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DEVELOPMENT AND FIRST APPLICATION OF A SINGLE HOLE FAST RESPONSE PRESSURE PROBE

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ABSTRACT

The paper presents design and first application of a cylindrical miniaturized single hole fast response pressure probe for aerodynamic measurements of 2D flow field in turbomachines.

For the aim of a wide Italian Inter-University two-year research programme about unsteady effects in high and low pressure axial turbines, a new test rig has been designed and built up. The facility located at Politecnico di Milano allows the installation of a high pressure stage of a modern axial turbine, specially designed for the purposes of the research. Both the test rig and the turbine stage are briefly described.

The probe - that in the final configuration will be applied downstream of the turbine stage - is equipped with standard and commercially easy available fast response miniaturised pressure transducer, in order to limit the cost of manufacturing and avoid the problem of pressure sensor chips handling and bonding. Thus, one of the goals of this paper is to verify the maximum frequency response that this kind of pressure probe, characterised by low cost and easy manufacturing, can reach. The method and results of both theoretical and experimental approaches for the evaluation of the dynamic behaviour of the probe are reported, showing a frequency response of about 12 kHz. A first application of the probe to a ventilated high performance automotive brake is also reported; flow field measurements carried out downstream of the impeller show good agreement with the ones obtained by means of hot-wire anemometry.

NOMENCLATURE

C	= speed of sound
d	= feeding hole diameter
D	= probe head diameter
f	= frequency
h	= impeller blade height
hw	= hot wire
L	= probe head feeding line length
M	= Mach number
P	= pressure
PP	= pressure probe
Re	= Reynolds number
S	= probe head feeding line cross section
V	= Volume of probe head inner cavity
u	= tangential velocity
z	= axial direction
θ	= tangential direction

Subscrits

p	= probe or pitch
t	= total
ref	= upstream reference
2	= outlet

INTRODUCTION

The real flow in turbomachines is in general unsteady. This is due to several effects like passing wakes, blade row interaction, shock waves, rotating stall, etc (Tiedmann and Kost, (1998); Sharma, Pickett and Ni (1992)). In particular the flow field downstream of a blade row is not uniform in space and time and this leads to a time dependent variation of total pressure, total temperature and flow angle for the subsequent row. Thus a different stage performance may be expected related to the above mentioned phenomena (Ken-ichi et al, (1997), Hodson and Dawes (1996)).

For the aim of the comprehension of these mechanisms a wide research program involving seven Italian turbomachinery Laboratories and supported by the Italian Ministry for the University and Scientific Research (MURST), is in progress. The program foresees both a low pressure (Arnone et al., (2000)) and a high pressure stage design, and the design and manufacturing of a fast response pressure probe. Authors' units are working both in the start up of test rig for high pressure axial turbine stages and in the design of a high frequency response pressure probe, suitable for application in both models.

Unsteady velocity measurements in turbomachines can be carried out by means of several techniques; whereas the unsteady pressure field can only be measured by means of fast response pressure transducers (Sieverding et al. (2000)).

Since many years several authors have been working on the development of fast response pressure probes based on the application of one or more miniaturised pressure sensor chips. (Gossweiler et al., (1990) and (1995); Brouckaert et al. (1999); Ainsworth et al. (1995)). The main disadvantage of multisensors probes is related to the higher dimension, whilst single sensors probes require to be rotated in order to resolve the 2D phase averaged flow field.

This probe technology manufacturing requires long experience and ability; therefore, time and costs required for the internal development of a high frequency response pressure probe are very heavy. At present time, no one fast response pressure probe is commercially available.

In order to overcome the above-mentioned difficulties, authors developed a single sensor cylindrical fast response pressure probe based on commercially available fast response pressure transducer. The paper points out limits and benefits of this easy and low cost choice.

THE POLITECNICO DI MILANO CLOSED LOOP TEST RIG

For the aim of the above mentioned research program at the Laboratorio di Fluidodinamica delle Macchine of the Politecnico di Milano a new closed loop test rig for axial and centrifugal turbomachines has been recently started up. Figure 1 reports the main functional diagram of the test rig while Table 1 presents the main operational characteristics. The facility can operate under closed conditions or forcing the inlet axial section to atmospheric pressure, in order to save the power required at the compressor engine while the turbine is operated in the axial section. Moreover, the electrical

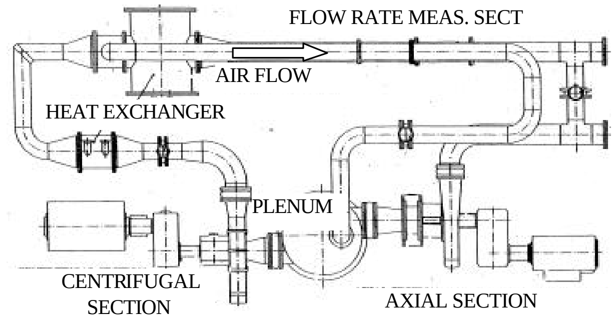


Figure 1: Politecnico di Milano closed loop test rig

power generated at the brake is reversed to the centrifugal section engine leading to significant savings in the electrical running costs.

The test rig is instrumented in order to allow the over-all performance evaluation of both turbomachines installed. Both sections are equipped with traversing systems for the measurement of the flow field inside and downstream of the turbomachines; optical accesses for LDV measurements are also provided.

The two reversible DC engines allow for a continuous rotational speed regulation. The centrifugal section is equipped with an impeller whilst the axial section is usually equipped with an axial turbine, but an axial compressor can also be installed.

	Engine/ brake	Max. outlet diameter	Max. rotational speed
Centrifugal Section	800 kW, 600 V DC	560 mm	45.000 rpm
Axial Section	400 kW, 540 V DC	400 mm	20.000 rpm

Table 1: Politecnico di Milano closed loop test rig main operational characteristics.

THE HP TURBINE AXIAL STAGE

For the purposes of the MURST research program concerning the analysis of unsteady effects on axial flow turbines, a model of a typical high pressure turbine stage has been designed and manufactured. The model will be installed in the axial section of the closed loop test rig.

The stage has been designed in co-operation with ABB-ALSTOM POWER Turbine Aerodynamic Unit of Baden (CH). For the preliminary definition of the stage geometry, a numerical computational tool for the evaluation of the flow field in the meridional plane, implemented by semi-empirical correlation within the channels, has been applied. A final analysis of the flow field has been performed by means of a fully viscous 3D steady computation on the whole stage.

Main geometrical constrains of the design and requirements of a quasi 2D flow field at blade mid span led to a significant blade twisting and to a positive vane leaning of 12 degrees.

Table 2 reports about main stage characteristics, while Figures 2 and 3 present vane and blade geometry and their main characteristic.

Turbine stator can be rotated with respect to the measuring section; axial gap between the two blade rows can be varied in order to evaluate the influence of this parameter.

Expansion ratio	2.47
Flow rate	2.67 kg/s
Rotational speed	12.000 rpm
Outlet power	210 kW
Degree of reaction at mid span	0.433
Tip diameter	400 mm
Hub diameter	300 mm

Table 2: High pressure turbine stage characteristics

THE FAST RESPONSE PRESSURE PROBE

Generalities and static calibration

The fast response probe here presented is the evolution of a cylindrical single sensor pressure probe already developed for steady measurements (Dossena and Osnaghi, (1998)). The probe head is equipped with a miniaturised fast response Kulite pressure transducer (XCQ-062) whose natural frequency response is 330 kHz. The sensor is flush mounted by manufacturer in a stainless steel cylindrical tube of 1.65 mm diameter. The whole cylinder is then installed in the probe head, whose geometry is reported in Figure 4.

The probe head is brass made and it is obtained by means of mechanical micro machining. The sensitive area of the pressure sensor, which is a square of 0.9 mm side, faces a cylindrical cavity of 0.3 mm height and 1.4 mm diameter. A hole of 0.3 mm feeds the inner cavity.

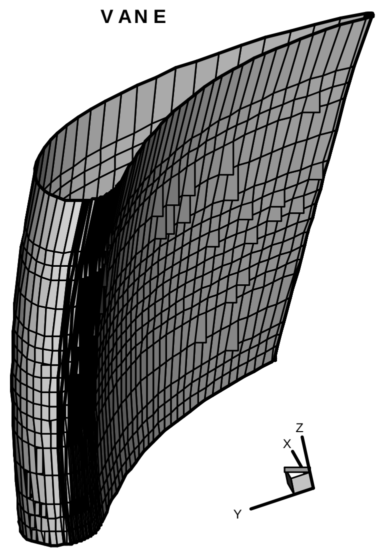
The design leads to a very easy and low-cost probe, but, as a consequence, the quite large volume facing the sensor limits probe frequency response, being this feature strictly related to the geometry of the line-cavity system.

In order to evaluate the frequency response of the probe both a theoretical and an experimental method will be presented in the following.

The application of the cylindrical single hole probe requires the rotation of the probe around its own axis in order to achieve information about the pressure distribution around the cylinder. As well known, the flow separation around the cylinder occurs at different angular positions depending on both Reynolds and Mach Numbers; this is showed in Figure 5 in terms of total pressure coefficient distribution (K_{pt}):

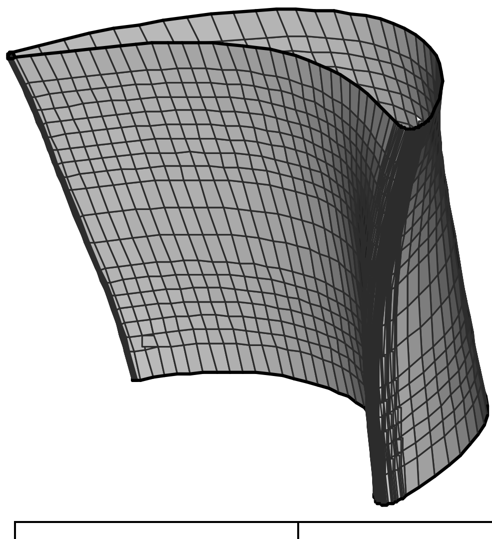
$$K_{pt} = \frac{\left(P_{t,ref} - P_p\right)}{\left(P_t - P_s\right)_{ref}}$$

Figure 5 shows that the separation occurs for a minimum rotation angle of about ± 72 degrees for Mach Numbers ranging from 0.2 up to 0.85. If we assume a



VANE LEAN = 12°	PROFILES N° = 22		
	HUB	MS	TIP
Axial chord	30.6 mm		
Pitch/ Axial chord	1.394	1.606	1.858
Outlet Mach Number.	0.929	0.873	0.791
α out	72.5°	75.2°	77.5°

Figure2: Main vane characteristics



PASSAGE FREQ. =5 kHz	PROFILES N° = 25		
	HUB	MS	TIP
Axial chord	46.7 mm		
Pitch / Axial chord	0.804	0.938	1.071
β in	56.89	47.6	25.48
β out	-67.7		
Outlet Rel. Mach Number.	0.848	0.867	0.854
Outlet Abs. Mach Number.	0.402	0.369	0.331

Figure3: Main blade characteristics.

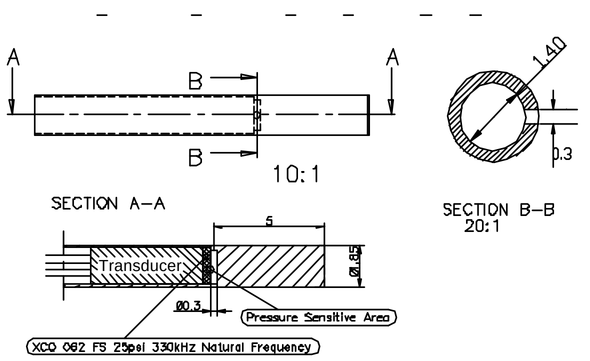


Figure 4: Cylindrical pressure probe head geometry

minimum of 3 rotations of 45 degrees (whose corresponding pressure readings will be called P_l, P_p, P_r in the following) for the evaluation of the local flow field, the angular range of the probe results in ± 27 degrees. This value is given by the difference between the separation angle and the first useful P_p position.

In order to obtain the yaw angle and the total and static pressure of the local flow field, the definition of two extra coefficients is required:

$$K_{ps} = \frac{\left[P_{t,ref} - (P_l + P_r)/2\right]}{(P_t - P_s)_{ref}}$$

and

$$K_{yaw} = \frac{(P_l - P_r)}{(P_t - P_s)_{ref}}$$

Because the three coefficients are also functions of the Mach Number, twelve tests at different Mach Number, in the range 0.1 - 0.85, have been performed. The angular step rotation during calibrations was 1 degree.

Static flow field measurement uncertainties are reported in table 3 as a percentage of the local value.

P_t	P_s	Yaw
± 0.2 %	± 0.7 %	± 0.5 deg

Table 3: Measurement uncertainties.

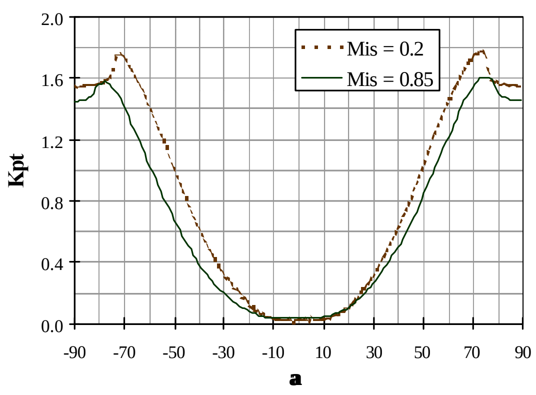


Figure 5: Total pressure coefficient distribution for 180 degrees of probe rotation

Frequency response evaluation

Theoretical Approach

This approach has been widely applied during the design process in order to evaluate the influence of probe head geometry on frequency response.

Because the main assumption of the Helmotz theory is that the axial volume length has to be negligible with respect to the line length, unfortunately, this easy approach cannot be applied in this case.

The method applied by authors is based on the use of a CFD code (Bassi et al. (2000)), allowing the integration of the 1D Euler Equations - taking into account for transverse section variations - by means of a 5th order accuracy in space and 3rd order accuracy in time Discontinuous Galerkin Finite Element Method.

The time response to several shapes of periodic pressure fluctuation at the feeding line inlet have been investigated, in order to simulate the passage of blade wakes characterised by different frequencies, amplitudes and pick values. Anyway the most significant result is reported in Figure 6, showing the computed line cavity-system response to a 10% pressure step at the inlet line section. The plot evidences the time trend of pressure in four representative sections of the system. After some oscillations, the whole system tends to the maximum pressure value: the time required for a pressure recovery equal to the 99% of the pressure step is 83 μs, corresponding to a frequency response of 12 kHz. This means that time varying pressure fluctuations, whose frequency spectrum is contained below 12 kHz, can be well captured by the pressure sensor without any distortion in terms of amplitude and phase. Other simulations based on different amplitude steps have shown very similar results.

Experimental Approach

In order to experimentally evaluate the frequency response of the probe and to validate the results given by the computational code, a pressure pulse generator has been designed (Figure 7). A plenum, fed with pressurised air, is equipped with two convergent nozzles of 5 mm diameter, discharging in front of a disc driven by an electrical engine. The disc is equipped with 60 holes exactly facing the nozzle outlet. Holes diameter is 5 mm, and holes spacing is 10 mm. The alternation of equally spaced solid and void leads to a pulsating flow field downstream of the holes circle. The engine allows a maximum rotational speed of 5000 rpm, giving two trigger signals: 500 pulse/revolution and 1 pulse/revolution.

The cylindrical pressure probe is positioned in the centreline of the jet stream discharged by the nozzle, just 5 mm downstream of the rotating disk. In order to evaluate the dynamic behaviour of the cylindrical probe, the reconstructed pressure field is compared with the one obtained by a fast response total pressure probe equipped with a high frequency response flush mounted pressure

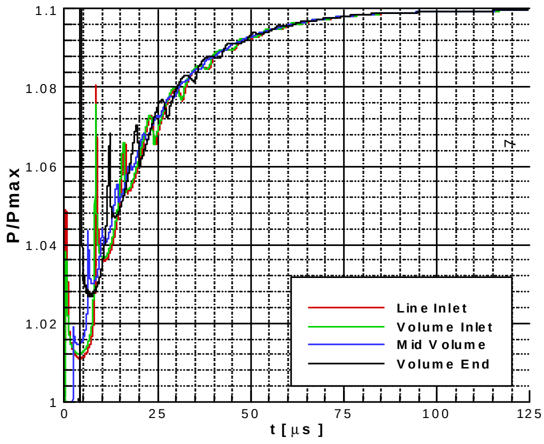


Figure 6: Numerical simulation result to a 10% pressure step at feeding line inlet

transducer.

Both signals are acquired by means of a 12-bit data acquisition card allowing for a maximum sampling frequency of 5 Msamples/s on 4 independent channels. 600.000 samples have been acquired for each test condition, at a frequency of 1 MHz for both pressure signals. A phase locked data acquisition and ensemble-averaging technique gives a number of 120 points per hole pitch, each one being the average of about 5000 measuring samples, depending on the rotational speed.

Figure 8 reports the comparison between total pressure probe and cylindrical probe results for a hole passage frequency of 2 kHz. The plot shows good agreement of the two pressure pulse shapes; the slight shift of the two signals can be related to a small difference in the position of the two sensors or to a small

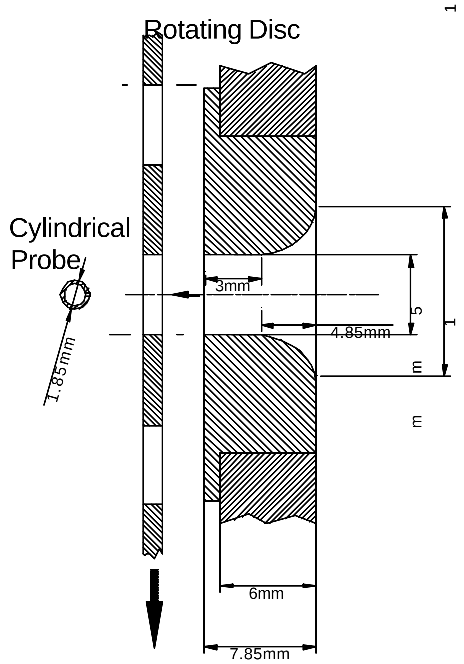


Figure 7: Pressure pulse generator

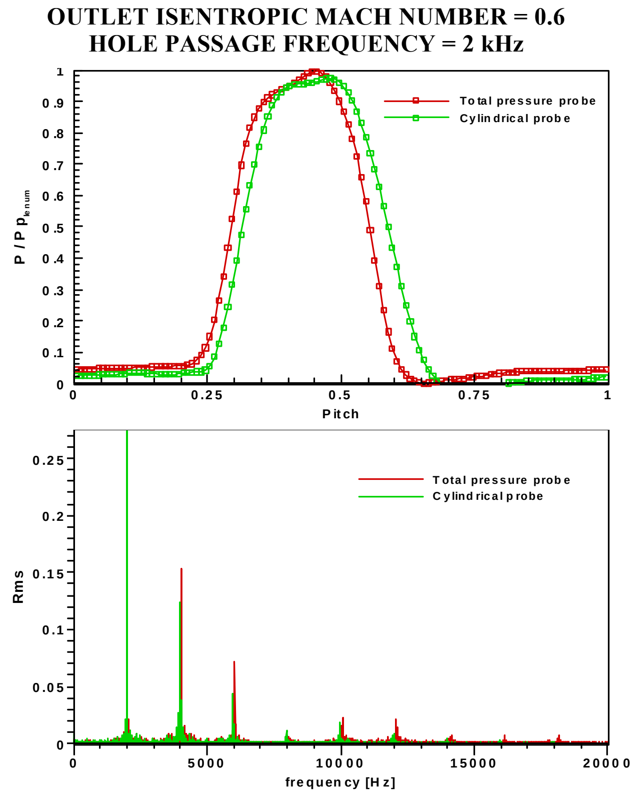


Figure 8: Total pressure distribution and RMS Spectrum downstream of the pressure pulse generator from cylindrical and total pressure probe

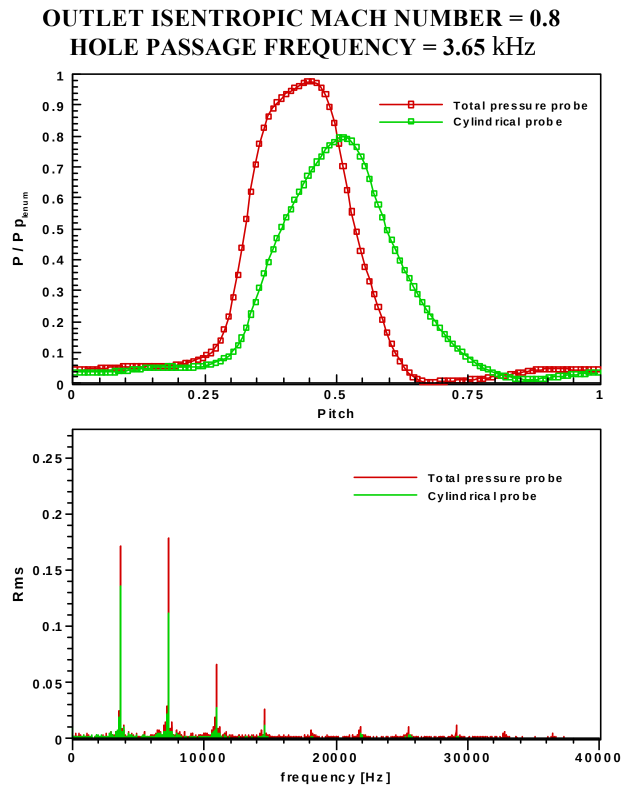


Figure 9: Total pressure distribution and RMS Spectrum downstream of the pressure pulse generator from cylindrical and total pressure probe

delay introduced by the line-cavity system. Moreover, the RMS spectrum in Figure 8 does not evidence significant harmonic components for frequencies over 13 kHz.

Unfortunately, the cylindrical probe accuracy lacks for higher frequencies as reported in Figure 9, where results for a hole passage frequency of 3.65 kHz are compared, showing a big difference in shapes and pick values. This maybe related to the fact that the pressure signal is characterised by a spectrum containing harmonics over 12 kHz, that the cylindrical pressure probe is not able to correctly capture. This is supported by Figure 9 where the comparison of the two RMS spectrums computed from the signals acquired by the two probes is presented.

Moreover, this is also confirmed by the pressure shape obtained by anti-transforming the signal acquired by the total pressure probe after forcing to zero all the harmonic components over 12 kHz. In fact, this operation leads to a shape that resembles with good accuracy the one of the cylindrical pressure probe.

As a final consideration after the experimental tests one can state that:

- The pressure probe is able to well describe phenomena with a frequency spectrum contained within 12 kHz without any need for numerical corrections;
- Test evidence good agreement with the results obtained by means of the theoretical approach presented in the previous paragraph.

The higher Mach number characterising the test reported in Figure 9 is required in order to keep a constant ratio between the hole speed and the jet speed and does not influence the above mentioned conclusions.

FIRST APPLICATION TO A VENTILATED AUTOMOTIVE BRAKE

The new fast response cylindrical probe has been applied for the first time downstream of a high performance ventilated brake.

The brake disc geometry, reported in Figure 10, is very similar to the one of a centrifugal impeller. The disc, whose discharge diameter is 340 mm, is equipped with 34 blades and during tests it was operated at 3000 rpm, this leading to a blade passage frequency of 1.7 kHz.

The measuring plane is located 3.8 mm downstream of the external diameter and 9 traverses were carried out in order to cover 120% of the blade height.

Measurements were performed both by means of a single wire constant temperature hot wire anemometer and the cylindrical pressure probe. Both sensors were rotated around the stem axis in order to get the absolute 2D flow field.

Both signals are acquired by means of a 12-bit data acquisition card allowing for a maximum sampling frequency of 1.25 Msamples/s. Data acquisition frequency for both signals was 50 kHz and the number of acquired samples per test was about 260.000. A phase locked data acquisition and ensemble-averaging technique gives a number of 28 points per blade pitch, each one being the average of about 9000 measuring samples.

Measurement uncertainties are reported in Table 4 in terms of percentage of local velocity value not taking into account for dynamic effect.

	Velocity	Flow angle
Hot Wire	± 3 %	± 1°
Cylindrical pressure probe	± 2.7 %	± 1°

Table 4: Measurement uncertainties

Figure 11 shows the velocity contours on the measurement plane obtained by means of hot wire and cylindrical pressure probe. All flow field characteristics are well described by the cylindrical pressure probe. The position of the wake generated by the blades, corresponding to the peak values in Figure 11, is well captured and so is the low velocity region due to the shroud and hub boundary layers.

The pressure probe measurements evidence a higher peak value and a corresponding higher velocity gradient in the wake region. These differences might be related to a different spatial resolution of the two probes.

This consideration is also supported by the RMS spectrum reported in Figure 12 where the spectrum obtained from the hot wire evidences a lower frequency content if compared to the one obtained from the pressure measurements. This feature is consistent with the higher gradients evidenced by the velocity contours from pressure probe measurements.

Figure 13 shows the discharge angle contours obtained from hot wire and pressure probe measurements. Even in this case the two distributions show good agreement. Some differences can be noticed in the wake region: they can be related to the difference in velocity gradient already pointed out in Figure 11.

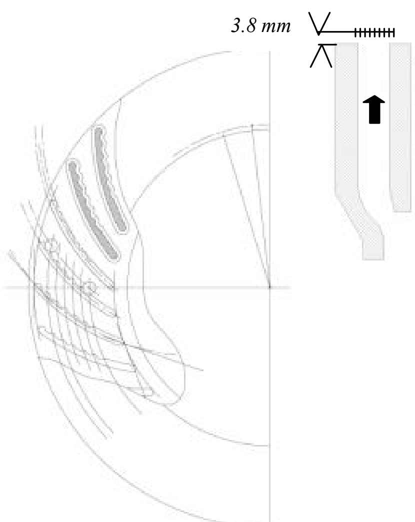


Figure 10: The ventilated automotive brake and traverses position.

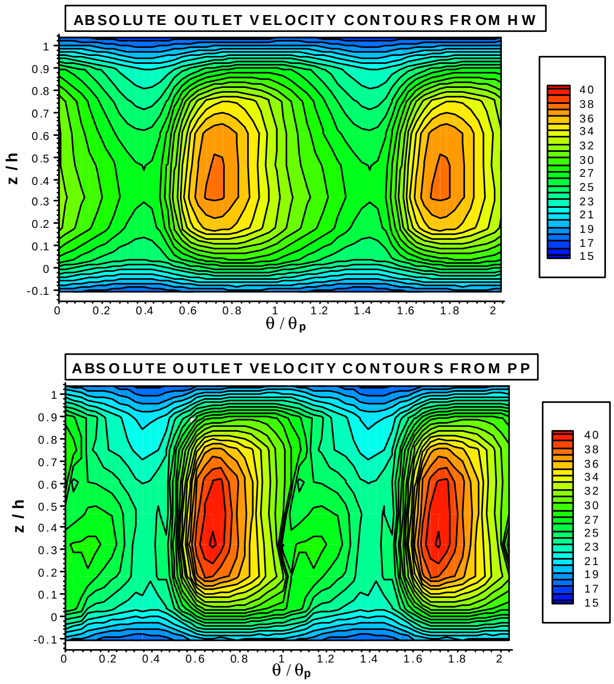


Figure 11 ; Absolute velocity contours downstream of brake disc (rpm = 3000)

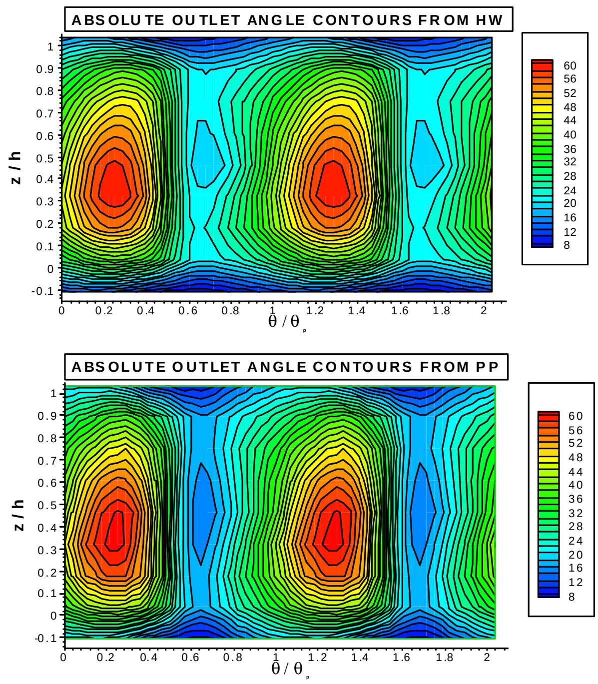


Figure 13 ; Absolute discharge angle downstream of brake disc (rpm = 3000)

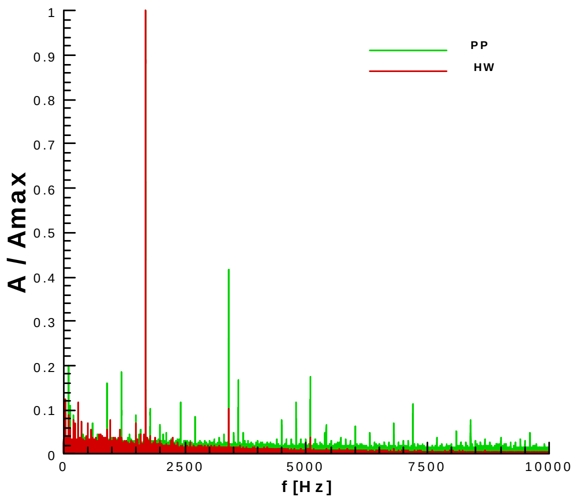


Figure 12;Non dimensional amplitude spectrum from hot wire and pressure probe measurements

As expected, high values of the discharge angles are evidenced in the "jet" zone corresponding to the low absolute velocity region.

CONCLUSIONS

A new test rig for turbomachines is now available at the Politecnico di Milano. The first application will be the study of unsteady effects in a typical high pressure turbine stage, in the frame of an Inter-Universities Research Program.

A cylindrical fast response pressure probe design has been presented.

The probe is equipped with a commercial available high frequency pressure transducer, leading to an easy and low cost manufacturing.

The probe frequency response is about 12 kHz, as results from computational code and validated by the results obtained from the application in an experimental facility.

Measurements obtained by means of the single hole pressure probe downstream of a high performance ventilated brake, very similar to a low pressure centrifugal impeller, have shown good agreement with the ones obtained by means of hot wire anemometry.

In order to improve the frequency response of the probe, new probe head geometry design is under course. Furthermore, a dynamic calibration facility, based on shock-tube technique, will be designed and built up. The results will allow the definition of a transfer function for the extension of the probe application to a frequency range wider then the one limited by the frequency response.

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REFERENCES

Ainsworth R. W., Allen J.L., Batt J.J.M., (1995), "The development of fast-response aerodynamic probes for flow measurements in turbomachinery", ASME Journal of Turbomachinery, Vol. 117/4.

Arnone, A.et al., (2000), "A two-stage low pressure gas-turbine model for rotor-stator aerodynamic interaction investigations", Proc. of 15th Symposium on Meas. Tech. For Transonic and Supersonic Flows in Cascades and Turbomachines, Firenze.

Bassi, F. et al., (2000), "Numerical simulation of a reciprocating compressor for household refrigerators", Proc. Of 15th International Compressors Engineering Conference, Purdue, West La Fayette, Indiana, USA.

Brouckaert J.F., Sieverding C.H., Manna M., (1998), "Development of fast response 3 hole pressure probe", Proc. of 14th Symposium on Meas. Tech. For Transonic and Supersonic Flows in Cascades and Turbomachines, Limerick.

Dossena V., Osnaghi C., (1998), "Analisi del campo di moto in prossimita' del bordo di uscita di una pala di turbina a gas con iniezione di refrigerante", Proceedings of 53° Congresso ATI, Firenze.

Gossweiler C., Humm H.J., Kupferschmied P., (1990), "TheUse of Piezo-Resistive Semi-Conductor Pressure Transducers for Fast-Response Probe Measurements inTurbomachinery", Proc. of 10th Symp.on Meas. Tech. For Transonic and Supersonic Flows in Cascades and Turbomachines, VKI , Brussels.

Gossweiler C., Kupferschmied P., Gyamarthy G., (1995), "On Fast-Response Probe, Part 1: Technology, Calibration and Application to Turbomachinery", ASME Journal of Turbomachinery, Vol. 117/4.

Ken-ichi F., Yoshinori S., Tadashi T., (1997), "Experimental Studies on Unsteady Aerodynamic Loss of a High-Pressure Turbine Cascade", ASME Paper 97-GT-52

Sharma O.P., Pickett G.F., Ni R.H., (1992), "Assessment of Unsteady Flows in Turbines", ASME Journal of Turbomachinery Vol. 114/79-90.

Sieverding C.H., Arts T., Dénos R., Brouckaert J.-F., (2000), "Measurement tecniques for unsteady flows in turbomachines", Experiments in Fluids 28, 285-321, Springler-Verlag.

Tiedemann M., Kost F., (1998), "Unsteady boundary layer transition on a high pressure turbine rotor blade", ASME Paper 99-GT-194.

Hodson H.P., Dawes W.N., (1996), "On the Interpretation of Measured Profile Losses in Unsteady Wake-Turbine Interaction Studies", ASME Paper 96-GT-494.