DESIGN AND TEST OF A NEW AXIAL COMPRESSOR FOR THE NUOVO PIGNONE HEAVY DUTY GAS TURBINES

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ABSTRACT

This axial compressor design was primarily focused to increase the power rating of the current Nuovo Pignone PGT10 Heavy-Duty gas turbine by 10%. In addition, the new 11-stage design favourably compares with the existing 17-stage compressor in terms of simplicity and cost. By scaling the flow path and blade geometry, the new aerodynamic design can be applied to gas turbines with different power ratings as well. The reduction in the stage number was achieved primarily through the meridional flow path redesign. The resulting higher blade peripheral speeds achieve larger stage pressure ratios without increasing the aerodynamic loadings. Wide chord blades keep the overall length unchanged thus assuring easy integration with other existing components.

The compressor performance map was extensively checked over the speed range required for two-shaft gas turbines. The prototype unit was installed on a special PGT10 gas turbine setup, that permitted the control of pressure ratio independently from the turbine matching requirements. The flow path instrumentation included strain-gages, dynamic pressure transducers and stator vane leading edge aerodynamic probes to determine individual stage characteristics. The general blading vibratory behavior was proved fully satisfactory. With minor adjustments to the variable stator settings the front stage aerodynamic matching was optimized and the design performance was achieved.

INTRODUCTION

The development of this axial compressor was undertaken to create a state-of-the-art design component for current and future Nuovo Pignone industrial gas turbine models. The first application was intended to provide the 10.5 MW PGT10 gas turbine with a higher flow compressor to assure a 10% increase in the rated power. At the same time, with the optimization of the blading flowpath and the use of advanced aero-design techniques, a substantial reduction in the stage and total blade number was sought, to simplify the general layout and reduce costs. The design objectives resulted in an 11-stage axial compressor, fig. 1, with wide chord, high strength blades that fits the existing gas turbine layout with only minor changes in the interfaces with other components and auxiliaries.

Fig. 1 11-Stage Compressor Assembly
The reference for the new design was the existing PGT10 compressor, and the target was to increase the mass flow by approximately 10% without substantially altering the interfaces with the rest of the existing equipment. The design of the present PGT10 compressor, fig. 2, was started in 1984 as a larger flow version of the 5 MW MS1002 gas turbine 15-stage compressor, with the addition of three transonic front stages and the removal of the last stage (see Benvenuti et al., 1988). To match the existing blading, the three new front stages had to be designed with a constant hub diameter and conically tapered outer casing. To maintain the integral rotor construction, the new stage rotor blades were made of titanium to limit stresses on the tangential root dovetails. To avoid stall during startup and at reduced operating speeds, five rows of variable stators, from the Inlet Guide Vanes through the stage 4 stator row were incorporated into the design. The design mass flow was set equal to 41.2 Kg/sec at a pressure ratio of 14.1.

For the new compressor, a 10% increase in the design flow was set as the target, to get a correspondingly higher gas turbine shaft power for the same firing temperature. With the turbine first stage nozzle area left unchanged, this increase requires a 10% higher pressure ratio. For the new design mass flow of 44.5 Kg/sec the pressure ratio results 15.4:1.

The geometric design constraints were imposed by interface requirements with the existing hardware; the principal consequences on the blading design were:

- maintain first stage inlet hub diameter unchanged to make it compatible with existing bearing number one sizes
- make the exit flowpath diameters compatible with the existing combustor-turbine transition piece
- assure compatibility of the overall compressor length with the existing baseplate and general accessory layout.

The requirement of keeping the overall length substantially unchanged with a smaller number of stages, made it possible to increase the airfoil chords, thus achieving a further reduction in the total blade number and assuring an increased mechanical strength.

Structurally, the solid rotor construction was changed into a bolted disk assembly for the first six stages, while the integral structure was maintained for the rotor rear end. The front-end disk construction with axial blade dovetails provided a much larger blade centrifugal force carrying capability, so that use of titanium was unnecessary. High strength 17-4-PH steel was therefore used on the initial 3 stages, while 13% chromium steel was left unchanged on the rest.

Variable stators were foreseen on the initial stages to prevent stall and surge during startup and at reduced operating speeds. On the prototype compressor, four rows of adjustable vanes were provided (Inlet Guide Vanes and stage 1 through 3 stators). However, after the initial tests the fourth adjustable row was fixed with no appreciable change in the measured startup dynamic stresses and in the flow control for power turndown. Therefore, the production units are being foreseen with only three continuously adjustable rows.

In addition to the increase in the blade chords, the space made available by the stage number reduction was used also to design a new inlet plenum and bellmouth assembly. The new design’s aerodynamic performance was experimentally checked on a 1:2 scale model. Very detailed flow surveys on the IGV inlet plane showed an average total pressure drop from the inlet flange lower than 0.5%, with swirl angles not exceeding 10 degrees. Measured total pressure drop and swirl angle circumferential distributions at IGV midspan can be seen in fig. 3, where losses and swirls appear significant behind the support struts and almost non existent elsewhere.
BLADING AERO-MECHANICAL DESIGN

Meridional Flow-Path Layout

The major difference between the new and the existing compressor design is in the meridional flow-path shape. On the present production compressor the initial four stages have a constant hub diameter, as a result of the original design requirement of matching the existing MS1002 compressor. In the new design, without such a constraint, the hub diameter could be increased right from the first stage, and kept growing up to a maximum compatible with the allowable hub/tip diameter ratio limit. This difference resulted in substantially higher blade peripheral speeds, with increased work and pressure ratio capability per stage without having to substantially increase the aerodynamic loading coefficients. The differences between the two meridional (flowpaths can be seen in fig. 4. The original flow-path design aim was to keep the outer diameter constant on all the stages. The final outer shape was instead tapered, starting from the fourth stage, to keep the maximum hub/tip diameter ratio within 0.92, beyond which secondary flow losses were expected to increase substantially. The blading exit annulus sizes were also checked for the requirement of matching the existing downstream components. The first stage inlet hub diameter was kept unchanged for compatibility with bearing number one size requirements. The outer diameter corresponds to a 7% increase in the annulus area with respect to the existing compressor, so that the blading inlet axial Mach number is slightly higher to achieve the 10% design flow uptake.

The required number of stages was evaluated with the aim of not substantially changing the aerodynamic loading parameters, primarily the stage work coefficients $\psi$ at the hub sections. With the new meridional contour the average hub peripheral speed $U$ is 1.25 times higher than in the existing compressor. Therefore the stages can provide a 1.56 times higher specific work $H$ without increasing the hub loadings. Theoretically, the number of required stages is therefore $17/1.56 \approx 10.9$, to be raised to 11 to account for a 4.5% total work increase required by the higher design pressure ratio. The stage number was actually set at 11, with the small difference being absorbed by a more even loading distribution throughout the entire compressor.

The larger blading exit annulus diameters made it necessary to shape the exit diffuser appropriately to match the existing downstream component sizes. A contour with a constant outer wall diameter and a curved, decreasing diameter inner wall was selected. The design was developed with the aid of a 3-D viscous flow analysis code, with a computational grid including the blading exit guide vanes for appropriately setting the diffuser inlet boundary conditions.

As for the existing compressor, two bleed ports are present along the flowpath. The stage four bleed is used for cooling and buffering low-pressure components, while the stage seven bleed is used for start-up and is closed during operation.

The overall bladed flowpath and diffuser length, fig 5, is shorter than that in the existing compressor. The remaining length accommodates the new, longer bellmouth, while the slight plenum rearward extension is still compatible with the existing baseplate design with minor modifications.
Aerodynamic Design

The reduced number of stages along with the above mentioned overall length requirements led to design of wide chord airfoils to fit the available space. Additional benefits resulting from this feature were a further reduction in the overall blade number and a higher mechanical strength. Furthermore, the associated lower airfoil aspect ratio, ranging from 1.1 for the first stage rotor blade to slightly below 1 for the last stage, can provide wider stage characteristic curves. The number of blades for each row was selected to achieve high enough solidities to keep the diffusion factors below 0.5 on all sections, as for the existing compressor. The inlet guide vane exit swirl angle was set to achieve a first rotor tip relative Mach number between 1.15 and 1.20. Use of multiple circular arc airfoils vs. the double circular arc type present on the PGT10 compressor was shown to be adequate by detailed viscous Navier-Stokes blade-to-blade analysis to limit shock losses in the supersonic region. The subsonic rear stage rotor blades and all the stator vanes were designed with standard NACA65 series airfoils.

A further reduction in the airfoil number was achieved at the rear end, where the existing final stage double-row stator and exit guide vane assembly was replaced by a single-row, high solidity and turning cascade. To keep the fluid turning close to 40° in this single-row exit guide vane, the rear end stage design flow coefficients were increased by 20% with respect to the existing compressor to reduce the rotor blade exit swirls. The potential penalty associated with the higher blading exit axial Mach number was offset by the diffuser design, with increased length and shape optimized with accurate viscous flow analysis calculations. The high-turning exit guide vane airfoil was designed with a special, analytically custom-tailored shape with surface Mach number distributions approaching those found on conventional airfoils with smaller flow turnings. The reduction in the overall stage number, the use of a single exit vane row as well as the low aspect ratio design led to the decrease in the total blade number to below 60% of those needed on the existing compressor.

Stationary and Dynamic Stress Analysis

Low aspect ratio blades, although providing benefits in terms of increased mechanical strength, entail strongly three-dimensional airfoil shapes with complex steady stress distributions and vibratory mode shapes. For the steady stress analysis, very detailed finite-element models were built and used to evaluate secondary stresses due to 3-D effects and to compensate them with appropriate airfoil section stacking. In the dynamic analysis, particular attention was paid to accurately predicting all the natural frequencies, and to interpret the high-order complex mode shapes typical of wide chord, thin airfoils. Among these high order modes, those with one or more nodal lines running almost parallel to the edges may be of concern due to the associated vibratory stresses. In fact, the thin airfoil “Stripes” bound by these lines and the edges can vibrate like independent airfoil portions with high dynamic stresses at the nodes. Nodal lines of the first “Stripe” mode at 2769 Hz for stage one rotor blades are shown in fig. 6. To check the Stripe mode stresses, the finite-element modal strain analysis was further utilized to assess the appropriate number of strain gage locations on the test compressor and to analyze the stress measurements as will be described below.

COMPRESSOR TEST SETUP AND INSTRUMENTATION

Testing of a full-scale compressor of this kind as an independent machine over the speed range typical of a two-shaft gas turbine would require a 15 to 20 MW power variable-speed driver. This could be achieved, for example, by using a production gas turbine, but would be very expensive due to the complexity of the setup and due to the costs associated with keeping an expensive commercial unit unavailable for sale during the entire test program. It was therefore decided to use a test methodology experienced almost at the same time for the new General Electric MS9001EC. 159.5 MW gas turbine compressor mapping test (Mezzadimi et al., 1995). According to this concept, the
The compressor itself was made part of a complete PGT10 gas turbine engine, so that it could be driven by its own internal turbine without requiring an external driver.

To explore the performance map extensively, it was necessary to provide a means for controlling the compressor back pressure well away from the turbine matching line. For this purpose, the turbine was modified and a special external flow control valve and piping layout was set up, fig. 7. To test the performance in the low pressure ratio range, special bleed ports with control valves were provided at the compressor discharge. They made it possible to decrease the pressure ratio down to the turbine no-load line. To achieve the high pressure ratio range, the standard turbine first stage nozzle throat area was decreased by modifying the airfoils in the trailing edge area. This throat area restriction was calibrated in order to achieve a pressure ratio midway between design and surge with the compressor discharge valves completely closed. To reach the surge conditions, the back pressure was further raised through the firing temperature increase produced by throttling the compressor suction. In this way the surge pressure levels could be lowered thus reducing risk of damage to the special flowpath sensors and instrumentation. Because of turbine first stage nozzle area reduction, to avoid surge during startup, substantial compressor discharge bleed was needed. To compensate for the large energy loss due to such a bleed, the regular starting motor had to be replaced with a 2.5 MW AC motor connected to a variable frequency grid.

Flowpath Aerodynamic Instrumentation

Correct stage aerodynamic matching is a key factor in the development of a high pressure ratio compressor of this kind, and must be ultimately checked in actual operation. Therefore, the prototype compressor flowpath was extensively instrumented to determine the individual stage characteristics. Interstage conditions were measured by means of multiple total pressure and temperature probes, fig. 8, installed in the leading edge of two stator vanes on each stage. Static pressures were measured along the casing at the leading and trailing edge planes of all the stator vanes. To measure average static pressure values, two groups of five equally spaced taps were provided at each plane. In turn, each group was located on the line connecting the edges of two adjacent vanes. The five taps were manifolded under the vane platforms, so that only one averaged pressure signal was sent to the acquisition system. A row-to-row data reduction program was used to calculate the individual stage characteristics in terms of flow, pressure and work coefficients and efficiency. To detect the approach or onset of non-stationary phenomena like rotating stalls and surge, casing dynamic pressure sensors were provided on a number of stages, including those adjacent to bleed locations.

Blade Strain Gages

Dynamic straingages were provided for all stator and rotor rows on two blades per row. Particular care was taken to evaluate straingage position and orientation on low aspect ratio airfoils for which high frequency, complex modes had to be checked. A general rule for straingage positioning was that of making each individual sensor capable of measuring dynamic stresses associated with more than one mode to keep the total number of straingages to an acceptable level. Thus it was not necessary to measure the full stress amplitudes, but only relative values to be corrected via suitable calibration factors. These factors were evaluated by means of the relative strain maps provided by the ANSYS dynamic analysis for each mode. Each map made it possible to calculate the ratio between the local and the maximum strain on the airfoil for any position on the airfoil surface. Therefore, it was not necessary to place each straingage at the point where the maximum strain for a particular mode was expected, but its...
position could be offset to measure relative stress values of multiple modes. This maximum offset was limited to avoid use of too large correction factors with a consequent loss in the measurement accuracy. To find the optimum strain gage location for multiple modes, an iterative process was employed, consisting in the computerized scanning of the ANSYS relative dynamic strain maps and in their interpolation over a great number of positions for all the modes of interest for each strain gage. The optimum strain gage location was then considered that for which all the "Transfer Factors", i.e., the ratios between maximum and local strains of each mode were closest to a minimum. As a general rule, all the rotor and stator blades were provided with straingages close to the airfoil roots to measure the low order pure flexural and torsion mode stresses. For the high order modes, the choice was made selectively on the basis of the anticipated importance of dynamic stresses associated with each mode. As an example, fig. 9 shows two straingages provided on the first stage rotor blade to measure strains for the first stripe mode shown in fig. 6 and for the third flexural mode.

COMPRESSOR TESTS

The compressor was extensively tested utilizing the particular layout schematically shown in fig. 7, with the actual setup shown in fig. 10. By suitably controlling the compressor discharge valve during acceleration to compensate for the turbine first stage nozzle area reduction, the regular startup matching line could be closely reproduced.

In the first test runs, all the four available variable stator rows were made continuously adjustable for startup, to increase the stall safety margins before some knowledge of the actual limits had been gained. After several successful tests, the fourth adjustable row was fixed in the design position, with no appreciable changes in the blade dynamic stress measurements during the following startups. Therefore, only three rows were finally left variable, according to the design intent.

The performance map was checked at variable corrected speeds from 85% to 110% of design speed. For each speed, the pressure ratio ranged from the turbine no-load line up to an upper limit set by the appearance of marked increases in the dynamic pressure transducer signals indicating the approach of stall. Actual surge points were checked at the very end of the test program to avoid possible premature damage to internal instrumentation, particularly straingages.

Blade Aeromechanical Behavior

The overall aeromechanical behavior in the startup speed range was in general excellent, with blade vibrations showing appreciable amplitudes only at resonances with the low inlet flow harmonics or with the upstream/downstream blade passing frequencies. The straingage location strategy proved very useful in detecting the dynamic stresses for all the modes involved, including high order, complex modes. Measured natural frequencies in operation were in general within 5% of predictions, with the exception of few, high order, complex modes on the very short rear-end stage blades and not affected by resonances in the operating range. Therefore, no unexpected resonances in the operating range were detected, so that no changes to passing frequencies or to blade natural frequencies were necessary. As an example, the experimental Campbell diagram of the stage 1 rotor blade is shown in fig. 11. The 90-100% speed range shown is related to normal compressor operation on a two-shaft gas turbine like the PGT10, equipped with variable power turbine inlet nozzles that are used to control the gas generator rotor speed. The vertical bars represent the dynamic stress amplitudes referred to the material high cycle fatigue limit, shown at the same scale on the diagram. Only stresses for which the Fourier straingage signal analysis showed amplitudes worth noting are represented. The dynamic stress levels did not exceed 25% of the high cycle fatigue endurance limit in the low speed range. The stresses
related to the two-stripe mode detected are low and this mode is out of resonance with any identified excitation source in the operating range. The stress amplitudes at resonances with low order harmonics due to inlet distortions and to the strut wakes are very low, and confirm the good quality of the flow at the blade inlet. Similar dynamic stress levels and relationships with upstream and downstream blade passing frequencies were observed on the subsequent stage blades.

Blade Aerodynamics and Performance

Interstage total pressure and temperature measurements made with the stator leading edge instrumentation, associated with 3-D flow analysis tools, were extensively utilized in correcting some stage flow mismatchings and to bring the overall performance to the design target. By an appropriate selection of the test point matrix, it was possible to determine the complete characteristic lines of each individual stage, from choke to near-stall conditions. The stage work coefficients and efficiencies were calculated between consecutive total pressure and temperature measuring stations. Therefore, each stage was considered consisting of the upstream stage stator (or IGV for the first stage) and the following rotor blade. The stage flow coefficient was in turn calculated at the rotor blade inlet utilizing the measured wall static pressure at the upstream stator exit.

The flow coefficients corresponding to design conditions were not located at the maximum efficiency points on some stages thus resulting in an overall performance penalty. Fortunately, these mismatchings appeared to be concentrated in the front-end stages provided with variable stators, that could be easily restaggered to shift the flow coefficients as needed. As an example, the second stage experimental work coefficient and efficiency curves for the stator design settings, fig 12, show a 2% efficiency loss due to mismatching. A 1% loss due to a similar mismatching was observed on both stage 3 and 4. The associated penalty in the overall efficiency was close to 0.5% in total.

A study of stator setting changes suitable to rematch the flow coefficients was performed analytically utilizing a 3-D viscous flow Navier-Stokes solver with multigrid capability for handling multiple rows simultaneously (Amone et al., 1993, Amone and Benvenuti, 1994). The number of rows depends on available workstation RAM and on the grid size. Typically, with a 250 MB RAM up to five rows can be handled with a number of mesh points sufficient to achieve good accuracy for performance predictions. Fig. 13 shows an example of the computational grid used for the first stage rotor airfoil on the hub surface. The inlet boundary conditions for the next row grid are automatically calculated by the code after completion of each iteration on the upstream row. In the meridional plane near the walls, the grid continues inside the blade tip clearance space in order to finely model the gap flows and losses, thus providing a detailed flowfield description right up to the endwalls.

Before using the computational tool to assess the effects of stator restagger on stage flow coefficients and performance, a number of runs were made to validate the code capabilities vs. detailed blade flow measurements. A typical analysis and test data match at the stage 2 stator leading edge for a test point very near to the design conditions is shown in fig. 14. The calculated back pressure was imposed to reproduce the test mass flow. With this condition, the calculated rotor 2 average exit total pressure and temperature profiles agree very well with measurements. Worth noting is the good reproduction of the radial total temperature profile, that shows the capability of faithfully modeling the end wall and tip clearance effects.
After the code check with the test data match, successive analyses with modified stage 1, 2 and 3 stator vane settings were performed to evaluate the effects on stage flow coefficients and efficiencies. Modified stator settings to bring each stage to operate at maximum efficiency without any need for airfoil design changes were finally identified. Final tests with the new nominal stator settings showed a recovery of the efficiency losses attributed to front-end stage flow coefficient mismatchings.

With the finally achieved performance, the application of this compressor to the PGT10 gas turbine permits a shaft power increase of 1 MW over the present rating, corresponding to the compressor design target.

CONCLUDING REMARKS

The approach followed in the development of a new, high pressure ratio compressor with a relatively small number of stages has proven successful in achieving the design targets with moderate tuning efforts. An important contribution to this result was given by the particular test methodology that made it possible to directly check the entire, full-scale compressor over the complete operating map in the actual gas turbine environment. The design strategy, aimed to achieve high stage pressure ratios without exceeding the existing PGT10 gas turbine compressor aerodynamic loadings, proved effective in terms of reaching comparable performance with a greatly reduced total blade number. The use of low aspect ratio, wide chord blade airfoils proved very effective in terms of dynamic response to aerodynamic excitations both in the startup and normal operating range.

REFERENCES


