EFFECT OF ENDWALL CONTOURING ON THE PERFORMANCE OF A THREE-STAGE HP TURBINE

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ABSTRACT

This paper describes experimental investigations of a three-stage high pressure research turbine at various operating conditions. The experimental investigation is carried out in a three-stage turbine facility at the Turbomachinery Performance and Flow Research Laboratory (TPFL) at Texas A&M University. Its rotor includes non-axisymmetric endwall contouring on the first and second rotor row. Experimental data were obtained using interstage aerodynamic measurements at three measurement stations, namely, downstream of the first rotor row, the second stator row and the second rotor row. Measurements were conducted using five-hole probes traversed in both circumferential and radial directions to create a measurement window. Performance measurements were carried out within a rotational speed range of 1800 to 3000 RPM. Interstage measurements were obtained at 3000 rpm. Moreover, the impact of the purge flow injection on the aerodynamic and performance behaviour of the turbine for five coolant-to-mainstream mass flow ratios of 0.0%, 0.5%, 1.0%, 1.5% and 2.0% were investigated. Interstage results exhibit the evidence of the mitigation of secondary flow intensity due to endwall contouring.

INTRODUCTION

The secondary flow and its impact on efficiency and performance of turbine components are described extensively by (Lakshminarayana, 1995) and (Schobeiri, 2012). As explained in detail by (Schobeiri, 2012), turbomachinery losses can be divided into endwall or secondary losses, profile losses and leakage losses. Profile loss is considered loss effects due to boundary layer development on suction and pressure surfaces of the blade. Analyses for profile loss often assume a primarily two-dimensional flow, and losses at the trailing edge are often included into the profile loss category. Endwall losses are commonly referred to as secondary flow losses and arise due to secondary flows generated as the annulus boundary layers in the blade passage. Tip leakage losses arise from leakage type flows across the blade tip for a rotor or stator hub, and depend on whether the blades utilize shrouds or do not. Leakage flows as stated by (Denton, 1993) interact strongly with the secondary flow patterns in the passage, thus it is generally considered that there is large degree interplay between the various types of losses encountered in turbomachinery.

Focusing on the secondary flow loss mechanisms, the fluid particles within the endwall boundary layers are exposed to a pitchwise pressure gradient in the blade channel. The particles move from the pressure side to the suction side and generate a system of vortices. These vortices induce drag forces that are the cause of the secondary flow losses. In addition, their interaction with the main flow causes angle deviation inside and outside the blade channel, resulting in additional losses due to angle deviation (Schobeiri & Lu, 2013).

In recent years, numerous papers have been published that deal with the effect of endwall contouring and leading edge filleting. With a few exceptions of rotating rig investigations that deal with the endwall contouring of LP turbines, most of the published studies are either numerically or experimentally performed in turbine cascades with steady inlet flow conditions. Numerical and experimental studies by (Hartland & Gregory-Smith, 2002), (Ingram, et al., 2002), (Eymann, et al., 2002) and (Sauer, et al., 2001) show a reduction of total pressure losses by as much as 50%.

In contrast to the tremendous multitude of the cascade endwall papers, from which only a few have been discussed above, there are only a few investigations of the impact on endwall contouring in rotating turbines. Brennan et al. (Brennan, et al., 2003) redesigned the HP turbine of the Rolls-Royce Trent 500 engine with the application of non-axisymmetric endwalls. The profiled endwall shape was determined by six control stations which were fixed at specified axial distances along the mean camber line of the airfoil. The addition of profiling to the endwalls of the HP Turbine is predicted to reduce secondary loss by 0.24% of the...
NGV and by 0.16% for the Rotor. The total improvement in stage efficiency for the HP Turbine is therefore +0.4%.

(Harvey, et al., 2002) redesigned the IP-turbine stage by applying non-axisymmetric endwalls to both the vane and blade passages. They reported an improvement in the stage efficiency of $0.9 \pm 0.4\%$ at the design point. Germain et al. (Germain, et al., 2010) studied the improvement of efficiency of a one-and-half stage high work axial flow turbine by non-axisymmetric endwall contouring. The endwalls have been designed using automatic numerical optimization by means of a Sequential Quadratic Programming (SQP) algorithm. Both hub and tip endwalls of the first stator as well as the hub endwall of the rotor were modified. The experimental results confirm the improvement of turbine efficiency, showing a total-to-total stage efficiency benefit of $1\%\pm0.4\%$, while the improvement is underestimated by CFD. (Snedden, et al., 2009) and (Snedden, et al., 2010) utilized five-hole probe measurements in a 1.5 stage low speed model turbine in conjunction with computational fluid dynamics to gain a more detailed understanding of the influence of a generic endwall design. Results indicated a 0.4% improvement in total-to-total rotor and stage efficiency as a result of the application of the generic non-axisymmetric endwall contouring. However, at higher loading the rotor efficiency was reduced by 0.5%.

In previously published paper (Rezasoltani, et al., 2014) effect of endwall contouring on film cooling effectiveness was investigated. The objective of the present study is to experimentally investigate the effect of endwall contouring on aerodynamic behaviour and performance of a rotating HP turbine. Results of comprehensive performance measurements and interstage traversing at different rotational speeds and mass flow rates are presented.

**BLADE ROW INTERACTION**

Endwall secondary flow losses and vortices play a significant role in reduction of stage efficiency in high pressure (HP) turbines. As explained in introduction, endwall contouring is one of the ways to reduce endwall secondary flows. Recent studies (Hodson, et al., 1995), (Chaluvadi, et al., 2001) and (Pullan, 2004) depicted how flow modified at the stage exit by upstream secondary vortices. (Hodson, et al., 1995) investigated the interactions of incoming wakes and secondary flow vortices in a single stage axial turbine. Results showed that stator secondary flows have small effect on rotor secondary flows. Three dimensional flow field inside the rotor passage forced these effects to be appeared toward the midspan. (Chaluvadi, et al., 2001) continued the work for understanding the blade row interaction by using smoke flow visualization and five-hole probes in a single stage high pressure turbine. Figure 1 shows a simple model of vortex transport from the stator passage through the rotor passage. Downstream blade row cuts the stator hub passage vortex in a pretty similar way to the wake. As a result, the bowed vortex seems to have two counter rotating legs: suction side leg (vortex 3) and pressure side leg (vortex 4) as shown in Figure 1. (Chaluvadi, et al., 2001) explained “At the hub the kinematic interaction between the stator and the rotor passage vortices has two effects. First, the suction side leg of the stator passage vortex is displaced radially upward over the developing rotor hub passage vortex. Additionally, the pressure side leg of the stator passage vortex is entrained into the rotor passage vortex. Similar phenomena were observed at the tip of the rotor blade row.”

(Pullan, 2004) studied the secondary flows and blade row interaction in a low speed research turbine facility. Some vortical structures were seen in the rotor passage caused by stator exit flow field. Unsteady numerical simulation was also performed to understand the formation of these vortices. (Porreca, et al., 2008) used PIV and fast response aerodynamic probes to measure steady and unsteady pressure and velocity fields. They found that vortex stretching and wake bending due to the flow interaction with stator are primary source of losses and unsteadiness in the rotor. (Porreca, et al., 2004) in another paper investigated the fluid dynamics and performance of shrouded blades in axial turbines. (Behr, et al., 2005) studied the effect of stator and rotor clocking in HP turbines.

(Gaetani, et al., 2006) experimentally investigated the effect of stator secondary flows on rotor flow field. Two different axial gaps between stator and rotor were studied and compared. The rotor flow field and interaction with stator vortices and wakes were explained. (Persico, et al., 2009) and (Persico, et al., 2012) continued the work and performed parametric study of the blade row interaction. Results showed increase in the magnitude of the rotor inlet vortices change the structure of flow field in the rotor significantly.

**EXPERIMENTAL FACILITY**

The overall layout of the test facility is shown in Figure 2. It consists of a 300 HP (223.71 kW) electric motor connected to a frequency controller which drives a three-stage centrifugal compressor capable of supplying air with a maximum pressure difference of 35 kPa and a volumetric flow rate of $4m^3/s$. The compressor operates in suction mode and its pressure and volume flow rate can be varied by the frequency controller operating between 0 to 66 Hz. A pipe with a smooth transition piece connects the compressor to a Venturi mass flow meter used to measure the mass flow.
through the turbine component. The three-stage turbine has an automated data acquisition system for detailed flow measurement at each blade row location in the radial and circumferential direction.

**Table 1: Turbine dimensions and operating conditions**

<table>
<thead>
<tr>
<th>Stage no., N</th>
<th>Mass flow</th>
<th>Tip Diameter</th>
<th>Hub Diameter</th>
<th>Reference speed</th>
<th>Current speed range</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>3.58 kg/s</td>
<td>685.8 mm</td>
<td>558.8 mm</td>
<td>3000 rpm</td>
<td>1800 to 3000 rpm</td>
</tr>
<tr>
<td>α2</td>
<td>19º</td>
<td>β3</td>
<td>161º</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C_x</td>
<td>41.6 mm</td>
<td>Blade tip design speed</td>
<td>215.34 m/s</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The turbine inlet has an integrated heater that prevents condensation of water from humid air during experiments. Turbine dimension and operating condition is shown in Table 1.

**Figure 2: The overall layout of TPFL-research turbine facility (Schobeiri, et al., 2004)**

**ENDWALL CONTOURING**

Using the conventional trial and error approach utilized in the literature, several cases were numerically simulated in (Schobeiri & Lu, 2013) and efforts were made to improve the known endwall configurations for implementation into a rotating rig. In (Schobeiri & Lu, 2013), a new and physics based endwall contouring method was introduced for turbine blading regardless of the application to HP-, IP-, or LP- turbine. Figure 3 shows the variation of the contour depth along the suction surface.

It is based on the continuous diffusion process and utilizes a prescribed deceleration of the secondary flow velocity from pressure to suction surface by a diffuser type of flow path that produces a desired target pressure. The diffuser raises the pressure on the endwall suction side thus reducing the secondary flow velocity, the strength of the secondary vortices, the associated induced drag forces and the total pressure loss due to the latter. The step-by-step instruction is presented in (Schobeiri & Lu, 2013).

**INTERSTAGE INSTRUMENTATION**

The three-stage air turbine component has a casing that incorporates stator rings to achieve greater versatility. Three traversing systems have five-hole probes with decoders and encoders for accurate probe position mounted upon them (Figure 4). A controller with feedback connected to each system can move the probes at a precision of 1/400 mm (2.5 μm). These three systems are mounted on a base plate, connected with the three T-rings installed in the casing. To seal the three 90° circumferential traversing slots, three T-rings are used. These T-rings prevent leakage of mainstream mass flow through the traversing slots by moving circumferentially. Altogether, this constitutes a traversing unit; it moves the unit in a circumferential direction and is connected to a fourth traversing system which is placed on top of a frame.

**Figure 3: 2nd stage contouring geometry (Lu, 2014)**

**Figure 4: Turbine cross section view of interstage instrumentation**
The traversing unit can move radially from 1 mm above the hub diameter to 1 mm from the blade tip. Each traverse (radial or circumferential) has individual stepper motors, allowing for accurate probe alignment during data acquisition.

Combined total temperature and total pressure rakes are utilized both upstream of the first stator row and downstream of the last rotor row for performance instrumentation. These rakes each combine 4 total pressure probes of the Pitot tube type, and 3 total temperature probes which are calibrated J-type thermocouples, placed equidistantly along the radial direction. Rakes were placed radially at the inlet at 45\°, 135\°, 225\°, and 315\°. Figure 5 shows the Angular position of the five-hole probes at stations 3, 4, and 5. In Figure 6, dimension of five-hole probe that used in the current research is shown.

![Figure 5: Angular position of the five-hole probes at station 3, 4, and 5 (Schobeiri, et al., 2004)](image)

![Figure 6: Dimension of probe that used in calibration for current research](image)

Table 2: Uncertainty analysis for the five-hole probes

<table>
<thead>
<tr>
<th>%uncertainty</th>
<th>Probe 1</th>
<th>Probe 2</th>
<th>Probe 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitch</td>
<td>2.72</td>
<td>1.95</td>
<td>2.83</td>
</tr>
<tr>
<td>Yaw</td>
<td>2.44</td>
<td>1.88</td>
<td>3.01</td>
</tr>
<tr>
<td>Total Pressure</td>
<td>1.23</td>
<td>0.96</td>
<td>1.46</td>
</tr>
<tr>
<td>Static Pressure</td>
<td>1.57</td>
<td>1.11</td>
<td>1.49</td>
</tr>
<tr>
<td>Absolute Velocity</td>
<td>2.47</td>
<td>2.33</td>
<td>2.65</td>
</tr>
</tbody>
</table>

Uncertainty analysis is performed for the five-hole Pitot probes after calibration and data reduction. The experimental variation in pressure measurement is approximately 0.5 Pa from the pressure transducers and 1\°C for temperature measurements, these values are then utilized in the uncertainty equations of the corresponding parameters to obtain the uncertainties in the parameters measured using the five-hole probes. The maximum uncertainties of all the independent parameters measured are thus calculated and are as shown in Table 2.

RESULTS AND DISCUSSION

In this section results of comprehensive performance measurements and interstage traversing at different rotational speeds and mass flow rates are presented.

Contour Plots

Figure 7 presents a general overview of the main flow features in terms of contour lines on the station 3 after rotor blades as measured by five-hole probe in the rotor. Absolute flow angle, relative flow angle, relative total pressure, total pressure, meridian flow angle and Mach number are shown in this figure.

![Figure 7: Contour plots of rotor exit flow (α, β, P_t, P_r, γ and M) at station 3](image)

According to meridian flow angle contour in Figure 7(e) almost 40\% of the span (from 25–65\%) is filled by essentially 2D flow. Close to the hub area, high meridian angle represents high vortical flow due to secondary flows. Similar behaviour is depicted in relative total pressure contour in which relative total pressure has some sudden changes in magnitude in those regions.

Also, a pretty large hub passage vortex centered at 20\% of span can be seen covering up to 15\% of the span. As shown in Figure 1, the interaction between the stator and the rotor passage vortices caused the suction side leg of the stator passage vortex moved radially upwards over the rotor hub
passage vortex. Moreover, the rotor passage vortex swept over the pressure side leg of the stator passage vortex.

At the tip, there is a region with vortical flows that covers almost 10% of the span and it centered at 95% span. It can be concluded that secondary flows and high vortical flows are caused by tip leakage flow. Another vortical structure region is placed at 80% span. This structure could be originated by either the rotor secondary flow or the interaction between stator and rotor casing secondary flow. This structure seems to be slightly weaker than tip leakage flow.

Like the hub stator passage vortex, this vortical structure is caused by the interaction between the tip leakage and stator casing vortex. This interaction moved the stator casing vortex toward the midspan on the suction surface.

The only difference between station 3 and station 5 is that hub secondary flows in station 5 are very limited and weaker compared to station 3. Second rotor has a contoured endwall and station 5 is right after the contouring. This is the consequence of the endwall contouring that has reduced the pressure difference between the pressure and suction surface leading to lower secondary flow vortices.

**Radial Pitchwise Averaged Distribution**

In this section, radial distribution of pitchwise averaged of flow parameters as a function of the immersion ratio $r$ are presented at 3000 rpm.
Figure 10 depicts the radial distribution of pitchwise average absolute flow angle. Some irregularities can be seen for all stations close to the tip start at 85% span. These regions are affected by tip leakage vortices and casing secondary flows which explained in previous section.

At station 3 and 4, areas near hub undergo strong changes in flow angles due to hub secondary flow vortices. These areas can be seen up to 20% of span. On the other hand, station 5 shows smooth flow angle distribution near hub which is due to endwall contouring that reduced the hub secondary flows in that region. The immersion ratio range between $r'=0.3$ to 0.7 shows a moderate change in the range of $5^\circ$.

![Figure 10: Radial pitchwise averaged distribution of the absolute flow angle](image)

![Figure 11: Radial pitchwise averaged distribution of meridian angle](image)

Figure 11 shows radial pitchwise averaged distribution of meridian angle. Station 4 has moderate change in immersion ratio between 0.15 and 0.85. It can be concluded that flow is two dimensional in this region. Station 3 shows more changes near hub and tip. Almost 40% of span in the area between $r'=0.25$ to $r'=0.65$ is two dimensional and the rest is affected by hub secondary flow vortices and tip leakage vortices. Interestingly station 5 doesn’t show any strong changes near hub compare to other stations. This is the result of the contoured endwall which reduced the secondary flow vortices. Station 5 similar to station 3 shows some strong changes near tip from $r'=0.8$ to $r'=1$.

Absolute Mach numbers as function of immersion ratios are shown in Figure 12. Uniform low sub subsonic absolute Mach numbers around 0.1 are encountered downstream of the first and second rotor, stations 3 and 5, whereas the second stator exit Mach number, at station 4, is around 0.3. That’s why five-hole probes were calibrated based on Mach number 0.1 and 0.3.

![Figure 12: Radial pitchwise averaged distribution of the absolute Mach number at 3000 rpm](image)

![Figure 13: Radial pitchwise averaged distribution of the relative velocity](image)
In the rotor, as discussed in previous section, 4 regions of high vortical flows affect the flow field. Near the hub secondary flows play an important role to increase losses at about immersion ratio 0.15. In the center, another region of slightly lower total pressure indicates the presence of relatively thick wakes. As stated previously, pneumatic (five-hole) probes were used for this investigation. These probes inherently have low frequency response; however, the flow within the axial gaps is highly unsteady. To capture the flow details, high frequency response probes such as hot wire probes are required. Figure 8 reveals a relatively thicker wake region immediately downstream of the stator blade trailing edge. The total pressure contour at station 5, (Figure 9), which is downstream of the second rotor, is similar to station 3 except at the hub secondary flows has been reduced by endwall contouring.

Like static pressure, total pressure at station 5 increased radially from hub to tip. Figure 7 shows the contour plot of the total pressure distribution downstream of the first rotor, at station 3. Three different flow regions are presented in this figure. Near the hub due to the combined effect of secondary flow and mixing, a slightly lower total pressure region is visible. In the center, another region of slightly lower total pressure indicates the presence of relatively thick wakes. As stated previously, pneumatic (five-hole) probes were used for this investigation. These probes inherently have low frequency response; however, the flow within the axial gaps is highly unsteady. To capture the flow details, high frequency response probes such as hot wire probes are required. Figure 8 reveals a relatively thick wake region immediately downstream of the stator blade trailing edge. The total pressure contour at station 5, (Figure 9), which is downstream of the second rotor, is similar to station 3 except at the hub secondary flows has been reduced by endwall contouring.

The interstage traversing provides the entire flow quantities, from which the absolute and relative total pressures and thus the total pressure loss coefficients for the stator and the rotor can be determined. By using circumferentially consistently averaged quantities, the total pressure loss coefficients for stator and rotor were defined as:

\[ \zeta_{\text{stator}} = \frac{P_{t3} - P_{t4}}{\frac{1}{2} \rho V^2} \]  (1)

\[ \zeta_{\text{rotor}} = \frac{P_{t5} - P_{t6}}{\frac{1}{2} \rho W^2} \]  (2)

Total pressure loss coefficient is presented in Figure 17. In the stator, hub secondary flows play an important role to increase losses at about immersion ratio 0.15. Near the tip starts at about 90% immersion ratio, effect of casing secondary flows on total pressure loss is visible.

In the rotor, as discussed in previous section, 4 regions of high vortical flows affect the flow field. Near the hub at about
immersion ratio 0.1 is affected by hub secondary flows. In this case since the endwall was contoured, significant reduction in total pressure loss coefficient can be seen.

Second high loss area is around 35% of span. Due to the interaction between stator and rotor, stator hub passage vortex displaced radially upwards and increased the loss in this region.

Third high loss area is at immersion ration 0.8. Similar case happens at the rotor tip, where the stator tip passage vortex and tip leakage interaction causes the vortical structure move toward the midspan.

And finally the highest loss in the rotor happened at immersion ratio 0.95 where the tip leakages dominate the flow field. Pitchwise averaged total pressure loss coefficient is as high as 0.38 (Figure 17).

Performance Results
This section focuses on the impact of the rotating purge flow on the performance of TPFL research turbine with non-axisymmetric endwall contouring. The purge flow investigation involves the reference case without endwall contouring followed by the investigation with endwall contouring. Before taking the final data, several preliminary experiments were conducted to ensure the reproducibility of the data.

After cutting the endwall contouring into the rotor hub, installing the blades, balancing, the rotor was inserted into the casing, instrumented and prepared for performance measurements. Final efficiency measurements presented in Figure 18 shows for the contoured rotor a maximum efficiency of 89.9% compared to the reference non-contoured case of \( \eta_{\text{ref}} = 88.86\% \). This is an efficiency increase of \( \Delta \eta = 1.04\% \), which is almost double the value obtained from the numerical simulation of \( \Delta \eta = 0.51\% \) presented in (Schobeiri & Lu, 2013). In addition to contoured endwall, second rotor blade number was also increased from 40 to 46.
This new method is particularly significant for applications to power generation steam turbines. The high, intermediate and low pressure units (HP, IP, LP) consist of many stages, with HP- and IP-units having moderate Zweifel coefficient. This coefficient makes them particularly suitable for application of this contouring method without changing the blade geometry. Contouring all rotor endwalls of these units will substantially increase the unit efficiency.

Figure 18 shows measured total-to-static efficiency as a function of \( u/c_{\infty} \). The dimensionless parameter \( u/c_{\infty} \) is calculated from:

\[
 u = \omega r_{\text{mean}} \quad c_{\infty} = \sqrt{\Delta H_f} \tag{3}
\]

where \( r_{\text{mean}} \) is average turbine radius and \( \Delta H_f \) is change in total enthalpy from rotor inlet to exit.

Injecting the purge mass flow causes the turbine mass flow to increase and the turbine efficiency to decrease. Figure 19 shows the mass flow distribution as a function of turbine rotational speed with the mass flow ratio MFR as a parameter. As seen the lowest mass flow corresponds to the reference case of MFR=0. Increasing the MFR increases the turbine mass flow. The increased mass flow, however, causes a reduction of the turbine total-to-static efficiency as shown in Figure 20.

**CONCLUSION**

This paper deals with the specific aerodynamics problematic inherent to high pressure (HP) turbine sections. The objective of the present study is to experimentally investigate the effect of endwall contouring on secondary flow behavior and performance of a rotating HP turbine. Decreasing the strength of the secondary flow vortices at the hub and tip regions reduce the secondary flow losses and the potential for endwall deposition, erosion and corrosion due to secondary flow driven migration of gas flow particles to the hub and tip regions. Major efficiency improvement has been achieved by introducing a completely new endwall contouring technology which decreases the strength of the secondary flow vortices. The contouring was cut into the rotor hub of the three-stage TPFL-research turbine using CNC-machining. Efficiency measurements show for the contoured rotor a maximum efficiency of 89.9% compared to the reference non-contoured case of \( \eta_{\text{ref}} = 88.86\% \). This is an efficiency increase of \( \eta = 1.04\% \). This new method is particularly significant for applications to power generation steam turbines. The high-, intermediate and low pressure units (HP, IP, LP) consist of many stages, with HP- and IP-units. Contouring all rotor endwalls of these units will substantially increase the unit efficiency. Interstage results also showed that how endwall contouring affected the flow field after the rotor passage and reduced the secondary flow vortices at the hub.

**NOMENCLATURE**

\( C_t \quad \text{Axial chord length of the rotor blade (C_t=4.16 cm)} \)

\( M \quad \text{Mach number} \)

\( P_t \quad \text{Total pressure (kPa)} \)

\( P_\phi \quad \text{Relative total pressure (kPa)} \)

\( r \quad \text{Radius (m)} \)

\( r^* \quad \text{Immersion Ratio} \quad r^* = \frac{r-r_{\text{hub}}}{r_{\text{tip}}-r_{\text{hub}}} \)

\( U \quad \text{circumferential velocity (m/s)} \)

\( V \quad \text{Absolute velocity (m/s)} \)

\( V_t \quad \text{Tangential component of absolute velocity (m/s)} \)

\( W \quad \text{Relative average velocity (m/s)} \)

\( \alpha \quad \text{Absolute velocity flow angle (°)} \)

\( \beta \quad \text{Relative velocity flow angle (°)} \)

\( \gamma \quad \text{Meridian angle (°)} \)

\( \rho \quad \text{Density (kg/m}^3) \)

\( \eta \quad \text{Efficiency} \)

\( \xi \quad \text{Total presser loss coefficient} \)

\( \omega \quad \text{Angular velocity (rad/s)} \)

**Subscripts**

1 At 1st stage stator inlet
2 At 1st stage stator exit (rotor inlet)
3 At 1st stage rotor exit
4 At 2nd stage stator exit (rotor inlet)
5 At 2nd stage rotor exit
T-S Total to Static

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