DETERMINATION OF THE EFFICIENCY OF A COOLED TURBINE STAGE TESTED IN A COMPRESSION TUBE FACILITY

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Method to derive efficiency
Determination of the mass flow
Effect of the coolant flows
Shaft power
  - acceleration
  - rotor inertia
Losses evaluation
  - mechanical and disk windage
  - heat transfer
Efficiency calculation
  - analysis of the results
Conclusions
Definition of the efficiency: Mechanical Method

- Areodynamic efficiency

$$\eta_{\text{areo}} = \frac{P_{\text{shaft}} + P_{\text{mech}} + P_{\text{wind}} + P_{\text{Heat}}}{\int \int \int \int m_{01} C_p T_{01} ds + \dot{m}_{\text{Coolants}} C_p T_{\text{Coolants}} - \int \int \int \int m_{03} C_p T_{03,is} ds + \dot{m}_{\text{leaks}} C_p T_{\text{leaks}}}

\text{with } T_{03,is} = T_{01} \left( \frac{P_{03}}{P_{01}} \right)^{\frac{\gamma - 1}{\gamma}}
Determination of the mass flow

- The stage mass flow is calculated using a model of the facility
- Results for 3 operating conditions:

<table>
<thead>
<tr>
<th>Stage</th>
<th>Ct3 Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 and ½ stage</td>
<td></td>
</tr>
<tr>
<td>0% rotor cooling [kg/s]</td>
<td>15.27</td>
</tr>
<tr>
<td>2% rotor cooling [kg/s]</td>
<td>15.35</td>
</tr>
<tr>
<td>3% rotor cooling [kg/s]</td>
<td>15.36</td>
</tr>
<tr>
<td>Uncertainty</td>
<td>+/- 1.6 %</td>
</tr>
<tr>
<td>Dispersion</td>
<td>+/- 0.21 %</td>
</tr>
</tbody>
</table>
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\[
\eta_{aero} = \frac{P_{Shaft} + P_{mech} + P_{wind} + P_{Heat}}{\dot{m}c_p T_0 1 - \left( \frac{P_{03}}{P_{01}} \right)_{\gamma}^{\gamma-1} + m_{Coolant} C_p \Delta T_{0_Coolant}}
\]
- The NGV internal coolant flow is taken into account:

- Isentropic power:

$$P_{is} = \dot{m}_{Stator} \cdot C_p(T_{01} - T_{03,is}) + \dot{m}_{StatCool} \cdot C_p(T_{0c} - T_{03,is})$$

<table>
<thead>
<tr>
<th></th>
<th>With cooling</th>
<th>Without cooling</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet mass flow [kg/s]</td>
<td>10.27</td>
<td>10.6</td>
</tr>
<tr>
<td>Coolant mass flow [kg/s]</td>
<td>0.33 (3%)</td>
<td>0</td>
</tr>
<tr>
<td>Real power [kW]</td>
<td>1023.3</td>
<td>1023.3</td>
</tr>
<tr>
<td>Isentropic power [kW]</td>
<td>1094.5</td>
<td>1103.3</td>
</tr>
<tr>
<td>Efficiency at midspan</td>
<td>0.935</td>
<td>0.927</td>
</tr>
<tr>
<td>Difference %</td>
<td>~ 0.9 %</td>
<td></td>
</tr>
</tbody>
</table>
Rotor Coolant

The rotor film coolant flow is taken into account:

- The coolant flow must be mixed with the main flow
- The rotor is performing as a radial compressor
  - the pumping work is taken into account

<table>
<thead>
<tr>
<th>Test #006 3% condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stage mass flow [kg/s]</td>
</tr>
<tr>
<td>Coolant mass flow [kg/s]</td>
</tr>
<tr>
<td>$P$ real [kW]</td>
</tr>
<tr>
<td>$P$ isentropic [kW]</td>
</tr>
<tr>
<td>$P$ pumping [kW]</td>
</tr>
<tr>
<td>$T_{02_{abs}}$ with cooling</td>
</tr>
<tr>
<td>$T_{02_{abs}}$ without cooling</td>
</tr>
<tr>
<td>$\eta$ with cooling</td>
</tr>
<tr>
<td>$\eta$ without cooling</td>
</tr>
</tbody>
</table>
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\[
\eta_{aero} = \frac{P_{Shaft} + P_{mech} + P_{wind} + P_{Heat}}{\dot{m}c_p T_{01} \left[ 1 - \left( \frac{P_{03}}{P_{01}} \right)^{\gamma-1} \right] + \dot{m}_{Coolant} C_P \Delta T_{0Coolant}}
\]
Rotor acceleration

- No power absorption system
- Rotor is accelerated during the test time
- Shaft power \[ P_{Shaft} = I \frac{\partial \omega}{\partial t} \omega \]
- Acceleration is derived by a linear fitting
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\[ \eta_{aero} = \frac{\int I \frac{d\omega}{dt} - P_{\text{mech}} + P_{\text{wind}} + P_{\text{Heat}}}{\dot{m}c_p T_{01} \left[ 1 - \left( \frac{P_{03}}{P_{01}} \right)^{\gamma - 1} \right] + \dot{m}_{\text{Coolant}} C_p \Delta T_{0\text{Coolant}}} \]
Rotor inertia evaluation

The inertia of the cooled rotor is evaluated:

- **Experimental set-up:**
  
  Period 1: The mass is falling down, the rotor is accelerated.
  
  Period 2: The mass is laying in the floor, the rotor is decelerated by the mechanical friction.

- **Results:**

  - Rotor inertia = 17,715 kg·m²  disp = +/- 0.41 %

Quadratic regression coefficients from the periods 1 and 2  *(Paniagua 1997)*
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  - mechanical and disk windage
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Mechanical losses

- Free deceleration of the rotor due to the losses:
  \[ P_{\text{Loss}} = P_{\text{mech}} + P_{\text{Disk}} + P_{\text{ventilation}} \]

- Loss correlation:
  \[ \text{Engine RPM} = \text{N}_{\text{mech}} \times \text{RPM} \]

- Numerical optimisation procedure:
  - Find the loss coefficients which provide the best fitting with the measured free deceleration
  - Axial loading is taken into account

- Axial loading is taken into account

- Losses:
  - Stator losses
  - Rotor losses
  - Bearing losses
  - Ventilation losses

- Losses:
  - Mechanical losses
  - Disk losses
  - Ventilation losses

- Evaluation:
  - 0% condition
  - 3% condition
  - Power [kW]
  - % of total power

- Results:
  - Power [kW]:
    - 14.31
    - 14.29
  - % of total power:
    - 0.93%
    - 0.96%
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\[ \eta_{aero} = \frac{\int \frac{\partial \omega}{\partial t} \omega + P_{\text{mech}} + P_{\text{wind}} + P_{\text{Heat}}}{m c_p T_{01} \left[ 1 - \left( \frac{P_{03}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} \right] + m_{\text{Coolant}} C_p \Delta T_{0\text{Coolant}}} \]
Heat transfer

- Evaluate the heat transfer in the control domain (no work is provided)

• Rotor :
  • Measured $T_{gas}$ and $T_{wall}$ history
  • Heat flux
  • Nusselt distribution \( (\text{Didier 2000, Chana 2000}) \)
    - heat transferred to the rotor blades
    - heat transferred to the rotor endwalls

• Stator :
  From LS89 measurements \( (\text{Arts 1990}) \)
  
  \[
  \begin{array}{c|c}
  \text{Re High } P/p \text{ Nom} & 52.94 \\
  \text{Total heat [kW]} & 52.94 \\
  \% of power & 3.65 \%
  \end{array}
  \]
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\[ \eta_{aero} = \frac{I \frac{\partial \omega}{\partial t} + P_{mech} + P_{wind} + P_{Heat}}{\dot{m} c_p T_{01} \left[ 1 - \left( \frac{P_{03}}{P_{01}} \right)^{\gamma - 1} \right]} + \dot{m}_{\text{coolant}} C_p \Delta T_{0\text{coolant}}} \]
# Efficiency results

<table>
<thead>
<tr>
<th>0%</th>
<th>$T_{01}$</th>
<th>$P_{01}$</th>
<th>Press ratio</th>
<th>Mass flow</th>
<th>$P_{\text{real}}$</th>
<th>$P_{\text{isentr}}$</th>
<th>Acc</th>
<th>Rpm</th>
<th>η</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean</td>
<td>480,74</td>
<td>2,221</td>
<td>2,690</td>
<td>15,265</td>
<td>1526,1</td>
<td>1743,1</td>
<td>1154,5</td>
<td>6513,2</td>
<td>0,8761</td>
</tr>
<tr>
<td>Std %</td>
<td>1,11</td>
<td>0,52</td>
<td>2,55</td>
<td>0,19</td>
<td>1,24</td>
<td>2,20</td>
<td>1,26</td>
<td>0,24</td>
<td>2,30</td>
</tr>
</tbody>
</table>

→ Accurate inlet pressure measurements

→ Large dispersion in $\rho$ is provided by the inaccuracy of $P_{03}$

---

Kiel probe, stage inlet

Kiel+thermocouple stage inlet and outlet
Efficiency results

Take into account the exit pressure $P_{04}$: lower test to test dispersion

Pressure ratio $\pi$ is evaluated as: $\pi^* = \frac{P_{01}}{P_{04} + \Delta P_0}$

<table>
<thead>
<tr>
<th>%</th>
<th>$T_{01}$</th>
<th>$P_{01}$</th>
<th>Press ratio*</th>
<th>Mass flow</th>
<th>$P_{\text{real}}$</th>
<th>$P_{\text{isentr}}$</th>
<th>Acc</th>
<th>Rpm</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean</td>
<td>480,74</td>
<td>2,221</td>
<td>2,751</td>
<td>15,26</td>
<td>1526,1</td>
<td>1778,6</td>
<td>1154,5</td>
<td>6513,2</td>
<td>0,8582</td>
</tr>
<tr>
<td>Std %</td>
<td>1,11</td>
<td>0,52</td>
<td>0,84</td>
<td>0,19</td>
<td>1,24</td>
<td>1,51</td>
<td>1,26</td>
<td>0,24</td>
<td>1,35</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>%</th>
<th>$T_{01}$</th>
<th>$P_{01}$</th>
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<th>Mass flow</th>
<th>$P_{\text{real}}$</th>
<th>$P_{\text{isentr}}$</th>
<th>Acc</th>
<th>Rpm</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean</td>
<td>480,66</td>
<td>2,223</td>
<td>2,671</td>
<td>15,36</td>
<td>1488,7</td>
<td>1744,3</td>
<td>1123,6</td>
<td>6520,5</td>
<td>0,8519</td>
</tr>
<tr>
<td>Std %</td>
<td>1,06</td>
<td>0,62</td>
<td>0,72</td>
<td>0,23</td>
<td>1,68</td>
<td>1,15</td>
<td>1,95</td>
<td>0,22</td>
<td>1,34</td>
</tr>
</tbody>
</table>
Efficiency results

- the pressure $P_{04}$ is not uniform

Presence of 6 module struts

Total pressure in plane 04

![Graph showing pressure distribution with probes and struts labeled]
Efficiency results

- 1 and ½ stage configuration:

Accurate efficiency evaluation
single test uncertainty +/- 1.44%

Most sensitive parameters:

Pressure ratio
- large dispersion detected in $P_{03}$
- take into account $P_{04}$

Acceleration
- dispersion is affecting sensibly the efficiency
Conclusions

A complete overall efficiency analysis is presented

- The mass flow is calculated thanks to an accurate modelling of the facility

- The effect of the NGV and rotor coolant flow is analysed:
  - non negligible contribution of the NGV coolant (3% of $\dot{m}_{\text{Stage}}$)
  - small influence of the rotor coolant (0.7 % of $\dot{m}_{\text{Stage}}$)

- An accurate evaluation of the mechanical losses ($\sim 1.4 \%$ of $P_{\text{tot}}$), heat transfer ($\sim 3 \%$ of $P_{\text{tot}}$) and rotor inertia is performed

efficiency is calculated and critical parameters are identified
Future plans

- Improve the accuracy of the exit rotor pressure measurements:
  - investigate the pressure variation at the stage exit
  - install rakes of Kiel probes

- Improve the accuracy of the rotor acceleration
  - investigate the accuracy of the actual system
  - design a new RPM detection system if required