quantitative results due to the actual lack of informations about the characteristics of the unsteady flow in radial machines.

Moreover, as pointed out by several authors, the jet and wake configuration is determined by the Coriolis force which decreases the entrainment on the suction side, hence, it can be expected that when the Coriolis force is small (low rotation speed or low flow rates), the jet and wake development is less evident. The Coriolis force can also be balanced by the backward curvature of the blades which increases the entrainment on the suction side. This fact clearly appears from the results of Howard and Kittmer (Ref. 4) who measured a non-separated quasi potential flow in a
low speed centrifugal compressor with backward bent blades.

On the contrary, high speed radial compressors always have jet and wake configuration with a wake region never delivering flow rates higher than 20% of the total flow rate.

Secondary flows are another important phenomenon occurring in radial impellers. Smith (Ref. 5) developed a theoretical approach which describes with good approximation the flow inside shrouded wheels, but which fails if the wheel is unshrouded. As a matter of fact, in this second case, leakage flows are superimposed on the main flow and they cannot be separated from secondary flows generated by an inlet vorticity.

Dean (Ref. 6) gave a schematic representation of the leakage flow behaviour: its appears (Fig. 1) that material is added to the wake through the jet boundaries. Furthermore, Eckardt (Ref. 7) shows (Fig. 2) that a large reverse flow bubble settles close to the blade tip and a limited one close to the blade hub. Therefore a quicker deterioration of the flow pattern can be expected closer to the tip than to the hub of the blade.

It is also known that at off design conditions, in a centrifugal compressor the static pressure is not constant at the wheel exit. Nevertheless, the effect of this non uniformity on the jet and wake configuration is not well documented in the literature.

Finally, no information is available on the values of the velocity fluctuations at the impeller periphery, and thus on their possible influence on the jet and wake configuration, on secondary flows and leakages.

Improvements in instrumentation and in measurement techniques are now making possible more detailed surveys such as the measurements of the blade-to-blade velocity distribution at the exit of compressor wheels and of the velocity fluctuations. It is the purpose of this paper to describe such a technique and to analyze the results obtained in view of the available theories.
An electronic chain including a hot wire anemometer and a conditional sampler was used to perform the measurements at the outlet of the impeller of a low speed radial compressor at two circumferential angles and two rotational speeds for various load conditions.
2. TEST SET-UP

2.1 The radial compressor

A low speed unshrouded centrifugal compressor with the following characteristics is used:

<table>
<thead>
<tr>
<th>Blades</th>
<th>$D_1/D_2$</th>
<th>$b_2/D_2$</th>
<th>$\beta_{1b}$</th>
<th>$\beta_{2b}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>.584</td>
<td>.103</td>
<td>90°</td>
<td>90°</td>
</tr>
</tbody>
</table>

The impeller is constituted by radial blades without inducer and it is coupled with a concentric volute.

The load circuit (Fig. 3) is equipped with a diaphragm and a control valve to measure and to vary the flow rate. A pitot probe placed inside the pipeline measures the total head supplied by the machine. This enabled the measurement of the compressor performance, which is given by figure 4 as pressure coefficient vs flow coefficient. A disk with 8 circumferential holes, which corresponds exactly to the position of the blade trailing edge, was fixed on the shaft (Fig. 5): a lamp and a phototransistor system are used to produce a pulse synchronized with each blade. In such a way a master signal is available to synchronize the conditional sampler; at the same time it is used with a counter to provide the rotational speed of the machine.

For the velocity measurements axial traverses were made at the radial position $R/R_2 = 1.015$ and at the circumferential angles $\gamma = 45^\circ, 135^\circ$. For each traverse, measurements were made at six axial positions $\frac{x}{L} = .143, .286, .429, .572, .715, .858$ (corresponding to the blade span).

From the analysis of the overall performance curve it was decided to make the measurements at the following conditions:
# TABLE 1

<table>
<thead>
<tr>
<th>Test No</th>
<th>φ</th>
<th>ψ</th>
<th>γ</th>
<th>RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>.343</td>
<td>.462</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>2</td>
<td>.267</td>
<td>.702</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>3</td>
<td>.194</td>
<td>.880</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>4</td>
<td>.139</td>
<td>.940</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>5</td>
<td>.267</td>
<td>.702</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>6</td>
<td>.194</td>
<td>.880</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>7</td>
<td>.139</td>
<td>.940</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>8</td>
<td>.341</td>
<td>.475</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>9</td>
<td>.294</td>
<td>.615</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>10</td>
<td>.218</td>
<td>.835</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>11</td>
<td>.124</td>
<td>.946</td>
<td>135</td>
<td>2400</td>
</tr>
</tbody>
</table>

## 2.2 The measurement chain

All the measurements were carried out using a hot wire probe connected to a constant temperature hot wire anemometer, built at VKI (Ref. 8). The directional sensitivity of the hot wire was used to resolve the velocity into its different components (mean velocity and fluctuation) by multiple measurements at each point. The data were analyzed, to obtain the time dependent profile, by an online synchronous sampling technique developed at VKI for measurements in periodic flows (Ref. 9).

### 2.2.1 Measurements of the velocity components with the hot wire probe

The output of the hot wire anemometer was linearized to obtain an output voltage directly proportional to the effective cooling velocity on the probe. This was a fork type probe on which was soldered a 5 μm tungsten wire copper coated at the extremities. The length of the sensitive portion of the wire was approximately 2 mm. The frequency response of the system was measured to be 40 kHz, a value which was considered adequate for this experiment.
The probe was mounted with the stem axis parallel to the axis of rotation of the wheel and thus perpendicular to the flow direction. It was supported by a traversing mechanism fixed to the outer case of the compressor and allowing longitudinal displacement and a rotation around the axis.

The determination of intensity and direction of the velocity vector and its resolution in a radial and tangential component was carried out by making use of the directional sensitivity of the hot wire.

A hot wire exposed, with a certain angle $\alpha$ to the flow, gives an output which is proportional to the absolute velocity times a function of the angle. This output expressed in terms of velocity is usually defined as the effective cooling velocity and can be seen as a combination of the velocity components perpendicular and parallel to the wire.

Many relations have been proposed in the literature for the evaluation of this angular sensitivity. However, it is the opinion of the authors that the best method consists in a direct calibration of the probe to be used in a uniform flow field.

The actual calibration curve obtained in this way is shown in figure 6. As may be seen, the curve is perfectly symmetrical and the best fit relation is a quadratic law of the form

$$V(\alpha) = V_0 (1 - 0.000125\alpha^2)$$

(1)

where $\alpha$ is expressed in degrees. This expression is valid for $|\alpha| \leq 50^\circ$.

By making two measurements at two different probe angles, $\alpha_1$, $\alpha_2$, fixed with reference to the zero direction, it is possible to determine the absolute velocity and its direction, $V_0$ and $\alpha$ (see Fig. 7), by using the relations:

$$V(\alpha - \alpha_1) = V_0 (1 - 0.000125(\alpha - \alpha_1)^2)$$

$$V(\alpha - \alpha_2) = V_0 (1 - 0.000125(\alpha - \alpha_2)^2)$$

(2)
where $\alpha$ is the angle of the velocity with reference to a chosen probe position.

The solution is obtained by first solving the equation obtained making the ratio of (2)

$$\frac{V(\alpha-\alpha_1)}{V(\alpha-\alpha_2)} = \frac{f(\alpha-\alpha_1)}{f(\alpha-\alpha_2)} = F_1(\alpha_1, \alpha_2, \alpha)$$

which gives $\alpha$. Then, by substituting in (2), it is possible to obtain $V_0$.

To insure a better accuracy and to evaluate the error in the angle measurement, a cross check was made by placing the wire at a third angle $\alpha_3$ to obtain

$$V(\alpha-\alpha_3) = V_0 \left(1 - 0.000125(\alpha-\alpha_3)^2\right)$$

and

$$\frac{V(\alpha-\alpha_2)}{V(\alpha-\alpha_3)} = \frac{f(\alpha-\alpha_2)}{f(\alpha-\alpha_3)} = F_2(\alpha_2, \alpha_3, \alpha)$$

If the differences $\alpha_2-\alpha_1$ and $\alpha_3-\alpha_2$ are equal, the two functions of $\alpha$ in (3) and (5) are identical ($F = F_1 = F_2$). This is the solution adopted in practice with

$$\alpha_2 - \alpha_1 = 40^\circ$$
$$\alpha_3 - \alpha_2 = 40^\circ$$

to ensure a high sensitivity in the evaluation of the function $F$. Furthermore, the alignment $\alpha_2$ was chosen such as the sensor is almost perpendicular to the estimated direction of the velocity vector at the wheel exit. The calculations were carried out on a computer using a best fit curve for the function $F$. When one of the computed values $\alpha - \alpha_i$ ($i = 1,2,3$) was larger than $50^\circ$ (limit of validity of the calibration curve) it was rejected and the other computed value taken as the correct solution. Otherwise, the true angle was calculated as the mean value of the two results.
Under these conditions the typical error (difference between the two computed results) was of the order of 1° for the instantaneous velocity profile.

From \( V_0 \) and \( \alpha \) (referred to the tangential direction), the different velocity components and the slip factor were computed with the relations:

\[
\begin{align*}
V_{R_2} &= V_0 \sin \alpha \\
V_{\theta_2} &= V_0 \cos \alpha \\
\mu &= \frac{V_{\theta_2}}{U_2}
\end{align*}
\]

2.2.2 Measurement of the velocity fluctuation

With the same measurement technique it is also possible to determine the mean square values of the velocity fluctuation. With the symbols of Fig. 8, the following relation can be written for a wire oriented at an angle \( \alpha \) to the mean flow:

\[
V' = (V + u) f(\alpha + \delta)
\]

and for small angles \( \delta \), that is relatively low turbulence levels:

\[
\delta = \frac{V'}{V}
\]

so that

\[
V' = (V + u) \left[f(\alpha) + f'(\alpha) \times \delta\right] =
\]

\[
= (V + u) \left[f(\alpha) + f'(\alpha) \times \frac{V'}{V}\right] =
\]

\[
= Vf(\alpha) + Vf'(\alpha) + uf(\alpha)
\]

(7)

neglecting second order terms.

The linearized output signal \( V' \) is composed of a mean output \( E \) and of a fluctuating signal \( e \), therefore

\[
V' = E + e
\]
and the mean square value of the fluctuation components is obtained from (7) as

\[ e^2 = \bar{u}^2 f^2(\alpha) + \bar{v}^2 f'(\alpha)^2 + 2\bar{uv} f(\alpha)f'(\alpha) \]  

(8)

To solve for the three variables \( \bar{u}^2, \bar{v}^2, \bar{uv} \), it is necessary to have three equations corresponding to three different angles \( \alpha \). For the measurements the same angles \( \alpha_1, \alpha_2, \alpha_3 \) were used as for the measurements of the mean flow.

It is thus possible by solving the system of eq. 8 for three angles to obtain the values (and with the technique described in the following section, the instantaneous values) of the turbulent intensity components \( \bar{u}^2, \bar{v}^2 \) parallel and perpendicular to the mean velocity vector, and the value of the turbulent shear stress \( \bar{uv} \) in the plane parallel to the wheel.

2.2.3 The synchronous sampling technique

To obtain the instantaneous velocity field, or in other words, to reconstruct the blade-to-blade velocity distribution in a rotating machine, a synchronous sampling technique was used.

The apparatus retained for these experiments was an analogue one, working on line with the hot wire anemometer.

The blade-to-blade velocity can be described in an absolute frame as the change in velocity seen by a fixed probe over a time interval \( T \) equal to the time taken for a complete pitch to sweep across the fixed sensor

\[ T = \frac{60}{Z \times \text{RPM}} \]

such as shown in figure 9.

This velocity versus time profile is normally different for each passage for essentially two reasons: first the effect of turbulence which has a high energy content at frequencies which are similar to the frequency generated by the blade passing. Second, because of small manufacturing differences between each blade.
If the second effect could be practically neglected in a well constructed machine, the first one is very important and leads to a modulation of the signal such that practically no useful information could be gained by the analysis of a single passage.

To obtain useful information it would be necessary to analyse statistically a large number of such passages to determine as a result the mean velocity distribution and the fluctuation intensity. An example of a single blade-to-blade velocity distribution for the three wire orientations is given in Figs. 10 a, b, c while Fig. 11 a, b, c show the envelope obtained by recording on a memory oscilloscope a large number of such distributions again for the three probe positions.

The averaging process is, for the measurements, performed by synchronously sampling the anemometer output. With reference to Fig. 12-a where two hypothetical blade-to-blade velocity distributions are reproduced, widely different one from the other due to the turbulent fluctuations, a sample is made for a very short time and the result memorized for a duration equal to the cycle period. The operation is repeated at the next cycle and the new result stored, and so on for a large number of cycles.

The resulting output will be similar to that in Fig. 12-b that is a series of square-wave like pulses of period T. This sampling operation is equivalent to the multiplication of the original signal with a Dirac function with a phase lag

$$\theta = 360 \frac{T}{T}$$

with respect to the cycle of the main periodic phenomena and to a memorization of the result. To ensure the constancy of the sampling phase, a master pulse is obtained from a light interruption probe situated across a perforated disk fixed to the machine shaft, as shown in Fig. 5.

If the resulting signal is now averaged over a time long enough, i.e., over a large enough number of blade passages, the result will be the average value (considered as the statistical
average over a large number of different events) of the velocity 
\(<V>\) at that particular phase.

This value, as shown in Fig. 12-c depends on the position 
of the hot wire with respect to the rotating blade, that is, it is 
dependent on the sampling phase \(\theta\). By changing the sampling delay 
\(\tau\) it is possible to obtain a complete exploration of the cycle of 
the phenomenon that is the complete distribution of the velocity 
in a blade-to-blade passage, which will correspond to the continuous 
line in Fig. 12-a. An example of such procedure is shown in Figs. 
13 corresponding to the pictures of Figs. 11 a,b,c.

At the same time, the difference between the average 
velocity and the sampled signal is the local value of the turbulent 
velocity \(u\). Thus, by computing the variance of the sampled signal 
it is possible to measure the intensity of the local velocity fluc­
tuation \(<u^2>\) for each value of the sampling phase. The procedure 
could be repeated, of course, to obtain its value over a complete 
cycle. If

\[
\text{u} = \text{V} - <V>
\]

The value of the \(<u^2>\) represents the statistical average value 
of the intensity of the turbulent fluctuations over a large number 
of independent events, and should thus be considered as a correct 
dication of turbulence intensity for periodic flows.

All the variables introduced are examplified in Fig. 12-c 
where \(V_m\) represents the mean value of the velocity over a complete 
cycle and thus the value of the velocity which will be measured 
for instance with a pitot probe. By definition, \(V_m\) is equal to the 
average value of \(<V>\) over a complete cycle.

As shown in section 2.2.1 the measured value of \(V\) for 
a general orientation of the hot wire probe is in reality a linear 
combination of radial and tangential velocity as well as \(<u^2>\) is 
a linear combination of \(<u^2>\), \(<v^2>\), \(<uv>\).
However, by applying the data reduction technique mentioned in that section to the results of the sampled and averaged signal it is possible to obtain the variation with time of the intensity of the absolute velocity vector and its angle in the plane of the rotating wheel, as well as the intensity of its fluctuation component and of the shear stresses in the same plane.

In the actual test the sampler used has a sampling frequency range of 15 Hz to 20 kHz, with sampling window of 4 μsec. The latter is short enough to resolve the highest frequency fluctuations of the turbulence components. The averaged output values \( \langle E \rangle \), and \( \langle e^2 \rangle \) were recorded on a x-y plotter as a function of the sampling phase.
3. RESULTS

3.1 Introduction

The results of the measurements are presented in Figs. 14 to 24. The flow field described by the radial velocity component $v_R$ and by the slip factor $\mu$ is plotted over one pitch and over the blade span.

The pitch length is given by the phase angle $\theta$ going (by definition) from $0^\circ$ to $360^\circ$; the sampling phase is chosen in such a way to have the blade at $\theta = 180^\circ$. Measurements are made at six axial positions, the first one corresponding to the blade tip (near front cover) at $\frac{x}{l} = .143$ and the last one to blade hub (near rear cover) at $\frac{x}{l} = .858$, $l$ being the traverse length.

3.2 Radial velocity

Figures 14 a,b,c,d and 15 a,b,c, show the radial velocity distribution respectively at the circumferential angle $\gamma = 135^\circ$ and $45^\circ$ for RPM = 3000.

It is apparent that in the blade-to-blade region the value of the radial velocity is a function of the circumferential angle.

From the detailed analysis of the velocity distribution a number of conclusions can be drawn. At the circumferential angle $135^\circ$ and RPM = 3000 (tests 1, 2, 3, 4) a jet (i.e., a high velocity region) appears close to the pressure side of the blade trailing edge; the maximum value of the radial velocity of the jet increases with the flow coefficient and it is almost constant transversally in the region between $\frac{x}{l} = .286$ and the hub of the blade. This means that the leakage flow generates a bubble on the front cover which influences the jet flow up to the position $\frac{x}{l} = .286$, while secondary flows apparently do not affect the jet flow field.
The remaining part of the flow field, where the velocities are lower, called "wake", is the region affected by separation on the suction side of the blade. The thickness of this region varies over the blade span, increasing from the tip (front cover) where a reverse flow can be detected, to the hub (rear cover), where positive values of the radial velocities are obtained. Furthermore, two minima in the radial velocity are apparent: the first on the blade suction side, the second at the middle of the passage. However, the radial velocity distribution in the "wake" is strongly non uniform across the blade span and from hub to tip we have: on the hub the second minimum disappears, at mid-span the two minima are almost of the same value; at the blade tip only the mid passage minimum is visible.

Such a complicated flow field emphasizes the importance of leakage and secondary flows in the region of lower radial velocities.

The results of the tests 5, 6, 7, carried out at the circumferential angle 45° and for the same operating conditions as before (see Table 1), are presented in Fig. 15. They show that the blade-to-blade distribution is strongly influenced by the circumferential angle at all the tested flow rates. The main differences between tests 1, 2, 3, 4 and tests 5, 6, 7 is in the jet region: at $\gamma = 45°$ a much lower value for the jet peak was obtained and furthermore the radial velocity in the jet region increases continuously from front to rear cover. The so called "wake" region involves the same flow phenomena as at $\gamma = 135°$. These facts indicate that, at the circumferential angle $\gamma = 45°$, leakage and secondary flows affect markedly the whole blade-to-blade flow field.

Figure 16 compares mean averaged values of the measured radial velocity (tests 1, 2, 3, 4, 5, 6, 7)

$$V_{R_{2m}} = \frac{\int_0^{360} \int_0^{b_2} V_R \, d\theta \, dx}{360 \times b_2}$$
with the theoretical radial velocity

\[ V_{R2T} = \phi U_2 \]

where \( \phi \) is obtained from the measured total flow rate.

At the circumferential angle 45° the radial velocity is decreasing with \( \phi \), while at 135° the slope is positive. The two lines intersect at \( V_{R2T} = 7.6 \) and \( V_{R2m} = 7.9 \). At this point the flow field impeller exit is axisymmetric and therefore one should have \( V_{R2T} = V_{R2m} \) by continuity. Thus, an error

\[ \frac{V_{R2m} - V_{R2T}}{V_{R2T}} = 3.9\% \]

was obtained. This is acceptable considering that the reverse flow between impeller and casing was neglected and that the spanwise integration is performed with only six points.

The results for RPM = 2400 (tests 8, 9, 10, 11) presented in Figs. 17 a, b, c, d, show, as expected, lower values for the radial velocity.

These surveys were made at \( \gamma = 135° \) so that the velocity distribution should be similar to the one obtained in the tests 1, 2, 3, 4, and the averaged radial velocity fulfils the similarity laws.

Table 2 gives a comparison of the percentage of flow rate delivered by the jet and by the wake and the relative dimensions of the jet region.
TABLE 2

<table>
<thead>
<tr>
<th>Test No</th>
<th>$Q_J/Q_P$ (%)</th>
<th>$Q_W/Q_P$ (%)</th>
<th>$A_J/A_P$ (%)</th>
<th>$\phi$</th>
<th>$\gamma$</th>
<th>RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>39.27</td>
<td>60.63</td>
<td>25.00</td>
<td>0.343</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>2</td>
<td>38.44</td>
<td>61.56</td>
<td>22.73</td>
<td>0.267</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>3</td>
<td>37.35</td>
<td>62.65</td>
<td>23.58</td>
<td>0.194</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>4</td>
<td>36.51</td>
<td>63.49</td>
<td>22.50</td>
<td>0.139</td>
<td>135</td>
<td>3000</td>
</tr>
<tr>
<td>5</td>
<td>55.35</td>
<td>44.65</td>
<td>41.67</td>
<td>0.267</td>
<td>45</td>
<td>3000</td>
</tr>
<tr>
<td>6</td>
<td>43.40</td>
<td>56.60</td>
<td>31.91</td>
<td>0.194</td>
<td>45</td>
<td>3000</td>
</tr>
<tr>
<td>7</td>
<td>34.01</td>
<td>65.99</td>
<td>25.51</td>
<td>0.139</td>
<td>45</td>
<td>3000</td>
</tr>
<tr>
<td>8</td>
<td>34.99</td>
<td>65.01</td>
<td>21.43</td>
<td>0.341</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>9</td>
<td>37.52</td>
<td>62.48</td>
<td>22.50</td>
<td>0.294</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>10</td>
<td>33.98</td>
<td>66.12</td>
<td>20.05</td>
<td>0.218</td>
<td>135</td>
<td>2400</td>
</tr>
<tr>
<td>11</td>
<td>38.97</td>
<td>61.03</td>
<td>21.30</td>
<td>0.124</td>
<td>135</td>
<td>2400</td>
</tr>
</tbody>
</table>

where $j$ indicates jet, $w$ indicates wake and $p$ indicates the blade-to-blade passage.

It appears that the flow passing through the jet region is comparable with the flow passing through the wake region. This fact contradicts the results obtained in supersonic or highly subsonic radial compressors (Refs. 6, 7), where the flow carried by the wake is only 20% of the total mass flow. Such a disagreement results from the fact that in a low speed radial compressor the Coriolis force is smaller than high speed ones. Therefore, the separation on the suction side is less noticeable and the relative velocity on the suction side may reach values of the same order of magnitude as on the pressure side.

3.3 Slip factor

The blade-to-blade slip factor distribution, defined as

$$\mu = \frac{V_{\theta_2}}{U_2}$$

is presented in Figs. 18 a,b,c,d for the tests 1, 2, 3, 4; in Figs. 19 a,b,c, for the tests 5, 6, 7 and in Figs. 20 a,b,c,d, for the tests 8, 9, 10, 11.
As expected the slip factor presents a minimum where the radial velocity is maximum and reaches the highest value on the suction side.

Figure 21 gives the mass averaged values of slip factor

$$\mu = \frac{\int_0^{360} \int_0^{b_2} \mu \frac{V_{R_2}}{d\theta} dx}{\int_0^{360} \int_0^{b_2} \frac{V_{R_2}}{d\theta} dx}$$

The mass averaged slip factor does not depend on the circumferential position $\gamma$ but varies with the rotational speed.

At the two circumferential positions tested for RPM = 3000, $\mu$ is approximately the same and between $0.19 \leq \psi \leq 0.27$, it can be assumed constant, with a value:

$$\mu = 0.90.$$

At RPM = 2400, the mass averaged slip factor is almost constant for all the tested flow rates

$$\mu = 0.91.$$

A very good agreement was found with Stiefel's nomogram (Ref. 10), which gives $\mu = 0.87$.

### 3.4 Blade-to-blade turbulence distribution

A mid span survey was carried out to measure the turbulence components $\sqrt{\overline{u^2}}$, $\sqrt{\overline{v^2}}$, and the shear stress $\overline{uv}$ over the blade-to-blade passage. The test conditions were:

- RPM = 3000
- $\gamma = 135^\circ$
- $\phi = 0.343$
- $\psi = 0.451$. 
The results are plotted in Figs. 22 a,b,c.

It appears that the intensity of the fluctuation component parallel to the absolute velocity \( \sqrt{u^2} \) (Fig. 22-a) reaches a minimum value when the mean absolute velocity \( V \) (Fig. 23) is maximum and vice versa. This minimum is even more evident for the relative intensity \( \sqrt{u^2}/V \).

The minimum in the parallel fluctuation component also corresponds to the minimum of the radial velocity component \( V_R \) (Fig. 24) whereas the maximum corresponds to the blade position.

The variations of the fluctuation component perpendicular to the mean absolute velocity vector \( \sqrt{v^2} \) are related to the radial velocity component. In the jet (\( V_R \) maximum) \( \sqrt{v^2} \) is maximum and in the suction side \( \sqrt{v^2} \) is minimum. The minimum in \( V_R \) at the middle of the passage does not influence the perpendicular fluctuation component.

The shear stresses (Fig. 22-c) are maximum in the jet and minimum at the blade trailing edge. This result is in very good agreement with the conclusion obtained by Johnston (Ref. 11) in his analysis of the effects of a rotation on boundary layers. The pressure side of the blade is a region of unstable flow, and thus turbulence producing, while the suction side is stable and thus act as a damper of turbulence.
4. CONCLUSIONS

A hot wire anemometer coupled with a conditional sampler was used to investigate the flow field at the impeller exit of a low speed centrifugal compressor. The blade-to-blade distribution of the absolute velocity, of the velocity fluctuations and of the shear stresses were measured.

Surveys were performed at different operating conditions and at two circumferential angles.

4.1 Detailed test results

Detailed analysis of the results showed a strong three dimensional flow field and the existence of strong secondary and leakage flows. Nevertheless, it was easy to identify a jet and wake configuration. The jet flow field appeared to be quite different from the one determined for high speed radial compressors. In the tested low speed radial compressor wheel, the mass flows were of the same order of magnitude in the wake and in the jet.

The so-called wake flow field is strongly affected by secondary and leakage flows, the jet, on the contrary, is only influenced by leakages.

Velocity fluctuations and shear stresses were evaluated. The distribution of the shear stresses over the pitch points out the action of the Coriolis force on the flow development.

4.2 Mass averaged flow characteristics

The mass averaged radial velocity is function of the circumferential angle and it follows the similarity laws.

The mass averaged slip factor is constant over a wide range of flow rates, it is independent on the circumferential angle, while it slightly increases with decreasing rotational speed. It was found to be consistent with Stiefel's nomogram.
REFERENCES


SCHEMATIC OF SEPARATED FLOW MODEL

FIG. 1

REL. FLOW PATTERN

Backflow model at centrifugal impeller exit

FIG. 2
FIG. 4

RPM

△ 1800
○ 2400
● 3000
$\frac{V}{V_0}$

\(\alpha\) 0 20 40 60 (°)

FIG. 6
FIG. 16

\[ V_{R2T} \]

\[ V_{R2m} \]
FIG. 17 a

TEST 8
FIG. 20 b
RPM

○ 3000.  135°
△ 3000.  45°
□ 2400.  135°