Experimental Investigation of Axial and Radial Pressure Distributions in a Transonic Axial Compressor

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ABSTRACT
This paper deals with experimental results of an axial compressor of a small power plant gas turbine engine. Tests were carried out during the engine operation (along operating line of the compressor). Time averaged axial and radial pressure distributions in each individual stage were measured at different rotational speeds. Acceleration and deceleration phases of the engine were divided into reasonable time intervals of constant rotational speeds. Consequently, data logging was performed during steady operation of the engine. Measured parameters included pressure and temperature distributions and air mass flow rate. Test results were used to calculate axial distribution of load factor along the compressor meanline. Experimental results showed that the spanwise total pressure reduces from meanline region towards the hub and casing at each stage. No significant variations in load factor of each stage were observed during acceleration and deceleration phases of the engine. Meanline load factor distribution was increasing from compressor head towards its end within experimental rotational speed range.

Key Words: Axial Compressor, Gas Turbine Engine, Pressure Distribution, Load Factor

مطالعه آزمایشگاهی توزیع فشارهای محوری و شعاعی در یک کمپرسور محوری گذر صوت

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چکیده
این مقاله به نتایج آزمایش‌هایی بر روی کمپرسور یک توربین گاز کوچک مولد انرژی الکتریکی می‌پردازد. این آزمایش‌ها در هنگام کارکرد کل موتور، یا به عبارتی روی خط عملکرد کمپرسور صورت گرفته‌اند. مقادیر متوسط زمانی توزیع فشار محوری و محیطی در دورهای مختلف برای هرطبقه کمپرسور اندازه‌گیری شده‌اند. فازهای شتاب‌دهی و کاهش شتاب موتور به فاصله‌های زمانی معقول با سرعت ثابت موتور در هر کدام تقسیم شده است. لذا، اخذ اطلاعات آزمایشگاهی در حالت عملکرد پایای موتور صورت گرفته است. پارامترهای اندازه-گیری شده شامل شدت گذر جرمی هوا، توزیع دما و فشار می‌باشند. داده‌های آزمایش‌ها برای محاسبه توزیع ضریب بار در امتداد محور کمپرسور مورد استفاده قرار گرفته اند. نتایج آزمایش‌ها نشان می‌دهند که فشار کل در هریک از طبقات از ناحیه خط مرکزی به سمت پوسته و طوقه کاهش می‌یابد. از طرفی، تغییرات زیادی در ضریب بار هریک از طبقات در طی فازهای شتاب‌دهی و کاهش شتاب موتور دیده نمی‌شود. لیکن، توزیع محوری ضریب بار از ابتدا تا انتهای کمپرسور در محدوده سرعت چرخش موتور حین آزمایش‌ها افزایش یافته است.

واژه‌های کلیدی: کمپرسور محوری، موتور توربین گاز، توریع فشار، ضریب بار

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Introduction

The requirement of development of new turbomachines with higher efficiencies and wider safe operating domains dictates to increase real detailed information about performances of existing ones. This can be achieved by precise setting up of suitable instrumentation and measurement systems with careful observation and monitoring operational conditions inside the machine. Up to now, many efforts have been made by various researchers to find suitable measuring and analysis methods [1-2].

The major component of a gas turbine engine can be referred to its compressor. Flow structure inside the compressor is among the most complex flows in the fluid dynamics. It is always three-dimensional, viscous and unsteady in nature. Different flow types in terms of subsonic, transonic and supersonic regimes may encounter at different regions, simultaneously. Even two-dimensional cascade flows can be complex due to the existence of shock wave structure at leading edge region or through the flow passage and their interaction with the boundary layer. Consequently, flow may separate from the surface and as a result, a reversed flow will occur, which makes the flow structure more complex. All these factors cause compressors design and testing procedures to become complicated.

Compressors of modern gas turbine engines require high pressure ratios and efficiencies associated with safe surge margins. In addition, weight and size constraints on air vehicle engines causes loading and flow speed on the blades to be considered as high as possible by designers in comparison to those of ground engines. Testing the current compressors can provide a large domain of data which can be utilized in the future modifications and designing of high performance engines.

In the present research work, a compressor of a gas turbine engine was tested under acceleration and deceleration phases of an engine. A large number of pressure tappings and thermocouples were mounted in suitable places in the engine. Measured data were used for post-processing purposes and calculation of the governing parameters encountered in compressor studies.

Experimental Set Up

A three-stage axial compressor, which belongs to a small gas turbine engine, was tested while the engine was running during acceleration and deceleration phases. Engine rotational speed was controlled via a throttling system installed on the fuel feeding line.

Tests were carried out during the operation of the whole engine, i.e., along operating line of the compressor. Time averaged axial and radial pressure distributions in each individual stage were measured at different rotational speeds. Acceleration and deceleration phases of the engine were divided into reasonable time intervals of constant rotational speeds. Consequently, data logging have been performed always during steady operation of the engine.

Sensoring was performed at stations located between each two subsequent blade rows together with entrance and exit sections of the compressor (row by row measurements). Figure **1** introduces these stations.

A suitable number of pressure tappings and thermocouple probes were distributed in the above axial stations. They were also suitably distributed every 120˚ at each station, circumferentially.

Two Pitot-static tubes were also mounted in bell-mouth air intake for measurement of mass flow rate entered into the compressor. Static pressure tappings were mounted in the compressor casing.

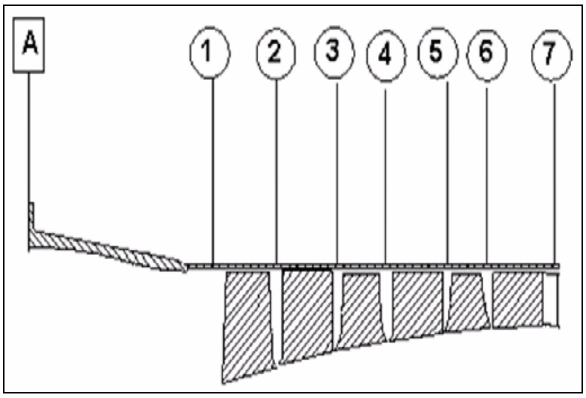

Fig. (1): Schematic drawing of compressor axial cross-section with measuring stations.

To measure the total pressure distribution in radial direction, stator blades, as shown in Fig. **2**, were drilled normal to their leading edges in some selected spanwise positions. Conical drilling at the leading edge caused the total pressure to become insensitive to the flow angle within a certain range. Generally, it depends mainly on the cone angle.

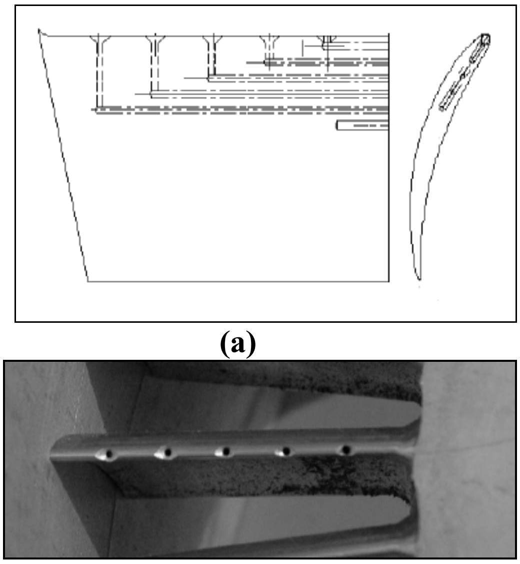

Fig. (2): Stator blade prepared for measurement of radial distribution of total pressure: a) Schematic drowns, b) real picture.

Performing wind tunnel testing, specially sat up for the present studies, showed no significant variations in the total pressures whilst the incidence angle was changing from -10˚ to +10˚ for a conical angle of 45˚. The relevant results are shown in Fig. **3**. Experiences and design features have shown that incidence angles of stator blades do not deviate from the above range and their magnitude usually do not exceed from about 5˚ or 6˚. As a result, mounting the pressure tappings along the stator leading edge, the stagnation pressures can be measured to a reasonable accuracy. This method of measuring the radial distribution of total pressure can be addressed for example to Gaillard [3] and Pettonin [4].

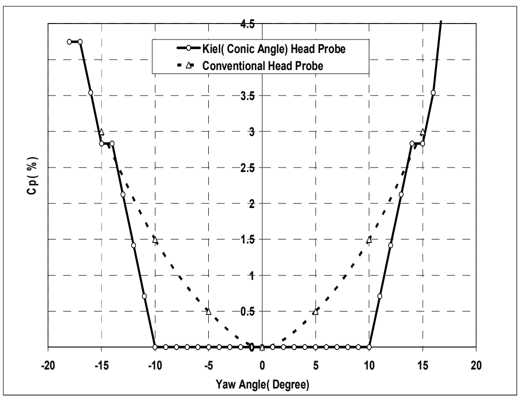

Fig. (3): Comparison of pressure results measured by Kiel (conical) and conventional head probes.

Low thicknesses of the third stator blades, made it impassible to mount any pressure tapping along its leading edges. Therefore, a decision was made to use rear frame struts for measuring the compressor exit stagnation pressure (see also references [3] and [5]). These struts were located after diffuser, which were connected to the compressor exit section. Because of low divergence angle of the diffuser, flow would remain attached to its wall and no separation of flow would occur. As a result, the diffuser losses could be ignored to a high degree of accuracy [6]. There were six struts evenly spaced circumferentially. One pressure tapping was considered for each strut along the mean stream-surface (i.e., along the mean line in the meridional plane) .

It is assumed in the present investigation that there are no significant changes between the absolute stagnation pressures at the exit section of the rotor blades and those at the adjacent stator blade entrance section nearly along the same stream-surface [7].

The instrumentation inside the present compressor performed in such a way to avoid any blockages due to the existence of foreign objects like any measuring probes. This could guarantee safe operation of the whole engine and high accuracy of the measured parameters.

Each pressure tapping was connected to a piezoelectric pressure transducer with a maximum pressure of 6 bar and a sampling rate of 10 KHz. All pressure transducers were mounted in a box specially designed and manufactured for the current experiments. The maximum length of each connection tube between the probe and the pressure transducer was about 4 meters. This low tube length resulted in a negligible time lag in logging the pressure pulses. As has already been mentioned, data logging have been performed always during steady operation of the engine. Acceleration and deceleration phases of the engine were performed step-by-step, with constant rotational speed for each step. Selected sampling rate of 1 KHz and waiting time of about 20 seconds, at each rotational speed for the data logging process, guaranteed the right values of measured data during the steady operation. Consequently, based on pick up speed of A/D cards and pressure transducers it was possible to monitor a large number of data in every second.

Each transducer was connected to a filtration circuit inside the sensors box for alleviation of all undesirable noises arisen from the surrounding. As a result, in addition to the steady state conditions, it was possible to conduct the experiments for both the acceleration and deceleration phases of the gas turbine engine. In other words, it was possible to record necessary data at different rotational speeds.

An industrial computer, equipped with a suitable number of analogue to digital (A-D) cards, was prepared to conduct the current set of experiments. The maximum sampling rate of A-D cards was 100 KHz.

A software was developed for data monitoring purposes and computing of some of the main parameters, simultaneously. Basic parameters governed in the compressor studies were also calculated through running a post-processing software specially developed for the present research work.

Test Results

Measured parameters provided to determine the most dominant thermodynamics and gas dynamic parameters governed in axial compressors. This was performed through a post-processing software which was specially developed for the present studies. However, for the sake of brevity, only the pressure and load factor results have been presented and discussed in this paper. All the parameters have been normalized with respect to their values at the design point conditions.

Figures **4** and **5** show the radial distributions of total pressure along the leading edges of the stator blades of the first and second stages, respectively. These results have been presented for some selected rotational speeds. As can be observed in the above figures, increasing the rotational speed has caused the total pressures to increase. The maximum total pressures have occurred around the midspan of each stator. Moving towards each blade ends in the radial direction, total pressure decreases, gradually. Less total pressures around each blade tip are due to end wall effects and vortices generated from the rotor blades tips. Tip vortex is physically due to the tip clearance and the pressure difference between suction and pressure sides of the rotor blade. Axial compressor designers, in initial steps of design process, usually distribute the spanwise total pressure uniformly. However, as already mentioned, less total pressures close to the casing and the hub are unavoidable.

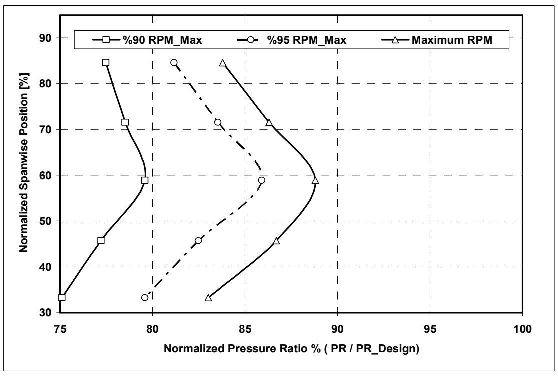

Fig. (4): Radial distributions of total pressure ratio along the leading edge of stator blade row of the first stage at different rotational speeds.

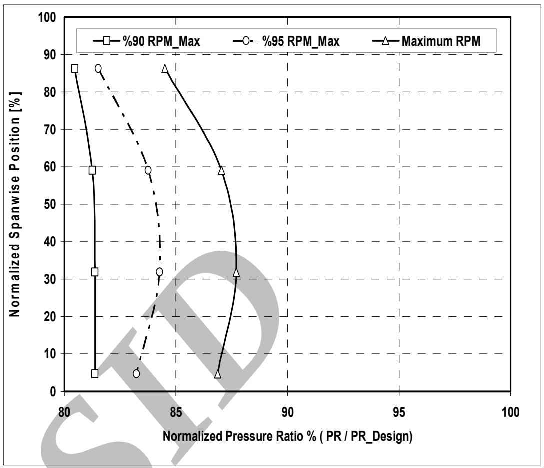

Fig. (5): Radial distributions of total pressure ratio along the leading edge of stator blade row of the second stage at different rotational speeds.

Variations of stage average total pressure ratio with rotational speed are shown in Fig's. **6** and **7** for acceleration and deceleration phases, respectively. It can be detected from these figures that the first stage is exposed to higher pressure ratios than the other stages. The pressure ratio decreases, by moving towards the compressor exit section. The same trend of pressure ratio variations can be observed in most axial compressors.

In other words, the first stage has been exposed to higher pressure ratios than the other stages. A maximum pressure ratio of about 1.6 has been detected for the first stage of the compressor of interest, which had occurred at the engine maximum rotational speed.

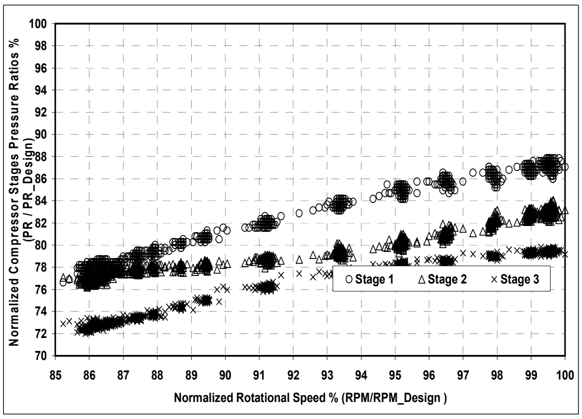

Fig. (6): Stage average pressure ratio at different rotational speeds for the engine acceleration phase.

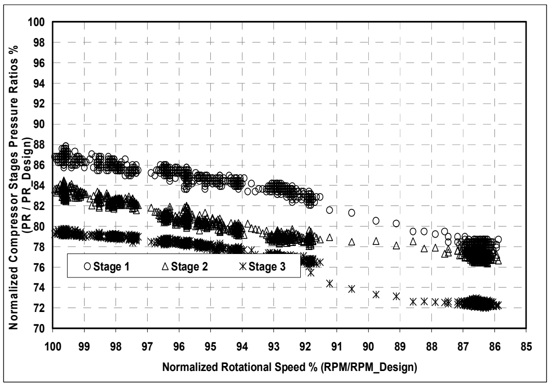

Fig. (7): Stage average pressure ratio at different rotational speeds for the engine deceleration phase

Figures **8** and **9** represent stage loading at different rotational speeds for the acceleration and deceleration phases, respectively. They show that the third stage has the maximum load factor in comparison to the first and second stages within the whole rotational speed range.

The load factor has been defined as $\Psi=\Delta h_0/\rho U^2$, with Δh_0 as the enthalpy rise in the rotor blade and U as the rotor blade tip speed. All the load factor values have been calculated less than about 0.45, which is satisfactory.

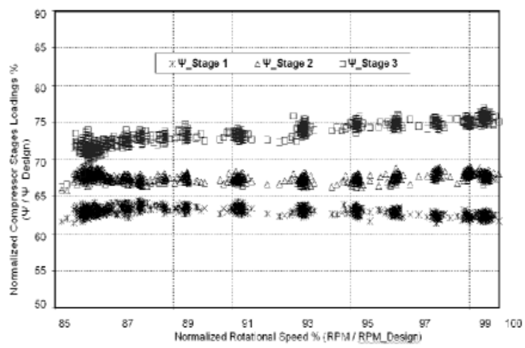

Fig. (8): Mean-line load factor of each stage at different rotational speeds for the engine acceleration phase

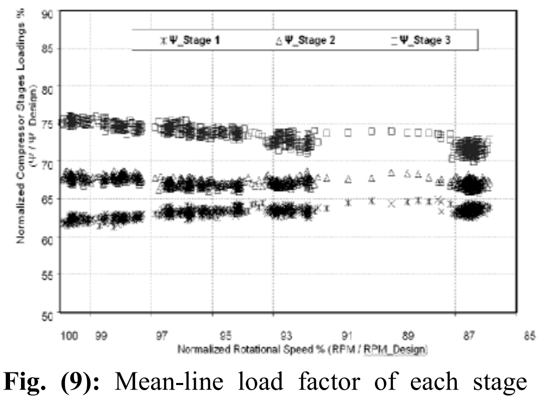

Fig. (9): Mean-line load factor of each stage at different rotational speeds for the engine deceleration phase.

Conclusion

Extensive tests performed on an available transonic three-stage axial compressor on its operating line provided a large domain of data which can be utilized in necessary future modifications and designing of higher performance new engines. The main conclusions withdrawn from the current experimental research work can be categorized as follows:

1- Mounting the pressure tappings along the leading edges of stator blades with a conical divergence head of 45° can be used to measure the spanwise total pressure distribution to a reasonable accuracy. So the total pressure distribution of compressor of a small power plant gas turbine engine from hub to tip can be measurable without blockage to flow angle of ±10 degrees,

2- First stage of the compressor is exposed to highest pressure ratio, which is in accordance with the most available axial compressors. The stage pressure ratio is decreasing by moving towards the compressor exit section. As these changes between first to third stages of this compressor in maximum speed reaches to 11% of designed pressure ratio,

3- Increasing the rotational speed causes the pressure ratio to increase along each stream-surface,

4- The maximum pressure ratio of all stages is distributed nearly around the mean stream-surface. In addition a boundary layer close to tip and hub of cascade, results in more losses close to the wall and despite of applying radial equivalence designing method in designing of this compressor, there is parabolic pressure distribution in flow passage,

5- No great variations in load factors are observed during acceleration and deceleration phases of engine. This result admits the designing of the compressor on the basis of radial equivalence method and

6- Mean line load factor are increasing by moving from compressor head toward its end within the test rotational speed range.

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