EXPERIMENTAL INVESTIGATION OF STALL PRECURSOR-SUPPRESSED (SPS) CASING TREATMENT IN A TWO-STAGE AXIAL COMPRESSOR

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ABSTRACT
In this paper, the effects of the stall precursor-suppressed (SPS) casing treatment on stall margin and efficiency are investigated experimentally in a two-stage axial compressor. The SPS casing treatment can improve the stall margin of the compressor by 3%-11% without significant efficiency loss. In addition, the mechanism of stall margin enhancement with such casing treatment is also revealed by observing the evolution of the precursor wave.

INTRODUCTION
In modern aeroengine design, high thrust and efficiency are usually pursued. At present, the compression ratio of a single stage fan/compressor can reach 2.2-2.4, and theoretically, the turbo-fan aeroengine of 15-20 thrust-weight ratio can be designed if the average stage pressure ratio reaches 1.85 [1]. However, since the birthday of F119, it is still impossible for manufacturers to carry out a product with the thrust-weight ratio of 20. Obviously, fan/compressor must have higher blade loading under this situation. However, it is extremely difficult to meet the requirement of sufficient stall margin in a multi-stage compression system. Based on this concern, the flow stability problem of multi-stage compressor is the key issue for improving the holistic performance of modern turbo-machinery aeroengine.

However, in the stall-onset prediction of multi-stage compressor, there are many difficulties in applying theoretical models and numerical simulations. Most of these difficulties are from some complex coupling of aerodynamic problems and structure problems. And the active control method [2-5] which is the most popular method in compressor stabilization nearly can not show any effect in multi-stage system. In a multi-stage compressor, it is really hard to detect and recognize the information of stall inception or stall onset.

Casing treatment is an effective way to improve the stall margin. Since the famous observation about casing treatment was guided by Koch [6] in 1970, but most of casing treatment design has been based on trial and error experiences so far. Obviously a better design depends on the understanding of the mechanism. The existing investigations have pointed out another possibility that the mechanism will base on unsteady flow. With the development of numerical simulation technology, the design of modern casing treatment can be realized through numerical simulations, although it is really hard in doing with multi-stage compressor systems. In 2012, Gourdain [7] used unsteady RANS numerical simulation investigating the effects of tip clearance and casing treatment on compressor stability. The results showed that casing treatment could obviously improve the stability range of compressor and could even make up the stall margin loss caused by the increasing of tip clearance. But this work did give out the mechanism of rotating stall and casing treatment. Well, the application of experimental investigation method can not be easily transferred to multi-stage system from single-stage system. In 2003, Akhlaghi [8] carried out an experimental investigation of casing treatment on a three-stage compressor rig. This work used a casing treatment with annular back-chamber and guide vanes to research the effects of casing treatment at different axial positions relating to the rotor tip. Besides, when applying stabilization technology into multi-stage compressor, not only the stall margin should be taken into concern, but also the matching of each stage should be considered. Several years ago, Kroeckel [9] applied a traditional casing treatment into a 2-stage high-pressure
compressor. They found the casing treatment can enhance the stall margin obviously but the performance and flow field were also changed. And Kern [10] used the method of air injection at the blade tip region to realize the purpose of stabilization on a 8-stage high-pressure compressor. Although in that research, the 12 injectors could use 5% mass flow to gain 8.5% stall margin enhancement, but the matching of each stage was also be changed totally. Therefore, the existing technics can not be easily applied into multi-stage compression system.

Based on the small disturbance theory and vortex-pressure wave interaction suggestion, the relevant theoretical investigations have been carried out [11-15]. A three-dimensional axial-compressor stability model also has been developed, which is adopted to predict the onset point of rotating stall of a multi-rows compressor. It is possible to consider a complicated boundary condition in the stability model due to its three-dimensionality. The equivalent surface source method was introduced into the model to take account of the soft wall boundary condition. The advantage of this method is that it computes the scattering wave only through the eigen-value of solid wall instead of that of soft wall, so the related effect of casing treatments can be included in this way. So, the model can also help to design the geometric parameters of casing treatments. According to these results, the SPS casing treatment consisting of back-chamber and perforated plate is suggested. Compared with the conventional casing treatment aimed at the improvement of blade tip flow structure, the objective of the SPS casing treatment is to affect the evolution of the stall precursors in order to obtain stall margin improvement. The perforated ratio of the SPS casing treatment is only 4-10%, far smaller than in traditional casing treatments (with over 50% open area ratio). The relevant experiments [16-22] were conducted based on the SPS casing treatment, which is installed above the tip of the rotor blade. The results showed that the SPS casing treatment designed according to the results of theoretical research using the stability model of rotating stall can result in the stall margin improvement without efficiency loss. In the present investigation, the experiments of the SPS casing treatment on the two-stage low-speed fan TA66 are going to explore its stabilization ability on the multi-stage compression system.

**Table 1 Structure and performance parameters of TA66**

<table>
<thead>
<tr>
<th>Structure Parameters</th>
<th>IGV</th>
<th>1R</th>
<th>1S</th>
<th>2R</th>
<th>2S</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade Number</td>
<td>38</td>
<td>47</td>
<td>45</td>
<td>47</td>
<td>45</td>
</tr>
<tr>
<td>Established Angle</td>
<td>0°</td>
<td>60°</td>
<td>10°</td>
<td>60°</td>
<td>10°</td>
</tr>
<tr>
<td>External Diameter/mm</td>
<td>600</td>
<td>600</td>
<td>600</td>
<td>600</td>
<td>600</td>
</tr>
<tr>
<td>Span-chord Ratio</td>
<td>1.69</td>
<td>1.69</td>
<td>1.69</td>
<td>1.69</td>
<td>1.69</td>
</tr>
<tr>
<td>Hub-tip Ratio</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
</tr>
</tbody>
</table>

**Performance Parameters**

<table>
<thead>
<tr>
<th>m_d (kg/s)</th>
<th>n_d</th>
<th>3000rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.5</td>
<td>0.276</td>
<td>1.176</td>
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</table>

<table>
<thead>
<tr>
<th>Power (kW)</th>
<th>η</th>
</tr>
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<tbody>
<tr>
<td>16</td>
<td>—</td>
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</table>

<table>
<thead>
<tr>
<th>x_d</th>
<th>p_s - p_t</th>
<th>4675Pa</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.056</td>
<td></td>
<td></td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Rotor Tip Clearance</th>
<th>0.6-0.8mm</th>
<th>U_0</th>
<th>94.20m/s</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Rotor-Stator Axial Clearance</th>
<th>8-20mm</th>
</tr>
</thead>
</table>

Fig. 2 shows the main steady pressure measurement probes, as the arrows indicating, the inlet/outlet static pressure can be measured averagely by 4 pressure sensors of each cross profile, and the inlet/outlet total pressure can be measured by two pressure ranks. All these total and static pressure sensors used here are 142PC01D Honeywell pressure sensors. These series sensors can provide an output voltage proportional to applied pressure. They operate from a single, positive supply voltage ranging from 7Vdc to 16Vdc. Signal conditioning results in directly usable outputs. Temperature compensation provides predictable performance over specified temperature ranges. And the rated pressure range is 1 PSI between two pressure ports (one port is atmosphere and another one is measured total/static pressure, so in the present work, pressures are measured with respect to an atmospheric pressure reference).
There are also 8 high frequency-response dynamic pressure sensors inlaid in the casing about half a chord length upstream of the rotor inlet of 4 cross profiles equally spaced around the circumference, as shown in Fig. 3. The model number of these 8 high-frequency response pressure sensors is XT-190M-1D, the rated pressure is 1 PSI in differential operational mode, and the sensitivity is about 5000.000mV/PSI.

![Fig. 3 Dynamic pressure measurement sensors](image)

The scheme of SPS casing treatment is shown in Fig. 4, it is installed in the upstream of rotor blade with about one-half overlapped area on the blade tip region. The perforated ratio of this SPS casing treatment is about 6%, and there is an annular back-chamber in 50 mm height, 100 mm length. This figure also briefly shows the mechanism of SPS casing treatment: the tip-region flow comes in and out the back-chamber generating some vortexes shedding which can interact with the perturbation pressure waves in the main flow, and this interaction can be recognized as a soft boundary condition of this system.

![Fig. 4 SPS casing treatment schematic and installing status](image)

**EXPERIMENT RESULTS**

The present work contains a series of experiments at different working speeds to demonstrate the compressor performance and stall margin enhancement of SPS casing treatment. In addition, time relevant analysis of dynamic stall signals is also presented to indicate the comprehensive characteristic of this test compressor.

**Relevant Parameters and Equations**

For convenience here, this paper uses CT to represent using the SPS casing treatment, SC to represent using the ordinary solid wall casing, and SM to stand for the stall margin, which is defined by:

$$SM = \left( \frac{\Delta p_d/\phi}{\Delta p_s/\phi} - 1 \right) \times 100\%.$$  \hspace{1cm} (1)

where the subscripts $d$ and $s$ respectively represent the parameters at design point and stall point.

The mass flow rate $G$ is measured at point 0-0 as shown in Fig. 3, and the relevant definition is

$$G = kA_0 \frac{P_0^*}{\sqrt{T_0^*}} q(\lambda_0).$$  \hspace{1cm} (2)

where $\varphi = 0.992$, $A_0$ is inlet area, $k$ is ratio of specific heat, $p_0^*$ is inlet total pressure, $T_0^*$ is inlet total temperature, $\pi(\lambda_0)$ a compressibility function and $\pi(\lambda_0) = \frac{P_0}{P_{atm}}$; the look-up table method is used to derive $q(\lambda_0)$. $P_0$ is inlet static pressure, and $P_{atm}$ is the atmospheric pressure.

The total-to-static pressure rise coefficient $\psi$ is usually a relevant parameter for rotating stall in low-speed machines:

$$\psi = \frac{p_2^* - p_1^*}{0.5 \rho U_{m}^2}.$$  \hspace{1cm} (3)

where $p_2$ is the static pressure at stator exit, $p_1^*$ is total pressure at rotor inlet and $U_{m}$ is tangential speed at mid-span.

The flow coefficient $\phi$ is

$$\phi = \frac{V_x}{U_{m}} = \frac{q_v}{U_{m}A_0}. \hspace{1cm} \text{ (4)}$$

where $V_x$ is inlet axial velocity, $q_v$ is volume flow and $\rho_0$ is inlet density.

Efficiency $\eta$ is expressed as

$$\eta = \frac{q_v \left( p_2^* - p_1^* \right)}{1000 \dot{P}_e}. \hspace{1cm} \text{ (6)}$$

where $p_2^*$ is total pressure at stator exit, $\dot{P}_e$ is the electrical power and $\eta_e$ stands for the motor power coefficient.

The $\dot{P}_e$ is measured by a torque meter whose measurement accuracy is about 0.001V and sensitivity is 5.002V/100N · m. Moreover, the high measurement accuracy of $\dot{P}_e$ enables the error range of efficiency to be small enough. Therefore, the efficiency results can show the differences between the different cases clearly.
Experimental Repeatability

**Fig. 5 Repeatability results of compressor performance**

In order to expound the error range of the major parameters, some repetitive experiments have been conducted and the results are shown in Fig. 5. From this repetitive experiments result, it is confirmed that the system error of total-to-static pressure rise coefficient and efficiency are both too small to excessively affect the comparisons between different cases.

**Compressor Performance**

The design speed of this two-stage compressor is 3000rpm, and 5 working speeds (60%, 70%, 80%, 90%, 100% design speed) were picked to demonstrate the performance of this compressor. The specific experimental results without using SPS casing treatment are shown in Fig. 6 and Fig. 7. Fig. 6 shows the pressure-rise curves of different working speeds, the design point is also shown in this figure. And this two-stage compressor has high stall margins under different working speeds (SM_{100%}=34.2%).

**Stall Margin Enhancement of SPS Casing treatment**

In order to indicate the effectiveness of SPS casing treatment in stall margin enhancement on this test rig, experiments with SPS casing treatments installed on each stage and both two stages were carried out. Because of the mechanism of SPS casing treatment, there is no big influence caused by the installing location of casing treatment. The strategy of SPS casing treatments installed on both two stages has the biggest perforated ratio and the best stabilization ability. So Fig. 8 and Fig. 9 are the results under this circumstance. From the pressure-rise curves and the efficiency curves, it is obvious that the SPS casing treatment can enhance the stall margin (SM_{CT100%}=46.1%) without induce additional efficiency loss.
Fig. 8 Pressure-rise curves of different working speed with and without SPS casing treatment

Fig. 9 Efficiency curves of different working speed with and without SPS casing treatment

Analysis of Dynamic Stall & Pre-stall Signals

For a multi-stage compressor, the rotating stall may happen in any one stage or stages. This test rig has two stages, and from the steady performance result, it can not be found that in which stage the rotating stall originates. So time relevant analysis were induced into this present investigation. Fig. 10 shows the measurement system of dynamic static pressure, there are 8 sensors inlaid in the casing near the rotor tip or the leading edge of stator.

Fig. 10 Measurement system of dynamic static pressure

Fig. 11 shows the results of the case in which the sensors were placed near the rotor tip of two stages. The normalized no-dimensional static pressure signals of 16 channels are shown in a same time axis. Some oblique lines in this figure show the revolution of rotating stall. It is clear that the stall onset can be firstly found in the 1st stage, and the stall signals detected in the 2nd stage were from the 1st stage.

Fig. 11 Dynamic static pressure signals of stall evolution

If the rotating stall starts in the first stage, the stall energy could be stronger in first stage than that in second stage. Therefore, this paper used FFT and time-relevant PSD methods to further analysis the stall signals such as shown in Fig. 11. Fig. 12 shows the frequency characteristics of stall evolution in two stages, and in this figure, the stall frequency is 17.2Hz whose amplitude is much higher in 1st stage than that in 2nd stage. Fig. 13 shows the time-relevant PSD results of stall evolution in two stages. The pressure perturbation energy in first stage is much stronger than that in second stage. All these comparisons mean that the rotating stall of this two-stage compressor is firstly generated in the 1st stage, which means that if considering the weight cost of using SPS casing treatment, the stabilization strategy would be setting only one casing treatment at the first stage.

Fig. 12 Frequency characteristics of stall evolution in First-stage and Second-stage
CONCLUSIONS

By illustrating the steady performance results, the stabilization ability of SPS casing treatment and the analysis of dynamic stall/pre-stall signals, some conclusions can be summarized briefly as follow:

1. This two-stage compressor has high stall margins under different working speeds and high overall efficiency at the design operation points;

2. From the pressure-rise curves and the efficiency curves, it is obvious that the SPS casing treatment can enhance the stall margin ($\Delta SM = 11.9\%$) without induce much additional efficiency loss;

3. The rotating stall of this two-stage compressor is firstly generated in the 1st stage.

NOMENCLATURE

- $G$: mass flow, kg/s
- $\psi$: total-to-static pressure rise coefficient
- $\phi$: flow coefficient
- $\eta$: efficiency
- $A_0$: inlet area, m$^2$
- $q\left(\lambda_0\right)$: mass flow function
- $\pi\left(\lambda_0\right)$: compressibility function
- $\pi_d^*$: total pressure ratio at design point
- $P_1$: static pressure at stator exit, Pa
- $P_0^*$: total pressure at rotor inlet, Pa
- $U_m$: tangential speed at mid-span, m/s
- $\rho_0$: inlet density, kg/m$^3$
- $V_c$: inlet axial velocity, m/s
- $\dot{q}_v$: volume flow, m$^3$/s
- $\eta_E$: motor power coefficient
- $\bar{P}_e$: electrical power, W
- $\dot{P}_e$: actual output power of motor, W
- $\Delta$: efficiency error range
- $P_i$: inlet static pressure
- $P_t$: total pressure
- $\bar{P}_t$: time-averaged total pressure
- $\rho$: density
- $p$: pressure
- $T$: temperature
- $SM$: stall margin
- $CT$: with SPS casing treatment
- $SC$: without SPS casing treatment (solid casing)
- $SPS$: stall precursor-suppressed

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