

Article

# Experimental Investigation of the Sensitivity of Forced Response to Cold Streaks in an Axial Turbine <sup>†</sup>

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**Abstract:** In turbomachinery, geometric variances of the blades, due to manufacturing tolerances, deterioration over a lifetime, or blade repair, can influence overall aerodynamic performance as well as aeroelastic behaviour. In cooled turbine blades, such deviations may lead to streaks of high or low temperature. It has already been shown that hot streaks from the combustors lead to inhomogeneity in the flow path, resulting in increased blade dynamic stress. However, not only hot streaks but also cold streaks occur in modern aircraft engines due to deterioration-induced widening of cooling holes. This work investigates this effect in an experimental setup of a five-stage axial turbine. Cooling air is injected through the vane row of the fourth stage at midspan, and the vibration amplitudes of the blades in rotor stage five are measured with a tip-timing system. The highest injected mass flow rate is 2% of the total mass flow rate for a low-load operating point. The global turbine parameters change between the reference case without cooling air and the cold streak case. This change in operating conditions is compensated such that the corrected operating point is held constant throughout the measurements. It is shown that the cold streak is deflected in the direction of the hub and detected at 40% channel height behind the stator vane of the fifth stage. The averaged vibration amplitude over all blades increases by 20% for the cold streak case compared to the reference during low-load operating of the axial turbine. For operating points with higher loads, however, no increase in averaged vibration amplitude exceeding the measurement uncertainties is observed because the relative cooling mass flow rate is too low. It is shown that the cold streak only influences the pressure side and leads to a widening of the wake deficit. This is identified as the reason for the increased forcing on the blade. The conclusion is that an accurate prediction of the blade's lifetime requires consideration of the cooling air within the design process and estimation of changes in cooling air mass flow rate throughout the blade's lifetime.

**Keywords:** axial turbines; aerodynamics; aeroelasticity; blade vibrations; forced response; experimental investigations



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## 1. Introduction

Reducing the environmental footprint is one of the main challenges in the current design process of aircraft engines. New highly optimised designs further reduce safety margins of engine parts, especially in the axial turbine, leading to higher aerodynamic, mechanical, and thermal loads. This results in extensive wear of turbine blades, which changes the aerodynamic flow field around the blades and impacts the aeroelastic excitations and responses [1]. Within the collaborative research centre (CRC) 871, these effects are studied, and a decision-based regeneration process is built to understand the influence of wear on aerodynamics, mechanics, and aeroelasticity and how to minimise the costs of aircraft engine overhaul [2].

Furthermore, new technologies, e.g., hydrogen combustion or increased combustion temperature, will lead to higher thermal demands of the components in the flow channel. Thus, increasing the necessity of cooling from the first stages of the high-pressure turbine leads to a higher cooling air mass flow rate. Malfunctions in burners and widening of cooling channels might lead to an inhomogeneous flow field that excites downstream blades. An example was shown by [3] that damaged fuel nozzles distort the flow, and the mixing of such distortions was studied by [4,5]. Ref. [6] measured different burner configurations of real aircraft engines and confirmed the inhomogeneous flow field. These and further numerous aerodynamic studies on hot streaks are complemented by aeroelastic studies where [7] investigated harmonic excitations due to burner-induced disturbance of the flow. It was found that the first harmonic excitation translates through the high-pressure and low-pressure turbines up until the last stages, which influences the blade force. Higher harmonics have less impact but can also influence the blade forcing, as shown numerically by [8] with a three-stage high-pressure turbine as a test case. In [9], it was shown for a uniform, a combustor modelled, and a combustor defect inlet profile the significance of considering inlet distortions for aeroelastic evaluations. Ignoring these distortions would lead to miscalculation of high-cycle fatigue.

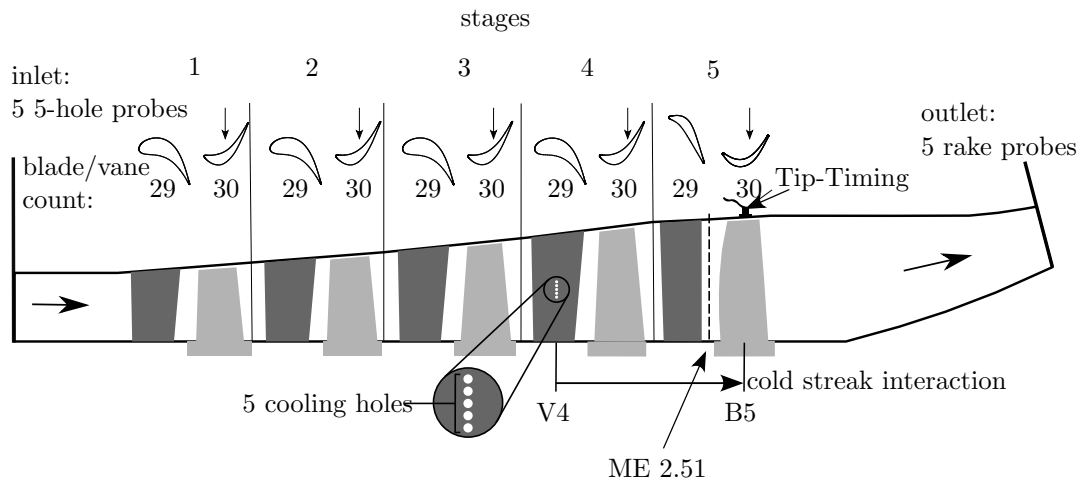
However, not only hot streaks might be the cause of higher blade vibrations, but cold streaks due to the cooling air of high-pressure turbine blades and vanes can be a source of changes in aeroelastic behaviour. Cold streaks have a similar effect as hot streaks, where temperature inhomogeneities across the circumference occur. However, the temperature and density gradient is inverse for the cold streak to the hot streak. For hot streaks, a mixing downstream occurs over several stages. If that is also the case for cold streaks, this might lead to high engine order excitation. For example, the cooling from a high-pressure vane can result in an excitation in the low-pressure turbine. These engine orders would need to be considered when designing a turbine. Furthermore, ref. [10] investigated the effect of hot and cold streaks on the downstream flow field in a 1.5-stage axial turbine. They showed that hot streaks are migrated toward the pressure side and cold streaks toward the suction side. This effect coincides with the findings by [11], where the change in density results in a change of relative flow angles in the rotating reference frame of a blade row. Periodic changes in regions with higher and lower temperatures result in different inflow angles of the blade, which might change the forcing on the blade. Based on these indices, there is a possibility of an excitation source that has not been considered in the past. Up to this point, the research focus of cooled turbine blades is the cooling efficiency (see [12]), wake mixing (see [13]), or impact on performance parameters (see [14,15]) without considering possible high or low engine order excitation of downstream blades.

For this purpose, a test setup in a multi-stage axial turbine is built, and the influence of a cold streak on the blade vibration of a downstream blade is experimentally investigated. Due to the novelty of the application of cold streaks in such a large test rig, a first analysis of the aerodynamic flow field is conducted, and afterwards the change in blade vibration is evaluated.

## 2. Experimental Setup

The experimentally investigated test rig is the five-stage axial turbine of the Institute of Turbomachinery and Fluid Dynamics (TFD). This configuration has 29 vanes and 30 blades through all stages. The last stage was designed for aeroelastic similarity with a last-stage low-pressure turbine of a commercial jet engine by [16]. For convenience, we will denote the vane row of this last stage as V5 and the blade row as B5. Figure 1 provides an overview of the overall turbine, measurement planes, and injection of the cold streak. Note that experimental studies with the institute's test rig with geometric variations in stages 4 and 5 have been carried out for three different operating points by [17,18]. In both publications, the experimental results coincide with aerodynamic and aeroelastic CFD solutions for a reference case with original geometry, and additionally, it was found that geometric

variations have a measurable impact on the forced response of the investigated rotor stage B5.



**Figure 1.** Schematic overview of the experimental setup.

At all times, the flow condition at the inlet is measured with five five-hole probes distributed around the circumference. At the outlet, there are five rake probes with four total pressure and total temperature probes and one five-hole probe on each rake probe. Different flow angles behind B5 for different operating points require rotatable rake probes, which are realised with an electric motor. Overall, these measurements ensure that the operating point is maintained constant between different investigations and are used for calculating performance parameters like isentropic efficiency or thermal power. Furthermore, the inlet plane is used for defining the corrected mass flow rate and corrected rotational speed, which are held constant for each operating point across the experiment. The correction of mass flow rate and rotational speed is necessary due to the open-loop test rig. This ensures comparability between testing on different days as well as between the reference and cold streak case.

Five-hole probe measurements are conducted in measurement plane ME 2.51 between V5 and B5. The probe is traversed circumferentially over more than one pitch to quantify flow periodicity. All five-hole probes used are calibrated in a high-velocity calibration tunnel to obtain calibration coefficients and automatically calculate total pressure, static pressure, total temperature, flow angles, and Mach number. Furthermore, the pressure profile is measured with static pressure tabs on V5 at 50% channel height on each suction and pressure side. All pressure and temperature measurements are normalised with respect to the pressure or temperature drop across the axial turbine. The normalised values, total pressure  $p_{\text{norm}}$ , and temperature  $T_{\text{norm}}$  are calculated with

$$p_{\text{norm}} = \frac{p_t - p_{\text{out}}}{p_{\text{in}} - p_{\text{out}}} \quad \text{and} \quad T_{\text{norm}} = \frac{T_t - T_{\text{out}}}{T_{\text{in}} - T_{\text{out}}}, \quad (1)$$

which allows a better comparison between the reference and cold streak measurements.

A commercial tip-timing system from AGILIS Measurement Equipment is installed in stage 5 to measure the vibration amplitude of B5 near the trailing edge. There are eight probes circumferentially distributed as described in [17]. Synchronous vibrations are investigated, and the data are analysed with a least squares model fitting. The forcing frequency, phase, blade vibration amplitudes, and model fit are calculated.

During the design process of the fifth stage, three different operating points were selected based on machine loading and resonance crossings in the Campbell diagram; see Figure 2. The same operating conditions are used for the running of the turbine in this work. Near the low-load operating point OP1 exists a crossing with the first structural

mode (first bending mode) of the blade and engine order (EO) 29. The same EO has a crossing with mode 2 (first torsional mode) for the part-load operating point. Additionally, blade mode 1 is excited by EO 14 and EO 15 near the part-load operating point OP2, which is shown in Figure 2. The design operating point OP3 has a crossing with EO 8 and the first mode, as well as EO 15 and EO 14 with the second mode. The resulting boundary conditions and the crossings for the different operating points are shown in Table 1.

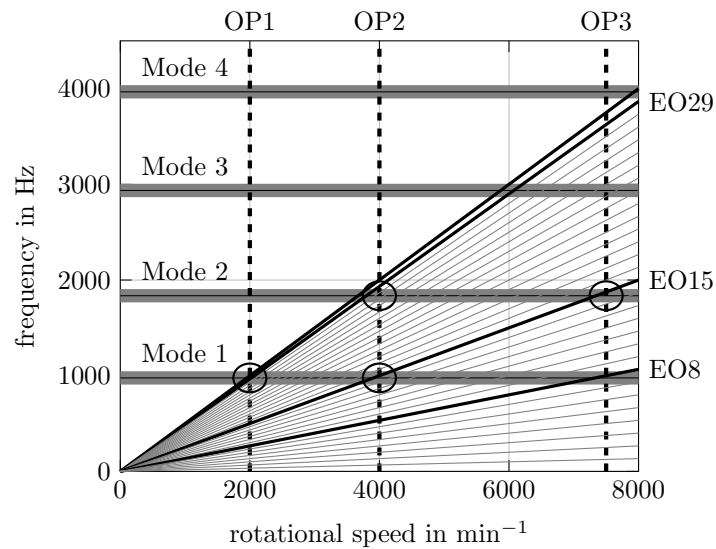


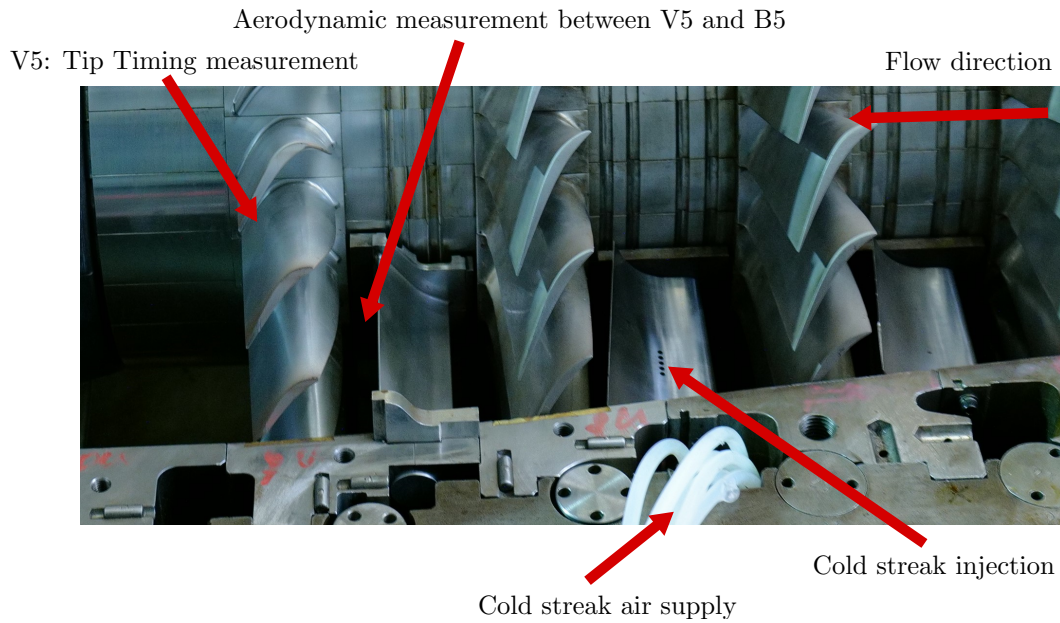
Figure 2. Campbell diagram.

Table 1. Operating points.

| Operating Point                             | OP 1 | OP 2                 | OP 3         |
|---------------------------------------------|------|----------------------|--------------|
| corr. mass flow rate in kg/s                | 6.65 | 7.54                 | 8.5          |
| corr. rotational speed in $\text{min}^{-1}$ | 2171 | 4312                 | 7500         |
| pressure ratio                              | 1.33 | 1.58                 | 2.74         |
| target outlet temperature in K              | 320  | 324                  | 330          |
| engine order excitation/mode                | 29/1 | 29/2<br>15/1<br>14/1 | 15/2<br>14/2 |
| rel. cooling air mass flow rate in %        | 1.6  | 1.3                  | 0.6          |
| cooling air temperature in K                | 305  | 308                  | 320          |

The cold streak is applied through the fourth-stage stator vane, V4. There are five cooling holes with a diameter of 2 mm on the pressure side of each vane, where the third cooling hole is in mid-span (see Figure 3), which allows for investigating the radial migration of cold streaks across the downstream stages. Air is provided by an external screw compressor, and every vane is connected independently with the main air source. Therefore, different engine orders can be investigated for the different operating points by cutting some blades from the air source. For example, EO 15 is excited by injecting cooling air through 15 blades and disconnecting every other blade. This results in a pattern where EO 14 and EO 15 are simultaneously excited. Based on a frequency analysis with normalised amplitudes, the forcing for each case (with 14 or 15 cold streak vanes) is equal. The temperature of the cold streak is measured just outside the blading area inside the outer turbine casing, where the pipes are connected to each vane. The static pressure is measured in a plenum outside of the turbine. The pressure drop and nominal mass flow rate through each piping and vane were measured before the testing to verify uniform pressure drop across the vanes. The mass flow rate of one vane was measured during

operation to obtain the final cooling air mass flow rate for each operating point. This results in a 95%-confidence interval of  $\pm 0.003\%$  measurement uncertainty of the relative cooling air mass flow rate through each blade. The relative mass flow rate of the cooling air as a fraction of the total mass flow rate in the axial turbine and the absolute cooling air temperature are shown in Table 1. For the reference and cold streak measurement, the corrected mass flow rate and the rotational speed have been held constant for each operating point. The whole setup allowed tip-timing measurements of both cases on the same day without disassembly by switching the main air source on or off.



**Figure 3.** Close-up of the experimentally investigated stages and cold streak application.

### 3. Operating Point

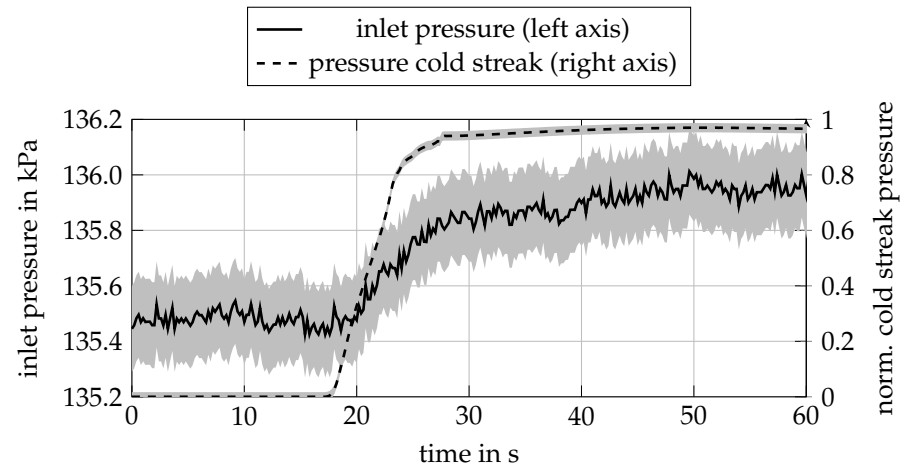
Applying the cold streak during the operation changes some of the main parameters of the axial turbine. In Figure 4, the normalised pressure of the cold streak and the inlet pressure of the five-hole probes are shown for a transient process during the application of the cold streak. The normalized cold streak pressure is calculated by dividing the pressure in the plenum by the cold streak pressure in steady state. After 19 s, the cold streak is switched on, and within 6 s, the pressure increases close to the maximum steady pressure. At the same time, the inlet pressure increases by 500 Pa. Note that to reach steady-state for the inlet pressure, it needs a longer time period; however, no decrease in pressure level over time was found.

The same increase of pressure is observed at the outlet with a lower relative magnitude. This leads to an overall increase in pressure ratio for the cold streak, even if the corrected mass flow rate and corrected rotational speed are kept constant across both cases. Furthermore, the total power output of the axial turbine increases due to the high-pressure cold streak, with a maximum relative increase of 7.6% for OP2 EO29 (see Figure 5). Even the thermal efficiency is increasing for the cold streak by a maximum of 3.4% (0.03% total thermal efficiency). The pressure ratio, thermal power output, and isentropic efficiency are calculated by

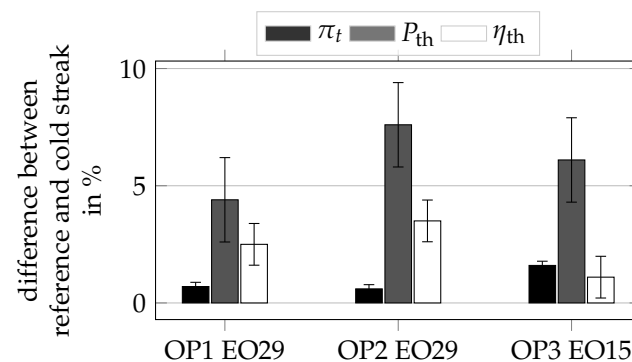
$$\pi_t = \frac{p_{t,in}}{p_{t,out}}, \quad P_{th} = \dot{m}c_p(T_{t,in} - T_{t,out}), \quad \text{and} \quad \eta_{th} = \frac{1 - \frac{T_{t,out}}{T_{t,in}}}{1 - \frac{p_{t,out}^{\frac{\kappa-1}{\kappa}}}{p_{t,in}^{\frac{\kappa-1}{\kappa}}}} \quad (2)$$

with heat capacity ratio  $\kappa$  calculated at the inlet of the turbine, heat capacity at constant pressure  $c_p$  calculated at the inlet of the turbine, and mass flow rate  $\dot{m}$ . This excludes

the consideration of additional mass flow rate and pressure from the cold streak. Adjusting the calculation of thermal efficiency to include the cold streak could be carried out by [19]; however, there are still some important variables missing that have to be assumed. Therefore, for this work, the thermal isentropic efficiency between the inlet and outlet will be discussed.



**Figure 4.** Inlet and cold streak pressure during the switch on the cold streak for OP1 EO29 within 95% confidence interval. Lines are measured value and grey area the measurement uncertainty.



**Figure 5.** Change in total pressure ratio  $\pi_t$ , thermal power  $P_{th}$ , and isentropic efficiency  $\eta_{th}$  for the three different operating points due to the cold streak within 95% confidence interval.

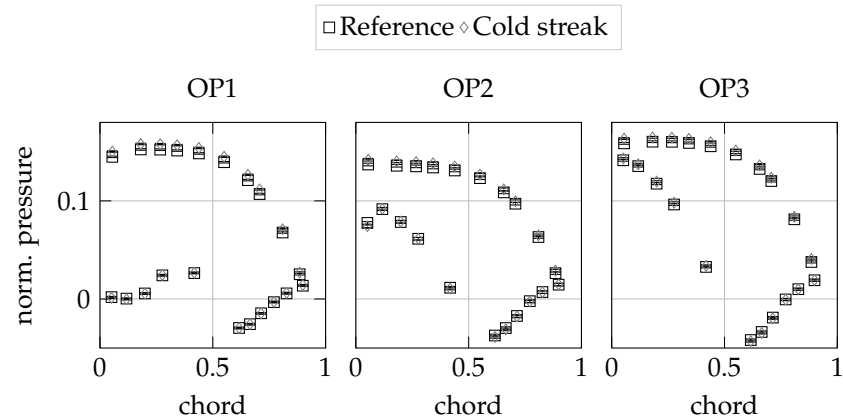
This is due to two effects:

1. Additional mass flow rate from the cooling air, and
2. potential effect upstream to the machine inlet.

Note that the increase in thermal efficiency does not mean that cooling air will increase an axial turbine's thermal efficiency in all cases, but it is important to investigate the change in the operating point in more detail for further understanding. There are two options for operating a test rig: either keep the corrected mass flow rate and rotational speed constant or control the turbine at a constant pressure ratio; thus, thermal power output. In the case of the (old) turbine test rig at the TFD, only the first was possible. For aircraft engines, the latter would have been closer to reality, where the thrust needs to be constant for the engine to achieve the same aircraft speed with different cooling air mass flow rates.

The observation from the performance data is confirmed by the profile pressures of the three different operating points (see Figure 6). On the suction side, the pressure profile is almost equal between reference and cold streak. However, on the pressure side, an increase of the pressure profile for the cold streak case of about 0.5% for OP1 and OP2 as well as a 0.4% increase in OP3 is observed. This increase is over the entire span of the blade on the pressure side. In OP1, the low-load operating point, a large separation bubble is detected

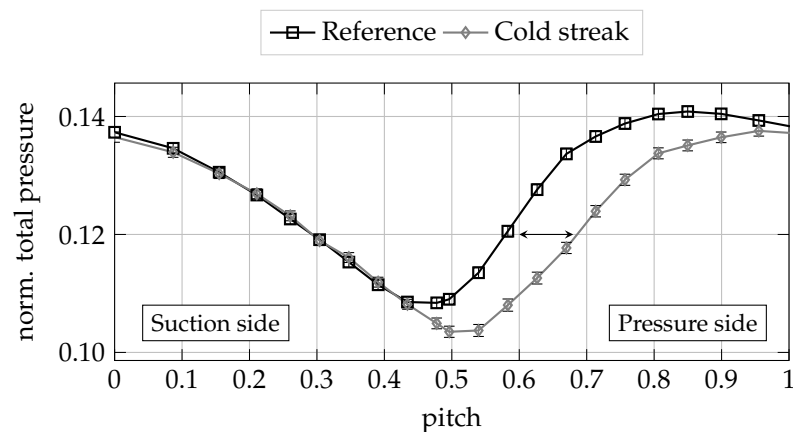
in both cases between the leading edge and about 40% chord length. This indicates high blade incidence towards the blade's suction side in this off-design operating point. This separation bubble is not observed for OP2 and OP3.



**Figure 6.** Pressure profile on V5 at 50% channel height for all operating points. 95% confidence interval in scale of the marker.

#### 4. Aerodynamic Flow Field

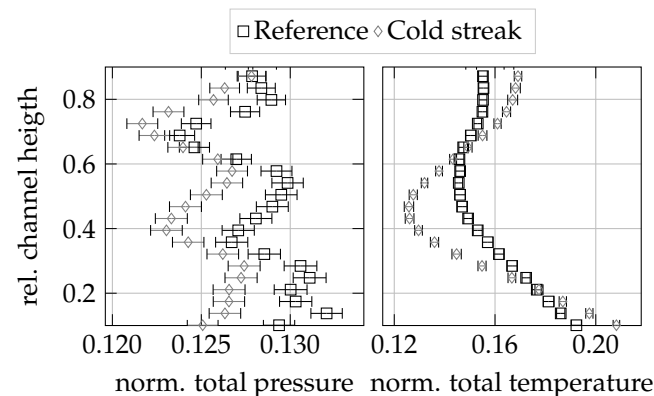
The five-hole probe measurements in OP1 are shown for a wake in Figure 7. Behind V5, the normalised total pressure in OP1 at 40% channel height agrees with the observation of the profile pressure in OP1. The total pressure at the suction side is almost equal between reference and cold streak, while the wake area increases by 10%, as shown in Figure 7. The overall maximum total pressure in the main flow at 80% pitch is decreased by 0.5% for the cold streak. The increase of total pressure in the main stream at 80% pitch in the reference case does not occur for the cold streak case.



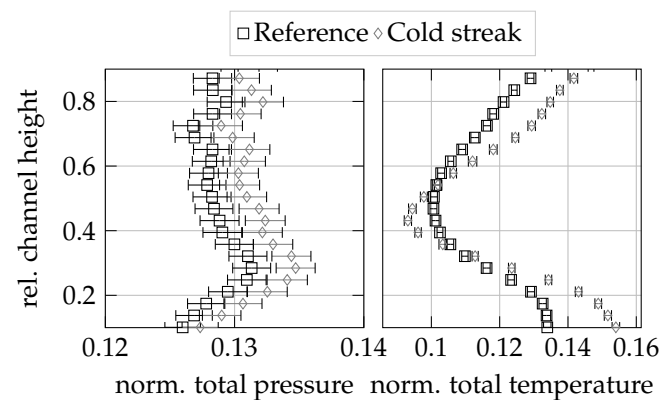
**Figure 7.** Norm. total pressure behind V5 for OP1 at 40% channel height.

The circumferentially area-averaged results of the measurement plane ME2.51 for the normalised total pressure show the same shape for all operating points between reference and cold streak. In OP1, a shift of the local minima and maxima of 4% is observed towards the casing for the cold streak case. A decrease in overall total pressure level for the cold streak in OP1 contradicts the increase in total pressure level in OP2 and OP3. The most significant change is in OP3, with an increase of 1% normalised total pressure. Therefore, no local influence on the total pressure profile is found due to the cold streak. This coincides with the studies on hot streaks, where the total pressure profile is unchanged between reference and variations (see [10,11]). Variations of total temperature are only expected by larger added mass flow rates that change the mass flow distribution across the radial height. The total temperature profile in OP1 shows a reduction in normalised total temperature

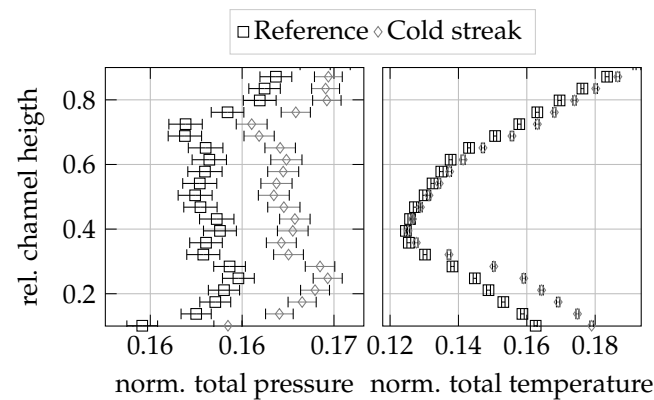
of 2.5% at 40% relative channel height, while an increase in total temperature is observed near the hub and shroud of less than 1% due to the cold streak. The shape of the total temperature profile is changed significantly with an increase between maximum and minimum total temperature for the cold streak in the domain, as shown in Figure 8. For OP2, the total temperature profile shows similar behaviour in the case of the cold streak, where the normalised total temperature is increased by 2% near the hub and shroud and decreases at 40 % relative channel height (see Figure 9). In OP3, there is no reduction of normalised total temperature due to the cold streak but an increase in temperature again near the hub and shroud. However, the shape, especially near the hub, of the normalized total temperature profile is changed again in both cases (see Figure 10). This increase in total temperature can be explained by the performance machine data, in which the application of the cold streak results in higher total pressure and temperature at the inlet relative to the increase at the outlet. Overall, a local influence is measured of the cold streak as well as an increase in the distorted total temperature profile. By increased overall mass flow rate and reduction of relative cooling air mass flow rate, the mixing of the cold streak is increased. The change in total pressure might lead to an overall change of forcing and damping in the downstream B5 row. The different temperatures across the radial height lead to changes in the relative inflow angle of B5 that can be an excitation source. However, the biggest inhomogeneities occur near mid-channel, where the influence on aerodynamic forcing is typically less relevant as at the blade's tip. The downstream migration of these thermal streaks coincides with the observations on hot streaks.



**Figure 8.** Area-averaged measurement plane ME2.51 for OP1. 95% confidence interval in back-to-back measurements.



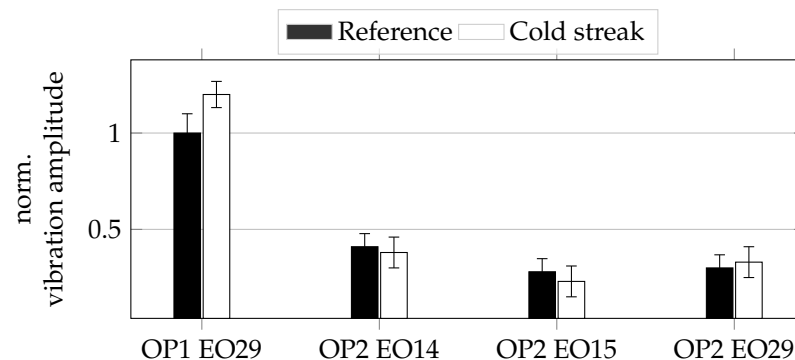
**Figure 9.** Area-averaged measurement plane ME2.51 for OP2. 95% confidence interval in back-to-back measurements.



**Figure 10.** Area-averaged measurement plane ME2.51 for OP3. 95% confidence interval in back-to-back measurements.

## 5. Blade Vibrations

All tip-timing measurements are conducted up to four times to attain a repeatable result. The final vibration amplitudes analysed are the arithmetic mean of all calculations and for all blades averaged with a 95% confidence interval. In Figure 11, the normalised vibration amplitudes of B5 are shown for OP1 and OP2. The normalised vibration amplitudes are calculated by dividing the averaged blade vibration with the averaged blade vibration of OP1. For OP3, it was not possible to obtain accurate blade vibrations for the considered engine orders due to high axial vibration, which resulted in high noise of the signal.



**Figure 11.** Vibration amplitude of B5 with 95% confidence interval.

The cold streak increases the averaged vibration amplitude by 20% in OP1, decreases for EO14 and EO15 in OP2 slightly within measurement uncertainties, and increases in a small margin for EO29 in OP2 within measurement uncertainties. The EO29 for OP1 has the highest relative cooling mass flow rate and therefore shows the biggest impact, which coincides with previous aerodynamic measurements discussed (e.g., Figure 8). The same is observed for OP2 EO29, with less overall impact. However, the relative cooling mass flow rate for EO14 and EO15 in OP2 is half of EO29 OP1. Therefore, less influence on the vibration amplitude is expected and confirmed. Furthermore, due to a vane count of 29, the excitation of EO14 or EO15 by 14 or 15 vanes with cooling air splits to both engine orders. Applying 15 vanes with cooling air also results in an excitation of EO14. This might result in less influence on each individual engine order considered.

This behaviour is confirmed by the individual blade amplitudes from a single test run (see Figure 12). Most of the blades have an increased vibration amplitude for the cold streak case in OP1, and only three blades clustered together have a lower vibration amplitude. Additionally, the strong mistuning of the bladed rotor can be observed, where some blades reach 100  $\mu\text{m}$  vibration amplitude while some have just 30  $\mu\text{m}$  amplitudes. For the reference case, blades 3 and 15 show a relatively lower blade response in comparison with the averaged amplitude. This changes for the cold streak case, where both blades show

a higher response. Furthermore, the clustering of blades with reduced vibration amplitudes (e.g., blades 24–27) or increased amplitudes (e.g., blades 9–11) in comparison with the averaged vibration amplitude is observed. The small change of the averaged vibration amplitude is confirmed with the individual blade amplitudes in OP2 EO14, as shown in Figure 13. Some blades have a small increase in vibration amplitude; however, most changes are well within measurement uncertainties. The individual increase in vibration amplitude is confirmed for all TT measurements on the operating points (e.g., blade 2 for OP1 EO29 and blade 13 for OP2 EO14).

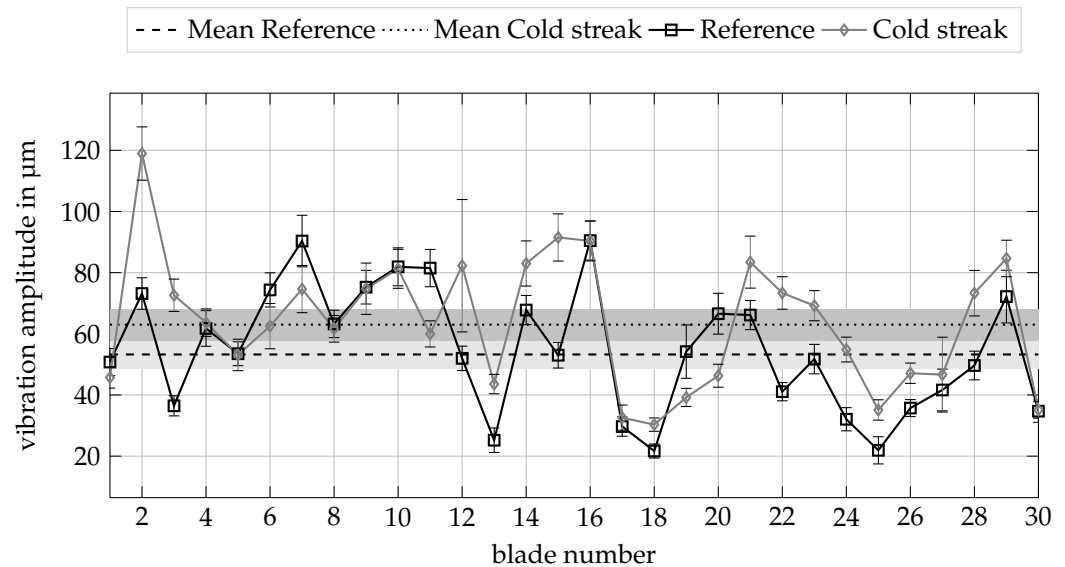


Figure 12. Individual blade vibration amplitude of OP1 EO29.

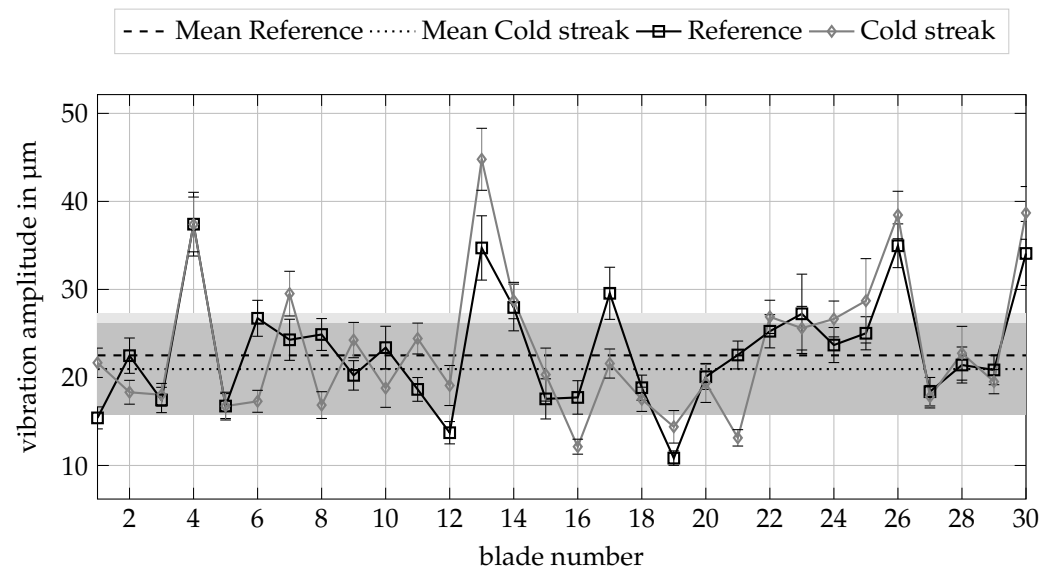


Figure 13. Individual blade vibration amplitude of OP2 EO14.

The range of peak blade frequency is between 947 and 966 Hz for OP1 EO29, as shown in Figure 14. This frequency is determined by the maximum vibration amplitude. The frequency of the cold streak and reference coincide for most of the blades. Blades 13 and 25 are outlying with deviations over 5 Hz, which could be due to the running up and down of the turbine. OP1 has the lowest rotational speed, which could result in small changes of the fixture in the fir tree of the rotor. For OP2 EO14, where the same mode is excited, the vibration amplitudes coincide for even more blades (see Figure 15). Even so, the relative

frequency mistuning between the blades in OP1 EO29 and OP2 EO14 is the same in both cases; the absolute frequency is increased by about 2 Hz due to stiffening effects with higher rotational speed.

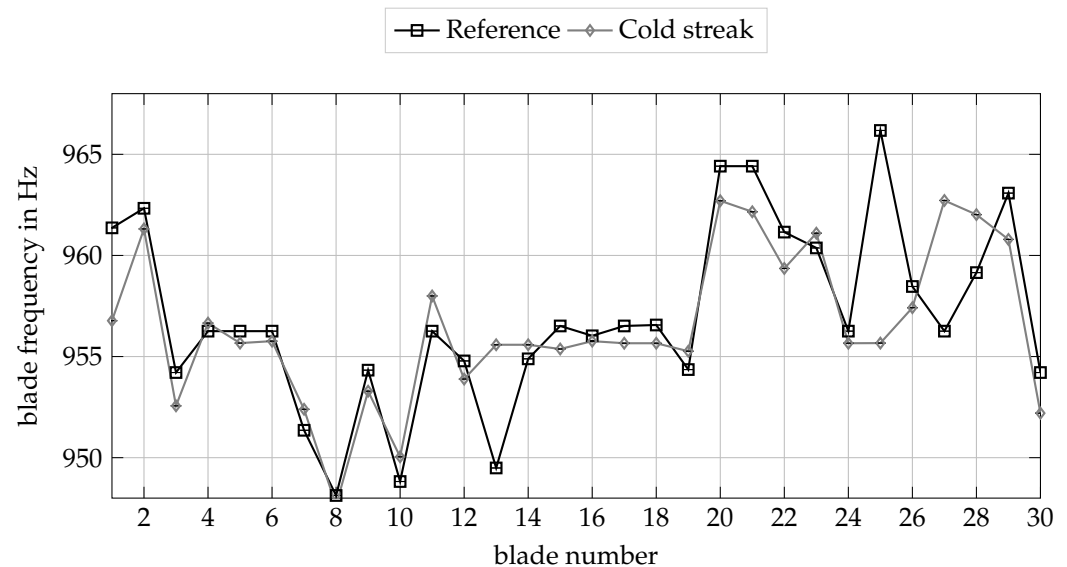


Figure 14. Individual mode frequency of the blade of OP1 EO29.

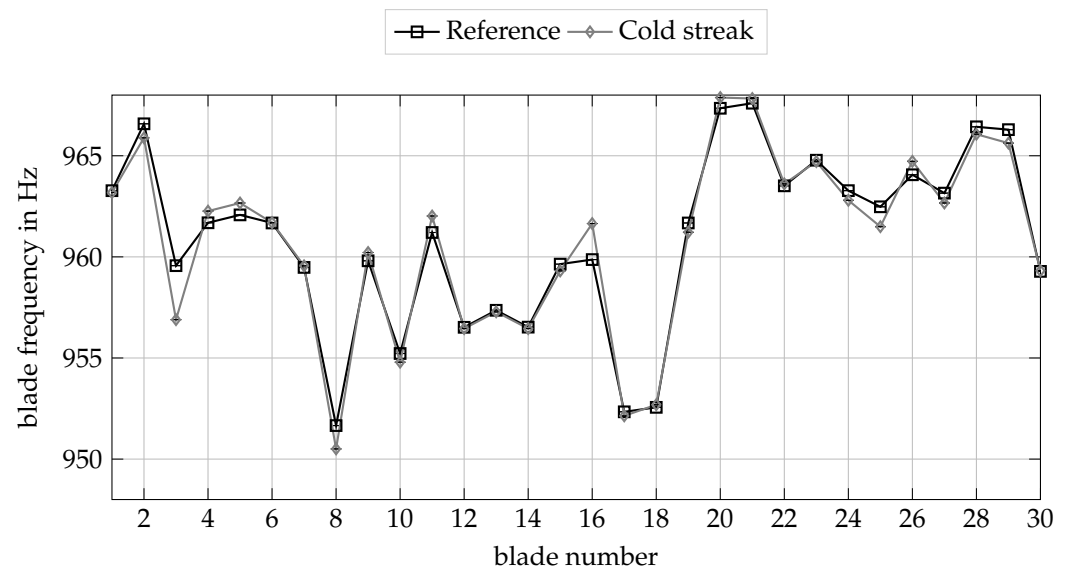


Figure 15. Individual mode frequency of the blade of OP2 EO14.

The change in vibration amplitude is based on three simultaneously occurring effects. First, changes in overall mass flow rate in the machine due to the increase in total pressure at the inlet influence the forcing on the blades. This might change the vibration amplitudes if the additional mass leads to similar forcing frequencies as the blade's natural frequency. Furthermore, additional mass flow rate might increase the aerodynamic damping of B5, which reduces the vibration amplitudes during resonance crossings. In the past, in some cases, a linear relationship between mass flow rate and aerodynamic damping could be observed [20]. Second, the additional local mass flow rate due to the cold streak increases the flow potential, which increases the excitation forces, similar to the first effect. These two effects are observed by the increase in wake width in Figure 7. Finally, the local total temperature, at constant total pressure, changes (see Figures 8–10), which results in a different density distribution across the channel. A different density across the

circumference might lead to a periodic change of the relative inflow angle of B5, which is a possible excitation mechanism.

## 6. Conclusions

The influence of cold streaks in a five-stage turbine test rig is investigated experimentally. While cooling air is added, the test rig is operated such that the corrected mass flow rate and corrected rotational speed are kept constant. However, performance parameters, e.g., isentropic efficiency and total pressure ratio, still change due to the influence of the cooling flow. The loading on vane 5 is increased by the cold streak on the pressure side only due to a change of the upstream wake. For the cold streak case, a 10% increase in wake width results in 20% higher averaged vibration amplitudes of the downstream blade at 38% nominal mass flow rate load and EO29 only. For operating points at higher loads (nominal and 50% of base load), the relative mass flow rate of the cooling air decreases, which reduces the impact on the wake width, circumferential total pressure, temperature average, and ultimately the vibration amplitude such that the vibrations are no higher than in the reference case, i.e., the change in vibration amplitudes is within measurement uncertainties for both operating points. The splitting of EO14 and EO15 due to a vane count of 29 can be another reason for reduced influence in OP2. The small change in operating point leads to the conclusion of negligible impact on aerodynamic damping in comparison with aerodynamic forcing, especially for the significant change in vibration amplitude of the low load operating point. The change of such a magnitude is expected due to significant change in aerodynamic forcing, even so the biggest flow inhomogeneities occur near mid-channel. The highest aerodynamic forcing is typically where the highest amplitudes of the mode shape occur. For B5, this is at the trailing edge for mode 1 and trailing and leading edge for mode 2.

These findings are confirmed for most of the individual blade vibrations for different operating points. However, the doubling of vibration amplitude at 50% part load for EO29 of blade 2 has a significant impact on high-cycle fatigue, thus the lifetime of such blades could be reduced. Even though the averaged vibration amplitude at 50% load at EO14 does not show a significant impact on vibration amplitude, the individual blade number 13 shows similar behaviour as blade number 2 at 50% part load and EO29, which might lead to early structural failure.

The experiments show, for the first time, the influence of cooling air in a multi-stage turbine in aerodynamics and aeroelasticity. Cooling air showed a significant impact in the considered turbine configuration on the blades' vibration, in particular in part load where the relative mass flow of the cooling air is high. If cooling air in the future is accounted for in the design process and in the planning of repairs, the otherwise resulting high-cycle fatigue could be avoided.

## 7. Outlook

A numerical setup has been created, and numerical models validated with this experiment. Further parametric studies on parameters like cooling air mass flow rate or temperature are ongoing. Especially the mass flow rate has to be increased for validation of realistic cooling air mass flow rates used in today's aircraft engines. Furthermore, the application of this approach will be used in future projects with more variability in cooling air parameters that allow distinction between mass flow rate-induced and temperature-induced vibrations. The results will be published in the near future.

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### Abbreviations

The following abbreviations are used in this manuscript:

|             |                       |
|-------------|-----------------------|
| p           | pressure              |
| $P_{th}$    | thermal power         |
| T           | temperature           |
| $\eta_{th}$ | isentropic efficiency |
| $\pi_t$     | total pressure ratio  |
| in          | at inlet              |
| norm        | normalized            |
| out         | at outlet             |
| t           | total magnitude       |
| B5          | blade 5               |
| EO          | engine order          |
| TT          | tip-timing            |
| V5          | vane 5                |

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