Influence of Hot Streak Circumferential Length-Scale in Transonic Turbine Stage

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ABSTRACT

A computational study is carried out on the influence of turbine inlet temperature distortion (hot streak). The hot streak effects are examined from both aeromechanical (forced blade vibration) and aero-thermal (heat transfer) points of view. Computations are firstly carried out for a transonic HP turbine stage, and the steady and unsteady surface pressure results are compared with the corresponding experimental data. Subsequent analysis is carried out for hot-streaks with variable circumferential wavelength, corresponding to different numbers of combustion burners. The results show that the circumferential wavelength of the temperature distortion can significantly change unsteady forcing as well as the heat-transfer to rotor blades. In particular, when the hot-streak wavelength is the same as the nozzle guide vane (NGV) blade pitch, there is a strong dependence of the preferential heating characteristics on the relative clocking position between hot-streak and NGV blade. However, this clocking dependence is shown to be qualitatively weakened for the cases with fewer hot streaks with longer circumferential wavelengths.

NOMENCLATURE

C – blade axial chord  
h – spanwise distance  
P – pressure  
T – temperature  
H.S. – hot streak

Subscript:
1 – inlet  
o – stagnation parameters  
av – time averaged value  
isen – isentropic

INTRODUCTION

Gas turbine performance is crucially influenced by the turbine inlet temperature, which is limited by the required blade mechanical integrity and life span. The heating of high-pressure (HP) turbine blades can lead to thermal fatigue and degrade turbine performance. An important issue to be considered during design is the non uniform gas temperature profile supplied to the HP turbine inlet due to circumferential discrete burners. Non-uniform inlet temperatures (‘hot-streak’) can have effects on HP turbine aero-thermal performance (load and efficiency), blade heat transfer and blade aeromechanics (forced vibration/response). As such, predictive capability and understanding of the effects of non-uniform temperature profiles are required to maximize turbine performance and reliability.

Basic understanding of the kinematic behavior of hot streaks and impacts follows the work on general turbomachinery unsteady wake transportation by Kerrebrock and Mikolajczak [1]. It is known that the hot streak causes significant heat load on HP turbine rotor blades, in particular rotor pressure surfaces tend to be heated further. This so called ‘preferential heating’ is due to an enhanced cross-passage fluid movement from the suction surface to the pressure surface in a hot portion of gas due to an increased incidence. This is confirmed experimentally by Butler et al [2], and computationally by Dorney et al [3], Krouthen and Giles [4]. A further modification of the heating effect can result from the NGV-rotor potential interaction, and as such the heating on rotor blades can be reduced by choice of the NGV/rotor blade count ratio, as reported by Shang and Epstein. [5]. Sondak et al [6] also show a considerable dependence of the heating behaviour on the NGV/rotor blade count.

In a turbine stage environment, the lossy fluid within a wake shed from the upstream relatively rotating row will be convected from the pressure surface to the suction surface because of the velocity deficit, so called ‘negative jet’. Similarly the cross-passage movement within a hot streak can be regarded as a ‘positive jet’. When these two opposing movements happen at the same time and in the same location within blade passage, they suppress each other. As such the lossy fluid in a wake will be less likely to be accumulated on the suction side, and the high temperature fluid in a hot streak is less likely to be accumulated on the pressure side. This phasing between the two leads to the clocking dependence of the...
preferential heating effects as studied by Dorney and Gundy-Burlet [7], Takahashi et al [8].

Apart from blade heat transfer, the effects on blade aeromechanics (forced response) will also need to be considered during design. The work by Manwaring et al [9] shows clearly that a temperature distortion at the turbine inlet can go through a multi-stage turbine and generate large unsteady forcing and thus blade vibratory response even in the last stage of the low-pressure turbine in a realistic aeroengine configuration.

The present work is aimed at further examining hot streak behaviour and effects both in terms of heat load and unsteady forcing. The analysis is focused on a transonic turbine stage configuration, noting that many of previous investigations on hot streaks are for low speed configurations. A particular aspect of interest is the circumferential wave-length of the hot streak (‘hot-streak count’). This is particularly relevant as most of the previous studies on the clocking (‘indexing’) effect adopt a hot-streak/NGV count ratio of 1:1. In realistic engine combustion configurations, the number of combustors/burners tends to be much smaller that that of the NGV blades. The aerodynamic loss characteristics of a multi-stage turbine subject to highly distorted inlet flow due to a partial admission are shown to be strongly dependent on the distortion circumferential wave-length, (He [10]). Therefore, it would be of interest to clarify the issue for situations with temperature distortions. The results should offer some guidance on the applicability of the findings based on an equal hot-streak/NGV count, they should also be relevant to the design choice of number of combustors/burners and cooling arrangement.

**COMPUTATIONAL METHOD**

The method used in the present study is a three-dimensional unsteady Navier-Stokes solver for general turbomachinery flow simulations [11]. The flow equations are solved numerically in a cylindrical coordinate system. For the turbulence closure, the option of the mixing length model by Baldwin and Lomax is adopted in the present studies. Some initial test calculations are carried out using the Spalart-Allmaras one equation model, giving very small difference in terms of the blade surface pressure and temperature distributions compared to those by using the mixing-length model.

The governing equations are discretized in space using a cell-centered finite volume scheme, together with the Jameson type blend 2nd and 4th order artificial dissipation to damp numerical oscillations. The baseline temporal integration of the discretized equations is carried out using the explicit four-step Runge-Kutta scheme. The explicit time-marching scheme is subject to a limitation on the length of time-step due to the numerical stability requirement, and this is very restrictive on unsteady viscous flow calculations. This difficulty can be overcome by using the solver options of the Dual-time stepping [12] and the time consistent multi-grid [11].

The computations are carried out in a multi-passage and multi-row domain, as shown in Fig.1 for a mid-span section of a turbine stage. The relatively moving rotor and stator meshes are patched together at the interface. There are two options for the interface treatment. The steady flow multiple-row solution is obtained by using the mixing-plane technique. At each spanwise section, the ‘mixed-out’ variables at both the rotor and stator sides are flux-averaged. The difference in the mixed-out variables across the interface represents a jump in characteristics. The procedure is to drive characteristics jumps to zero in a non-reflective manner. The second method is for unsteady flow calculations. A 2nd order interpolation and correction method enables local instantaneous information to be transferred directly across the interface.

On blade/endwall surfaces, the log-law is applied to determine the surface shear-stress and the tangential velocity is left to slip. This slip wall condition is preferred for unsteady 3D multi-passage calculations because of the relatively coarse meshes to be used. For the present H.P. turbine case, the over-tip leakage effect is not included. For the energy equation, the adiabatic wall condition is applied. This means that the heat transfer between the fluid and the surface is not included. It is recognized that this will introduce errors in calculated surface temperature distributions. But as the cross-passage migration and redistribution of the temperature field associated with the influence of hot streaks are largely determined by the corresponding fluid kinematics, the use of the adiabatic wall condition should not have significant bearing in the qualitative characteristics of rotor blade heat loads caused by different hot streak configurations considered here.

At the circumferential periodic boundaries, the direct periodic (repeating) condition is used in the present study. This means that the numbers of rotor and stator passages included in the domain need to be such as to have the same total circumferential length.

At the inlet, stagnation parameters and flow angles are specified. The detailed inlet stagnation temperature profiles will be described later. At the exit, the pitchwise mean static pressure at each spanwise section is specified, and the local upstream-running characteristic is formulated to drive the pitchwise average pressure to the specified value, while the local pitchwise non-uniformity is determined by the downstream-running characteristic.

**TURBINE STAGE AND UNIFORM INLET RESULTS**

The test configuration used in the present study is a transonic HP stage, MT1, which has been extensively tested for heat transfer and aerodynamic performance at QinetiQ, as described by Chana et al [13]. The rotation speed is 9500 RPM and the calculated stage pressure ratio is about 2.8. The turbine stage has 32 NGV blades and 60 rotor blades. Initial calculations are conducted using the mixing-plane option which needs only one NGV passage and one rotor passage. The unsteady stage computations are carried out in a multi-passage domain. For this case, the domain needs to contain 8 NGV blade passages and 15 rotor blades passages to enable the direct periodic/repeating condition to be applied at the circumferential boundaries. A mesh density of 40x77x40 per NGV passage and
40x89x40 per rotor passage is used, giving 3.12 millions grid points for the total of 23 passages. Fig. 1 shows the stage mesh views on a blade-to-blade section and on a meridional plane.

Fig. 1 Computational mesh for a turbine stage

The first set of calculations is carried out at a uniform inlet stagnation temperature of 444 K, a uniform inlet stagnation pressure 460 kPa, and an exit static pressure 142.8 kPa. A well-established periodic solution can be obtained in 3 beating periods (1500 time steps for each beating period, which covers 15 rotor blade-passing periods and 8 NGV blade-passing periods) when a multi-passage unsteady computation is started from a single-passage steady mixing-plane solution.

Fig. 2 shows the NGV blade surface isentropic Mach number distributions at the mid-span in comparison with the time-averaged experimental data. The calculated results are from a pure steady flow solver (the mixing-plane treatment), and the time-averaged results of the unsteady simulation. The unsteady potential interaction between the NGV and rotor should mainly affect the region near the NGV trailing edge, and this is the area where we can see some clear difference between the steady and unsteady solutions. The unsteady result is closer to the experimental data than the steady one. In particular, around 80% chord on the suction surface, the unsteady solution produces a larger supersonic region similar to that in the experiment, whilst the oblique trailing-edge shock from the mixing-plane solution hits the suction surface farther upstream.

The trailing-edge overshoot of the pressure on the pressure surface is much reduced in the unsteady result. Following a typical flow pattern around a turbine blade trailing edge, both these two features seem to imply that the NGV exit flow angle predicted by the steady mixing–plane treatment would be smaller (hence less turning) than the time-averaged one from the unsteady solution. Overall, the comparison between the unsteady solution and the experimental data for the NGV is very good.

Fig. 2 NGV mid-span isentropic Mach number distribution

For the rotor, the result at the rotor mid-span section is not as good as that for the NGV, as shown in Fig. 3. The pressures are all normalized by the maximum surface pressure, corresponding to the rotor stagnation pressure. The calculated surface pressures for the most of the suction surface are higher than the experimental data. Nevertheless, the time-averaged unsteady calculation is closer to the experimental data than the steady mixing-plane solution. Compared to the mixing-plane results, the time-averaged unsteady flow field around the rotor seems to be subject to a higher incidence, as indicated by the pressure distribution over the frontal part of the suction surface. This is in line with the higher turning provided by the NGV in the unsteady calculations discussed earlier.

Fig. 3 Rotor mid-span surface static pressure distribution

The calculated unsteady pressure variations are compared with the experimental data, as shown in Fig. 4 and Fig. 5. The results are for the unsteady pressures at different chordwise
locations of the rotor blade mid-span section, normalised by the corresponding local time-averaged values. The reference phase-angle is chosen to match the calculated phase with the experimental value for the leading edge on the suction surface.

Fig. 4 Unsteady pressures at different chord-wise positions (SUCTION SURFACE), — Calculation; ■ Experiment.

Note the plot for the suction surface (Fig. 4) is not in the same scale as that for the pressure surface (Fig. 5). It is clear that highest unsteady pressures are around the frontal part of the suction surface. The large phase change (higher than 180°) between the leading edge and 51% chord indicates the suction surface pressure variation is dominated by the sweep of the NGV trailing edge shock wave, typical of transonic turbine stages (e.g. Miller et al[14], Denos et al [15]). The pressure side is, on the other hand, not so directly affected by the NGV shock wave. The unsteady pressure amplitudes change much less along the chord, and the slight phase lead in the rear part of the pressure surface indicates that the local flow is subject to the upstream propagation of reflected waves. Overall, the calculated magnitudes and relative phase angles of unsteady pressures compare well with the experimental data.

ANALYSIS OF INLET TEMPERATURE DISTORTIONS

For the hot streak calculations, stagnation temperature profiles are specified at inlet to the computational domain. The main interest here is to analyse the influence of the circumferential wave length of the hot streaks. The results for two different hot streak configurations, 32 or 8 hot streaks are presented here. The computational domain containing ¼ of the annulus, therefore have 8 or 2 hot streaks as shown in Fig. 6. The stagnation temperature profile is chosen with the same sinusoidal radial distribution of the pitchwise-mean value as shown in Fig. 7. At each spanwise section, the stagnation temperature also varies in a sinusoidal fashion in the circumferential direction, according to a given wavelength (number of hot streaks). For all the cases calculated, the minimum value of the stagnation temperature is taken to be 400K and the maximum is 600K. Hence the peak temperature ratio at inlet is:

$$\frac{T_{\text{omax}}}{T_{\text{omin}}} = 1.5$$
This temperature ratio is typical of those used in previous studies on the hot streak effects. The overall averaged value at the inlet matches closely that for the uniform inlet case presented earlier.

It should be mentioned that combustion exit temperature distortions are typically measured in terms of the Radial Temperature Distribution Factor (RTDF),

$$\frac{\overline{T_0}_{\text{pitch}} - \overline{T_0}_{\text{overall}}}{\overline{T_0}_{\text{overall}} - \overline{T_0}_{\text{combustor entry}}}$$

and the Overall Temperature Distribution Factor (OTDF),

$$\frac{T_{0\max} - \overline{T_0}_{\text{overall}}}{\overline{T_0}_{\text{overall}} - \overline{T_0}_{\text{combustor entry}}}$$

In the present cases with different numbers of hot streaks, the pitchwise averaged value $\overline{T_0}_{\text{pitch}}$, the maximum value $T_{0\max}$, and the overall averaged value $\overline{T_0}_{\text{overall}}$ are kept the same. Hence the two cases should have the same OTDF and RTDF values, regardless of the number of hot-streaks and the corresponding wave-length.

**Rotor Surface Temperatures (32 Hot-Streaks)**

The calculations firstly are conducted for 32 hot streaks, corresponding to Fig. 6b). Note that the number of NGV blades is also 32. The hot streak configuration with the same hot streak-NGV blade count is widely used in previous studies in particular with respect to the hot streak-NGV clocking effects. It is useful to adopt this configuration first to compare the present results with those from the previous work, before looking at the effects of different hot streak wave lengths. For high-speed turbines, the temperature migration due to the hot streak can be clearly traced in terms of contours of entropy due to its convective nature. The mid-span instantaneous entropy contours for the case with the inlet hot streaks located at the mid-passage of NGV blades are shown in Fig. 8a. Those with the hot streaks impinging on the NGV blade leading edge are shown in Fig. 8b.
surface cross-passage movement, ‘negative jet’. However, for a case subject to a non-uniform temperature field, the opposite happens within a hot streak due to an extra relative velocity caused by heating. Consequently, the hot fluid with high entropy would tend to be convected towards the pressure surface. This so called preferential heating is clearly identifiable by the high entropy fluid (in blue) accumulated on the rotor pressure (Fig. 8a). The NGV mid-passage hot streak gives the strongest preferential heating effect because the two opposing cross-passage movements are now completely staggered. The negative-jet convects the lossy fluid in a NGV wake to the suction side without strongly interacting the positive-jet which convects hot fluid towards the pressure side.

It follows then that the preferential heating should be considerably hindered when the hot streaks are impinged on the NGV blades. Now the hot-streaks are in phase with the NGV wakes. The net cross-passage movement will depend on relative strengths of the two opposing mechanisms. For the present case, the temperature difference is quite high so that the cross-passage movement is still determined by the positive jet. But the preferential heating is seemingly weakened (Fig. 8b). Unlike Fig. 8a, the hot fluid accumulation on the pressure now is not easily identifiable. It is conceivable that the preferential heating on the pressure side may disappear or be reversed if the temperature difference (temperature ratio) of hot streaks impinging on NGV blades is sufficiently small.

So far the discussions are based on a 2-D consideration. The temperature migration behaviour is also affected by 3-D effects. The pressure surface temperature distributions on rotor blades at 10%, 50% and 90% span sections are shown in Fig. 9. Firstly some comments need to be made regarding the NGV exit / rotor inlet flow. Note the time-averaged temperature differences around the leading-edge of the rotor blades as shown in Fig. 9. The rotor leading-edge region should not be affected as the rotor cross-passage movement and the preferential heating mechanism. In terms of the pitch-averaged value, the two inlet hot streak cases should give a higher temperature at the mid-span than those at the tip and hub sections, compared to the uniform inlet case (see Fig. 7). The lead-edge temperatures for the hot-streaks located in the NGV mid-passage follow the expected trend. For the impinging hot streaks case, however, the temperature follows the expected variation only at 90%span (Fig.9c). At the mid-span (Fig.9b), the rotor leading-edge temperatures for the two hot streak cases are noticeably different. At 10% span (Fig. 9a), the temperature of the impinging hot streak case is even higher than that of the uniform inlet. Thus the spanwise location of hot streaks as they leave the NGV would depend on the NGV-hot streak clocking. NGV blade surface temperature contours (not shown here) indicate that when hot streaks impinge on the NGV blades, the fluid within a hot streak is radially shifted from the mid-span towards the hub section over the rear part of the NGV blades. A possible explanation is that the radial pressure gradient as required to maintain the swirl at the NGV exit would drive the low momentum fluid in the blade boundary layer and wakes to move inwards. Hence the hot fluid will also be convected radially inward by the NGV secondary flow of this kind if the hot streak impinges on the NGV blade. This should explain why the temperature for the impinging hot streak case is much higher than that of the mid-passage hot-streak case around the rotor leading edge at 10% span (Fig. 9a). The radially inward migration in the NGV row by the secondary flow mechanism should also lead to a relatively lower rotor inlet temperature at the mid-span for the impinging hot-streak case, as shown in Fig. 9b.

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![Fig. 9 Time-averaged temperatures at 3 spanwise sections of rotor pressure surface (32 Hot Streaks)](image-url)
Regarding the flow kinematics inside the rotor passage, there is also a strong secondary flow due to the rotor passage-vortex. Fig. 10 shows instantaneous stagnation temperature contours at an axial mesh plane 90% rotor chord for the impinging hot streak clocking position. The rotor blading sections are leaned along span, positively (pressure surface facing inward) near the hub and negatively near the tip. This kind of compound lean stacking is typically used to reduce the secondary flow in tip and hub regions. For this case, there seems to be a stronger secondary flow due to a passage-vortex in the hub region than that near the tip. In the mid-span region, the contours show the signature of the migration of hot fluid from the suction surface to the pressure surface, as expected from the preferential heating view point. They also indicate clearly the spanwise hot streak migration towards the hub. This might to some extent be attributed to the buoyancy effect as discussed by Shang and Epstein [5]. However, this spanwise inward migration can hardly be traced in the corresponding contours (not shown here) for the case with hot streaks located at the NGV mid-passages. Therefore the buoyancy mechanism, which should be largely independent of the NGV clocking, is not judged to be the main contributor to the noticeable hot streak movement towards the hub. The rotor passage secondary flow seems to be the key. Also shown in Fig. 10 is a close-up of one passage superimposed with the instantaneous secondary flow velocity vectors. A strong passage-vortex near the hub is clearly visible. The corresponding secondary flow velocity component will convect the mid-span hot fluid on the pressure surface towards the hub. Furthermore, the passage vortex would actually convect the hot fluid from the hub-pressure surface corner towards the suction surface, as the shape of the high temperature contours around the corner indicates.

It is also useful to note that when hot streaks impinge on the NGV blades, they are in phase with the NGV wakes. Hence the hot areas seen on the contour plot (Fig. 10) are also the instantaneous signature of the NGV wakes. The cross-passage movement due to the NGV wakes is in the same sense as that in the hub region caused by the passage vortex. This phasing between the NGV wake and rotor passage vortex would enhance the cross-passage movement from the pressure surface to the suction surface near the hub.

Overall the cases with 32 hot streaks show a strong dependence on the clocking between the hot-streaks and the NGV blades. The key seems to be the phasing among several fluid kinematic features with both temporal and spatial length-scales being the NGV blade passage. The results for unsteady surface pressures and blade forces as presented later will also show that for this case, the hot streaks have a relatively small influence on the rotor passage flow in a dynamic sense.
Rotor Surface Temperatures (8 Hot-Streaks)

When the number of hot streaks is reduced to 8, the circumferential wavelength is increased by a factor of 4. Again two hot streak-NGV clocking positions are considered. The case with hot streaks impinging on the NGV blades is as shown in Fig. 6a. Similar to the 32 hot streaks case, a ‘mid-passage hot streak’ situation is achieved by circumferentially shifting the temperature profile by half a NGV blade pitch. The calculated time-averaged temperatures on the rotor blade pressure surface at 10%, 50% and 90% span sections are shown in Fig. 11.

The results for this long wave length case are qualitatively different from those for 32 hot streaks. The time-averaged surface temperatures are almost identical, showing no dependence on the hot streak-NGV clocking position. Looking at the unsteady temperatures in terms of the maximum and minimum values, we also see only very small differences between the two hot streak clocking positions (Fig. 12).

The insensitivity of rotor surface temperatures to the hot streak clocking for the 8 hot streak case is directly attributed to the difference in length scales between the hot streaks and the NGV blade passage. Basically the rotor is now subject to two temporal disturbances, a) the aerodynamic disturbance from the NGV, and b) the NGV modulated hot streaks. For the 32 hot streak cases, both disturbances have the same temporal and spatial frequencies. As such the phasing between the two in terms of the corresponding kinematics can either enhance or reduce the cross-passage movement and migration. But for the 8 hot streak case, a phasing between two disturbances at different frequencies no longer matters.

A further comparison is made between the 8 hot streak case and that with 32 hot streaks for the clocking position when the hot streak is placed at the NGV mid-passage. The time-averaged temperatures are given in Fig 13. The maximum and minimum unsteady temperatures are shown in Fig. 14. Because the temperature field of a hot streak can no longer be phased with the NGV velocity field, the 8 hot streak case shows a noticeably reduced preferential heating effect. The time-averaged temperature is about 5% lower than that of 32 hot streaks for most of the pressure surface (Fig. 13).
On the other hand, the unsteady temperature variation for the 8 hot-streak case is roughly by a factor of 3 larger than the 32 hot-streak case (Fig. 14), although both cases have the same inlet temperature distortion magnitude (OTDF and RTDF). The longer residence time for the long wavelength case with 8 hot streaks generates much larger unsteady response. This behaviour is similar to that of a rotor pressure field under an influence of an inlet stagnation pressure distortion. It is also noted that the maximum unsteady temperature for the 8 hot streak case is about 8-10% higher than that with 32 hot streaks (Fig. 14). The maximum temperature should be relevant to blade thermal fatigue life. This marked influence of the circumferential wavelength/number of hot streaks on the rotor blade heat load is certainly relevant as realistic industrial gas turbines typically have 6 or 8 combustion burners. Hence the clocking characteristics gathered from a configuration, where the number of hot streak is taken to be the same as that of NGV blades, may not be applicable to situations with fewer hot streaks.

**Rotor Blade Forcing**

Given that the rotor incidence changes with hot streak, the associated unsteady loading needs to be examined for aeromechanical design considerations.

The spectra of unsteady tangential forces on rotor blades are shown in Fig. 15 for the cases calculated. For the case with 32 hot streaks, the frequency of the hot streaks is the same as the NGV blade passing frequency (i.e. 32 events per revolution). The influence of the hot streaks on unsteady forcing can thus be qualitatively measured by comparing the force amplitudes (Fig.15b) with that of a uniform inlet (Fig.15a). The results show that the extra unsteady forcing contributed by the hot streaks is very small, in a region of around 1% of the time-averaged force. The differences in higher harmonics of the fundamental hot streak/blade passing frequency are also proportionally small. This is in line with the most of the previous researches showing relatively small hot streak influence on the rotor pressure field. The secondary flow velocity field such as that shown in Fig. 10 indicates no detectable difference with or without the inlet temperature distortions. The corresponding temperature field is largely driven passively by the passage velocity field in both NGV and rotor rows.

However, we have a very different picture for the case with 8 hot streaks. The unsteady forcing is indicated by the peak at a frequency of 8 per revolution (Fig.15c). Now the amplitude is about 8% (~16 % peak-to-peak) of the time-averaged tangential force. It is also of interest to note that apart from the two primary disturbances (the hot streak at 8 per rev and the NGV at 32 per rev) and their higher harmonics, the sub-harmonics due to the cross-coupling/nonlinear interaction between the two primary disturbances (i.e. components at frequencies of mωNGV ± nωhot streak for any integers m and n) also show up on the spectrum (e.g. the component at 24 per rev). Nevertheless, the relatively small magnitudes of the sub-harmonics suggest that the unsteady flow responses are still largely linear.

Finally, it should be commented that the calculated rotor blade forcing shows little dependency on the hot streak/NGV clocking for 8 hot streak cases. This is similar to the temperatures (Fig. 11 & 12). For the case with 32 hot streaks on the other hand, the clocking effect on the unsteady forcing is also shown to be insignificant, simply because the magnitude of the forcing component attributed to the hot streak with a short wavelength is very small.

**CONCLUDING REMARKS**

In the present work, steady and unsteady calculations for a transonic turbine stage at a uniform inlet condition are firstly carried out. The results are in reasonable agreement with the experimental data for validation purposes. Further analysis with different hot-streak circumferential length-scales reveals significant influence on both rotor blade forcing and heat load.

When the number of hot streaks is the same as that of NGV blades, the rotor blade heat load is strongly dependent on the hot streak-NGV clocking, giving a maximum difference of 8% in time-averaged temperature on the rotor pressure surface. The unsteady pressure and force on rotor blades, however, are largely unaffected by the hot streaks. The temperature migration inside the rotor row is dictated by the phasing among
several cross-passage motions associated with NGV wakes (negative jet), hot streak heating (positive jet), NGV and rotor passage secondary flows.

For the case with 8 hot streaks, the hot-streak/NGV clocking is shown to have very little effect on rotor surface temperature distributions. The time-averaged temperature on the pressure surface is around 5% lower than that for 32 hot streaks located at the NGV mid-passage. However, the unsteady temperature fluctuation is much larger, with the maximum temperature being 8-10% higher than that for 32 hot streaks. Regarding blade aeromechanics, the unsteady forcing on the rotor blades due to the hot-streaks is at least 5 times higher than that with 32 hot streaks. The marked differences in both unsteady forcing and surface temperature between the two cases imply that the number of combustor/burners might be used as a design variable for HP turbine blade aeromechanics as well as heat transfer.

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