FLOW AND TURBULENCE SURVEY FOR A MODEL OF GAS TURBINE EXHAUST DIFFUSER

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ABSTRACT

This paper presents the results of an extensive set of measurements on a model of an exhaust diffuser for gas turbines. The diffuser is of the straight-wall annular-axial type, typically employed in small-to-medium size gas turbines. It features six high-solidity struts, which support, in the real machine, one of the shaft bearings and have piping for oil supply inside.

The 35%-scale model has been tested on a special test stand developed at the University of Perugia, using the suction side of a centrifugal-flow industrial fan of suitable capacity. Inlet speed is around 80 m/s, allowing satisfactory accuracy for flow measurements and the similarity in terms of Reynolds number.

The instrumentation, the movement of the measurement point and data acquisition system were designed for automatic running of the tests. Both pneumatic and hot-wire or hot-film probes can be used on the same facility. The same wind tunnel, previous a quick replacement of the model with a probe calibration test section, can be used for calibration of both pneumatic and hot-wire/hot-film probes. A three hole directional pneumatic probe was used for stationary flow measurements to determine the global performance parameters of the model and a split-film probe was used to determine the turbulence characteristics.

For four test sections, contour plots are produced of average velocity components, flow angle and turbulence quantities as three components of the Reynolds stress tensor.

NOMENCLATURE

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INTRODUCTION

Performance of diffusers is generally determined by using global parameters and relationships which correlate static and/or total pressure at inlet and outlet. The design of all types of diffusers is generally based on performance maps and on the "minimum divergence angle" rule (Kline, et al., 1959). However, small divergence angles do not always allow a high pressure recovery as short diffusers are generally necessary to limit the overall length of the system.

Axial compressors and turbines blades also produce a highly turbulent flow at inlet and the endwalls generate a skewed velocity profile (Pfeil and Going, 1987). In addition, annular diffusers may feature structural elements which support the hub and influence the flow field and the pressure recovery. The distortion of the inlet flow and the presence of obstacles in the diffuser are not considered in the performance maps.

Of interest is also the off-design behaviour of diffusers which can not be predicted by global correlations neglecting turbulence and 3-D effects produced by swirled inlet conditions.

MODEL DESCRIPTION

The diffuser which has been characterized in the present work is a 0.35:1-scale model of the exhaust diffuser of the Nuovo Pignone PGT10 gas turbine. The GT diffuser has six high-solidity struts which support one of the shaft bearings and have piping for oil supply inside. The inner diameter of the GT diffuser is slightly decreasing in the direction of the exhaust gas.
The model features 24 straight guide vanes at the inlet, which have a dual function: to generate a wake similar to that produced by the blades of the last turbine rotor and to control the inlet swirl in order to simulate diffuser off-design operation when the GT exhaust velocity is not in the axial direction.

Inner and outer diameter of inlet section are 190 and 320 mm respectively. At the outlet, the outer diameter is 420 mm, resulting in an aperture angle of 6.7°. The area ratio of the diffuser is 1.53 and the length/width ratio is 2.01. The axial length of the diffuser from the inlet guide vanes to the outlet is 340 mm. Measurement sections are located at 6, 90, 220, and 310 mm after the guide vanes.

FIGURE 1: MODEL OF GAS TURBINE EXHAUST DIFFUSER

The four automated carriages inside the inner hub and the external stepping motor, which allows its rotation, are controlled by a dedicated board on a personal computer.

The model is connected to a 22 kW centrifugal-flow industrial fan, and the flow rate is controlled by a louver at fan discharge. The suction of the model is connected to a large settling chamber which includes air filters.

The model was designed to operate in geometric and Reynolds number similarity with the GT diffuser. In fact, the GT diffuser, operating with 41.3 kg/s at 462 °C, has a Reynolds number of over 10^6, whereas the Reynolds number is higher than 6*10^5 in the model, with ambient temperature air. However, the Reynolds number in the model is high enough to assume similar flow conditions in the GT diffuser and in the 35%-scale model.

MEASUREMENT TECHNIQUES

Two different measurement techniques were employed in the model with 2-D pneumatic and split-film probes. The 2-D pneumatic probe was used to perform stationary measurements which were used to determine the global performance of the diffuser. The split-film probe has been used to measure mean velocity components, flow angle and turbulence quantities such as Reynolds stress components to detect wake decay after the inlet guide vanes and the struts.

The 2-D pneumatic probe is of the "Cobra" type, specially assembled to be used inside the model. The probe is connected to piezo-electric pressure sensors, whose voltage output is read by a 16-channel fast sampling board (100 kHz), connected to a PC and featuring a 12 digit A/D converter. The sampling board frequency of pressure sensors acquisition has been set to avoid the influence of pressure fluctuations in the measurement of average quantities.

The split-film probe (DANTEC 55R57) can be positioned in the same locations as the pneumatic probe, thus allowing measurement of turbulence quantities, where average quantities are measured. This type of probe is not sensitive to radial velocity fluctuations and the measurements can be assumed completely 2-dimensional. The dual-channel anemometric system by A.A. Labsystems includes amplification and filtering for each channel. The anemometer output is also connected to the mentioned sampling board and the results are elaborated by a PC.

The similarity of flow conditions among measurement made at
different times is guaranteed by keeping track of ambient conditions and keeping the centrifugal fan flow rate constant. This assumption also allows to make measurements with only one probe at a time thus preventing the negative influence of a simultaneous presence of more probes inside the flow field.

Results were produced for one 60° sector containing one strut in the middle because the flow field repeats every 60°.

AVERAGE FLOW QUANTITIES

To describe the flow field in the diffuser, a first set of measurements has been performed to determine the average flow quantities such as mean velocity components, flow direction and total pressure losses. This is important to detect the influence of the guide vanes and the struts on the flow and their interactions.

Figure 3 shows the contour plots of the axial component of the velocity, at the four sections (B to E), where the wakes of the guide vanes and the struts can be easily detected. In section B the presence of the strut at 30° modifies the wake of the guide vane, as it causes an acceleration of the fluid which alters the wake since the beginning (Figure 3.a). That is possible because the velocity is still over 60 m/s in a large part of section C located between the stagnation zone at the leading edge of the strut and the wakes of the guide vanes which can be easily located at 15° and 45° (Figure 3.b).

In section D, a wide area is affected by the wake of the strut, but some traces of the guide vanes are still detectable (Figure 3.c). This traces are not negligible because they seem to cause the onset of flow separation at the shroud. This means that the struts only accelerate the flow between them and that the wake they produce does not affect significantly the other wakes.

Moreover, no separation is detected before section D. The region behind the strut and close to the hub (n.m.) has very high turbulence level and some flow inversion, and no measurements were possible with the pneumatic probe.

An evident zone of separation can be observed between the strut wakes and the adjacent guide vanes wakes, in proximity of the shroud (Section E, Figure 3.d). As the angle of the diffuser is constant over the length, the interaction of the wakes is definitely important for the development of the separation at the shroud.

Along the hub, moving from section B to E, it is possible to observe a considerable reduction in the velocity in a wider and wider region. Differently from the region close to the shroud, this can be mainly attributed to the growth of the boundary layer and not to the interaction of wakes.

Total pressure losses confirm the presence of the wakes generated by inlet vanes and struts (Figure 4).

In section C, the stagnation line along the leading edge is clearly identified by a decrease in velocity without a significant pressure loss (Figure 4.b). After the strut a large wake can be observed (Figure 4.c), and there is a generalized pressure loss at diffuser outlet, with small traces of the strut and inlet vanes wakes and with maxima in the regions close to the hub and the shroud, where separation occurs (Figure 4.d). Following the direction of the flow, it is interesting to note how the region with pressure losses lower than 20 mmH₂O shrinks, until it practically disappears in Section E.

The flow direction is also affected by the presence of the guide vanes and the struts in a very small region behind them. However deviation caused by the struts never exceeds ±12° from the axial direction, and no deviation is detectable in wide regions of all the sections.

FIGURE 3: CONTOUR PLOTS OF AXIAL VELOCITY.
PERFORMANCE CHARACTERISTICS

Different performance parameters have been defined to evaluate the performance of diffusers (Rundstadler, et al., 1975, Kline, et al., 1959, Japikse, 1986).

The ideal pressure recovery coefficient is determined by the area ratio of the diffuser:

\[ C_n = 1 - \left( \frac{A_1}{A_2} \right)^2 \]  

(1)

The 2D pneumatic probe has also been used to determine the following global performance parameters. Performance curves are generally plotted as contour lines of the real pressure recovery coefficient \( C_r \) versus length/throat diameter and area ratio (Kline, et al., 1959):

\[ C_r = \frac{p_{st2} - p_{st1}}{\rho U_1^2} \]  

(2)

A diffuser efficiency has been defined as:

\[ \eta_D = \frac{C_r}{C_n} \]  

(3)

whereas a useful parameter which has a linear relationship with \( C_p \) is the following (Japikse, 1986):

\[ K = C_n - C_r \]  

(4)

| \( C_r \) | 0.501 |
| \( K \) | 0.211 |
| \( \eta_D \) | 0.704 |

Most of the performance maps have been built for diffusers with high length/throat diameter ratios. However \( C_r \) and \( K \) have resulted in agreement with Japikse (1986).

TURBULENCE MEASUREMENTS

Measured turbulence quantities are the axial and tangential components of velocity fluctuations and their product. Samples of a given number of acquired data have been measured, and their average and rms calculated. The rms of the two velocity components of each sample (turbulence intensity) is then normalized to mean velocity. Therefore the turbulent components of velocity are the following:

\[ \frac{\sqrt{u'^2}}{U} \]  

and \[ \frac{\sqrt{v'^2}}{U} \]  

(5)

Turbulence at inlet is lower than 2% of average velocity over the largest part of section B (Figure 5.a). The presence of inlet guide vanes generates a higher level of turbulence whose axial component may reach 6%. The tangential component does not exceed 3% (Figure 6.a); this means that turbulence generated by inlet vanes is quite isotropic.

Between the guide vanes and the strut there is a general increase in the turbulence level with respect to the average velocity, due to the spreading of the wakes of the inlet vanes. However the maximum values of turbulence are comparable to those in section B (Figure 5.b). The main effect of the strut is the conversion of fluctuations from the axial direction to the tangential direction, due to the deviation of the flow (Figure 6.b).
In section D, the axial fluctuations are still increased (up to 20%) in the region behind the strut, owing its high solidity (Figure 5.c). An indication of the considerable turbulence production of the strut is also given by the tangential fluctuating component, exceeding 30% (Figure 6.c). Consequently, the turbulence structure can be considered isotropic at mid-channel, while a remarkable anisotropy is present around the strut, with prevailing fluctuations in tangential direction. It is also possible to detect the growth of separation near the hub and the shroud and the disappearing of the tangential fluctuations generated by...
the inlet vanes.

Figures 5.d and 6.d show turbulence quantities in section E. In this case the axial velocity fluctuations, generated by the guide vanes and the struts, have been considerably damped, while the tangential fluctuations have practically disappeared. Moreover, the region with separated flow at the hub and the shroud has grown towards the center of the flow passage, and it is mainly associated with axial fluctuations.

CONCLUSIONS

The present work allowed to describe the main flow characteristics of a model of gas turbine exhaust diffuser. This evidenced the most interesting regions where a more detailed analysis may produce significant results. Some of the phenomena which need a more careful description are:
- inlet guide vanes produce wakes which propagate until the outlet of the diffuser, and play an important role in the flow separation at the hub;
- the interaction of strut's with guide vanes wakes is the main effect which produces the separation at the hub;
- separation at both the hub and the shroud is mainly influenced by axial velocity fluctuations;
- the strong damping effect on the tangential velocity fluctuations is probably produced by the shedding of vortexes generated by the strut;

More detailed measurements behind the strut may also help show the vortex shedding generated by the strut. An analysis of the turbulence scales can also be of interest in particular regions of the diffuser.

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