Numerical Simulation of a Turbulent Flow Over a Backward Facing Step With Heated Wall—Effect of Pulsating Velocity and Oscillating Wall

An accurate prediction of the flow and the thermal boundary layer is required to properly simulate gas to wall heat transfer in a turbulent flow. This is studied with a view to application to gas turbine combustors. A typical gas turbine combustion chamber flow presents similarities with the well-studied case of turbulent flow over a backward facing step, especially in the near-wall regions where the heat transfer phenomena take place. However, the combustion flow in a gas turbine engine is often of a dynamic nature and enclosed by a vibrating liner. Therefore apart from steady state situations, cases with an oscillatory inlet flow and vibrating walls are investigated. Results of steady state and transient calculations for the flow field, friction coefficient, and heat transfer coefficient, with the use of various turbulence models, are compared with literature data. It has been observed that the variations in the excitation frequency of the inlet flow and wall vibrations have an influence on the instantaneous heat transfer coefficient profile. However, significant effect on the time mean value and position of the heat transfer peak is only visible for the inlet velocity profile fluctuations with frequency approximately equal to the turbulence bursting frequency. [DOI: 10.1115/1.4007278]

Keywords: backward facing step, pulsating flow, oscillating wall, skin friction coefficient, heat transfer, Stanton number

Introduction

In this paper, the transient and oscillatory flow in gas turbine combustor geometry is investigated in more details and in isolation of other model problems like combustion, swirling flow, and acoustics. To this end, the combustor is reduced to its most elementary geometry, namely a backward facing step. The performance of a gas turbine depends critically on the heat transfer between the hot combustible flow and the liner [1]. An important design characteristic of the combustor is the flow of a mixture of air and fuel entering at high axial and tangential velocity and small radius the combuster and expanding to larger radii. This determines the flame stabilization and results in a short flame length. A major complication here in that the hot combusting flow is forced outward due to centripetal forces and is directed toward the liner wall. Hence, the hot gases flow at high velocity and with a thin and developing boundary layer past the liner wall, inducing a high rate of heat transfer and heating the liner to the temperatures of 800°C or more. The equilibrium liner temperature depends on the balance between heat loss to the cooling air flow at the cold outside wall of the liner and the heat input at the hot inside wall of the liner. To complicate the situation, the hot side heat transfer can be enhanced by a transient and oscillating boundary layer due to spontaneous oscillations of the mainstream hot flow. These can be caused by coupled flame to burner acousto/aero dynamic feedback processes, as presented in Ref. [2].

The flow over the backward facing step with heat transfer is well documented with experimental data for different geometry configuration, e.g., Refs. [3–8]. Experimental data of Vogel and Eaton [3] are used for validation. The gas to wall heat transfer in a turbulent flow over the step, with specific attention to the accurate prediction of the flow, thermal boundary layer with resulting wall friction coefficient and heat transfer coefficient is studied. The application of the research can be extended to any flow over sudden-expansion channel with heat transfer, where the pulsating flow conditions and/or vibrating walls may affect the flow characteristics.

The characteristic attribute of the flow over a backward facing step is a separation of the boundary layer at the edge of the step. Behind the step, as an effect of an adverse pressure gradient, a primary recirculation region is formed, see Refs. [9–14]. The length of this region is specified by factors related to the flow properties as well as the geometrical dimensions of the sudden-expansion channel, as shown in Refs. [15] and [16]. Typically for the backward facing step, the maximum heat transfer coefficient is observed within the recirculation region close to the reattachment point [3,17]. The position and value of the peak in the heat transfer coefficient are correlated in stationary flows to the position and value of the skin friction coefficient. In a typical combustion chamber of a gas turbine engine, flow of reacting gases is not steady but affected by wall vibration and acoustic pulsation. Because of that, the overall characteristics of the separated and reattached flow together with the position and magnitude of maximum heat transfer may be altered significantly. Several studies of the effect of fluctuating velocity and aspect ratio on the rate of heat transfer were conducted [15–18]. They have shown that more has to be done to reveal the heat transfer mechanisms in such complicated flow structures.

To investigate the influence of the steady state and dynamic flow parameters on the variations in the heat transfer and wall friction coefficient, stationary and transient calculations are performed. The stationary solutions are validated with experimental
data. For stationary calculations, the standard k-ε, k-ω, and shear stress transport (SST) models [19,20] as available in the Ansys CFX code are used. The transient calculations of pulsating inlet velocity and oscillating wall are performed with the SST turbulence model. This model has shown the best agreement with the experimental results in stationary flow situation. The influence of a pulsating inlet velocity and oscillating moving wall on the recirculation region and the rate of heat transfer coefficient are explored. The axial inlet velocity is fluctuated in the frequency range of 5–1000 Hz. The behavior of a stationary flow over an oscillating moving wall is explored at a wall vibration frequency of 10 Hz.

The presentation of the results is done on the basis of dimensionless numbers, i.e., the Stanton number: $St = \frac{h}{\frac{1}{2} \rho U^2}$ and skin friction coefficient: $C_f = \frac{f U}{\frac{1}{2} \rho U^2}$, which are defined in terms of the heat transfer coefficient ($h$) and wall shear stress ($\tau_w$), respectively. For steady and developed flow, a correlation between the Stanton number and the skin friction coefficient exists [21] which might not hold for unsteady flow. Thus in unsteady flows, both factors are of interest.

### Numerical Model

Due to simplicity, robustness, and wide industrial applications, the RANS/URANS model is used for computations. Two-equation family models: k-ε and k-ω and SST are considered. Models constants are presented in Table 1. The well-known k-ε and k-ω models are described in details in Refs. [22] and [23], respectively. In order to gather the best from the k-ε and the k-ω models, a blended model called SST was developed, see Ref. [24]. The SST model calculates the flow in the near-wall region using a k-ω formulation whereas in the bulk flow the high Reynolds k-ε formulation is applied. A smooth transition between the two formulations is ensured by the use of additional blending factors, which are functions of the wall distance. The implementation of the SST model permits the automatic shift from the low-Re number formulation to the wall function scheme according to the grid resolution [25]. Since the velocity of the flow is relatively small (<0.05 Ma), the effect of the mean flow kinetic energy is not included later in the heat transfer model (thermal energy model), for details see Ref. [25].

### Computational Domain

The investigated in the literature test rig [3] is a two-dimensional, sudden-expansion wind tunnel. To reproduce the experimental conditions, the sudden-expansion channel, as depicted in Fig. 1, is used for the numerical calculations. Air enters the computational domain upstream of the step. Across the bottom wall, downstream of the step, a constant heat flux of 270 W/m² is prescribed. The top wall is considered adiabatic and the grid resolution [25]. Since the velocity of the flow is relatively small (<0.05 Ma), the effect of the mean flow kinetic energy is not included later in the heat transfer model (thermal energy model), for details see Ref. [25].

![Fig. 1 Geometry, coordinate system, and flow pattern for the backward-facing step calculations](image)

According to the experiment of Ref. [3], the geometrical dimensions, based on the step height (H), are the following: The computational domain length is equal to 40 H, the height upstream of the step is equal to 4 H, and downstream to 5 H for an expansion ratio $ER = \frac{W}{H}$ of 1.25. The value of $H$ is taken as 3.8 cm.

The Reynolds number based on the step height (defined as $Re_\theta = \frac{U H}{\nu}$) during the steady state and oscillating wall calculations is fixed to 28,000 and for transient computations with pulsating velocity is varied from 23,000 to 33,000.

The flow field behind the backward facing step may have three-dimensional vortex structure as presented, e.g., by Ref. [27]. Therefore, to validate the assumption that 2D numerical domain represents well the experiment, 3D backward facing step geometry is studied first. The 3D model with 2,455,000 elements is used along the centerline for velocity profiles. These data serve also for the grid independency study. Different grid resolutions are investigated. Most of the elements are placed in the region close to the step and the heated wall. The near-wall region is created with the use of prism elements in order to avoid generating highly distorted tetrahedral elements next to the wall. The maximum value of $y+$ at the bottom heated wall is equal to 2 with average value of 1.8. The Ansys CFX-10/11 code always operates in a three-dimensional domain [25]. Thus, 2D numerical models have a third dimension in a span-wise direction with a thickness equal to the size of one numerical element and periodic boundary conditions are applied. 2D or 3D simulations where the span-wise direction is treated as statistically homogenous with periodic conditions prescribed at span-wise boundaries of the backward facing step is common way to model the numerical problem, see Refs. [12], [33], and [34]. In Fig. 2, results of grid independency study for chosen grids (2D and 3D) are presented. Only minor differences are observed in the velocity profiles. Similarly, the point of the flow reattachment is predicted nearly at the same location (differences are below 4%) by all models. Keeping in mind that in this study transient calculations with pulsating inlet velocity and oscillating walls are desired, 2D numerical domain with 230,000 elements is applied for further research.

### Stationary Flow

Before the transient computations are performed, the steady state numerical results with different turbulence models are validated with experimental data. Thereafter, the turbulence model presenting the best agreement with the experiment is selected for transient computations. This way the uncertainty in the prediction of the skin friction and heat transfer coefficient behavior under pulsating flow and oscillating wall conditions is minimized.

All turbulence models show similar general behavior of the flow over the backward facing step. The skin friction coefficient is

### Table 1 Models constants

<table>
<thead>
<tr>
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<th>k-ε</th>
<th>k-ω</th>
<th>SST</th>
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<td>$C_{f1}$</td>
<td>1.44</td>
<td>$\alpha$</td>
<td>5/9</td>
</tr>
<tr>
<td>$C_{f2}$</td>
<td>1.92</td>
<td>$\beta$</td>
<td>0.075</td>
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<td>1.30</td>
<td>$\sigma_{f2}$</td>
<td>2.00</td>
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predicted negative in the recirculation zone and positive downstream of the channel, as shown in Fig. 3. The reattachment point is defined as the axial position where the skin friction coefficient changes from negative to positive values, see Refs. [19] and [28].

Downstream of the reattachment point, a recovery zone with monotonically increasing skin friction coefficient is observed. The presence of a secondary recirculation bubble close to the step corner is also visible. The reattachment length is predicted with 15% error (at position $x/H = 6.74$ and $6.82$). The position of the maximum negative skin friction coefficient is also predicted correctly. More significant errors are observed for the $k$-$\omega$ model. The secondary recirculation bubble which appears close to the step corner is hardly visible.

The reattachment length is predicted with 15% error (at position $x/H = 5.66$) with respect to experimental data. The error magnitude for $k$-$\omega$ calculations is consistent with the results reported in the literature, e.g., Ref. [35]. All investigated turbulence models significantly underestimate the skin friction coefficient downstream, in the recovery region, where the overall error is much higher. This behavior is observed also in the work of other researchers [19,29].

The behavior of the Stanton number is generally opposite to the skin friction coefficient, e.g., Refs. [3] and [29]. The investigated models predicted the same, typical behavior. A sudden drop of the heat transfer coefficient just downstream of the step is followed by a sharp increase till a maximum value near the reattachment point, followed by a monotonic decrease further downstream in the recovery region. Both, the $k$-$\omega$ and SST turbulence models predicted well the position of the maximal Stanton number peak, as illustrated in Fig. 3. The SST model also predicted correctly the peak magnitude, whereas the other models gave a significant underestimation. For the $k$-$\omega$ model, the Stanton number profile is shifted upstream, similar to the skin friction coefficient results.

In stationary developed flow, the heat transfer coefficient can be correlated to the skin friction coefficient and the level of turbulence intensity. This correlation, called the Chilton and Colburn J-factor analogy, is defined as $\frac{C_\text{St}}{C_\text{f}} = 2 \cdot \frac{Pr}{Re^{1/3}}$ and it is an extension of the Reynolds analogy (known as $C_\text{St} = 2$), taking into account that the Prandtl number is not equal to unity in most of the cases. For air at experimental conditions, the Prandtl number is approximately equal to 0.7; thus, the analogy should be preserved at the ratio of skin friction coefficient to Stanton number equal to 1.58. The correlation does not hold in the recirculation region as the skin friction coefficient changes sign at the edge of the recirculation bubble. At large distance downstream from the backward facing step ($x/H > 30$), it is expected that the flow has developed to the pattern of channel flow and the correlation of Chilton and Colburn to be applicable. From Fig. 3, it can be read that the friction coefficient is predicted below the Stanton number. The poor prediction of the skin friction coefficient in the recovery region by all turbulence models makes the analogy not applicable to the numerical solutions. Similar results are obtained by Ref. [29] during the large eddy simulation calculations of the backward facing step with heat flux through the bottom wall. In that study, for the heat flux of 1 kW/m² the Chilton and Colburn analogy is not satisfied, whereas for 2 kW/m² and 3 kW/m², the analogy is fulfilled and even overpredicted, respectively. Since at the location of measurements, i.e., $x/H = 20$, the Stanton number for all three heat fluxes has the same value, see Ref. [29], the main impact on the analogy comes from the numerical prediction of the skin friction coefficient. The results suggest that the skin friction coefficient recovers faster to the flat-plate behaviour downstream of the reattachment region for higher heat flux. The experimental results of Ref. [3] have confirmed the Chilton and Colburn analogy. At the location of the most downstream experimental data, i.e., $x/H = 22$, see Fig. 3, the ratio $\frac{C_\text{St}}{C_\text{f}} = 1.16$ and the

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**Fig. 3** Comparison of the skin friction coefficient and the Stanton number for the $k$-$\omega$, $k$-$\omega$, and SST turbulence model

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trend lines of the skin friction coefficient and Stanton number develop to fulfill the analogy somewhere downstream of the channel. A comparison of the temperature, velocity, and turbulence intensity profiles at positions selected according to Ref. [3] is presented in Figs. 4-6, respectively. The temperature gradient is often well predicted ($x/X_r = 0.25, 0.65, 1.05$) but occasionally underpredicted ($x/X_r = 0.05, 1.45, 2.25$), see Fig. 4. All turbulence models behave similarly. The temperature profiles show a steep temperature gradient in the near-wall region. The exception is the location of a temperature profile near the step, i.e., at location $x/X_r = 0.05$. The presence of the secondary bubble and the small axial velocity at this region changes considerably the temperature gradient across the domain. All computed temperature trends have more significant errors in the near-wall region than far away from the wall. Surprisingly, the k-ε turbulence model predicted the temperature trend at the location close to the step better than the other turbulence models. This is most likely due to the significantly underpredicted length of the recirculation region with higher temperature gradient in comparison to other models.

The velocity field (presented in Fig. 5) shows good agreement with the experimental data. There is no significant difference with the experimental data. There is no significant difference
observed between the various used turbulence models. All three, 
\(k-e\), \(k-x\), and SST, captured well the changes in the velocity at 
different locations, including the reversal flow near the step.

Downstream of the reattachment point, the flow at the wall has a 
downstream direction and a boundary layer develops.

In case of the turbulence intensity profiles, a minor overpredic-
tion of the experimental results can be observed, especially in the 
neighborhood of the heated wall, see Fig. 6. The turbulence levels 
are very low in the main flow area \(Y/H > 1\). In the shear layer 
between the main flow and the wall, \(Y/H < 1\), turbulence is 
produced and its levels peak approximately at \(u/U_0 \approx 0.17\) for 
\(Y/H \approx 0.5\). The \(k-e\) model again presents the worst agreement 
with the literature data, whereas the \(k-x\) and SST show comparable 
behavior, with a minor advantage of the latter near the reattach-
ment point.

**Transient Flow—Inlet Velocity Pulsation**

In this section, transient calculations performed with a pulsating 
inlet velocity are presented. The turbulence model which gave the 
best prediction in the steady state configuration, i.e., SST, is used.

In order to investigate the influence of the pulsating flow on the 
recirculation region and the heat transfer coefficient, several 
calculations are performed at different forcing frequencies of the 
velocity profile. Since the gas turbines suffer mostly from instabil-
ities in frequency range below 1 kHz, the investigated here fre-
quencies are equal to 5, 10, 50, 100, 400, and 1000 Hz. The 
corresponding Strouhal number, defined as \(Sr = fH/U_0\) (to avoid con-
fusion between Strouhal and Stanton number, the former is 
denoted as \(Sr\)), ranges from 0.017 for the 5 Hz excitation to 3.5 
for the excitation equal to 1000 Hz. The forcing amplitude of pul-
sation is fixed to 20% of the mean velocity profile according to 
\(u = u_0(1 + 0.2 \sin(2\pi t))\). The Reynolds number based on the 
step height is varied in the range of 22,000–33,000. Other param-
eters are left unchanged. The calculation time step is adjusted 
depending on the investigated frequency. For each study, 20 sam-

For all frequencies, the instantaneous minimum value of the skin 
friction coefficient is several times lower than in the steady state 
calculations and its axial position varies according to the changes
In the recirculation region, see Fig. 8 for 10 Hz pulsation frequency.

In the 10 Hz frequency case particularly, the mean profile differs significantly from the steady state results. The mean value of the maximum negative peak of the skin friction coefficient is about two times higher and shifted upstream of the domain in comparison to the steady state profile, as depicted in Fig. 8. Thus, the mean recirculation area is also shorter. The secondary bubble formed near the step corner with positive axial velocity is clearly visible. In the recovery region, the mean skin friction coefficient profile presents similar behavior as in the steady state case. This behavior of the mean skin friction coefficient profile can be explained as a result of significant changes in the instantaneous velocity profiles. During velocity fluctuations, the secondary bubble increases together with the decreasing of the inlet velocity till its maximal length. At this point, the velocity starts to increase again. As a result of the time delay between changes in the inlet velocity profile and response of the recirculation region and the secondary bubble, the recirculation zone is broken up in two separate parts, as shown in Fig. 7. Thereafter, the part close to the step behaves like the main recirculation region and it grows to the size of the original recirculation area. Inside this new region, the secondary bubble is formed again. The part far from the step disappears with time advancing and a whole cycle is repeated yet again. The increase of the size of secondary bubble is also well visible in Ref. [30], but the pulsation is done through small slit in the corner of the step. It is not possible to specify exactly the instantaneous reattachment point of the 10 Hz excitation cycle, due to simultaneous presence of two recirculation bubbles. A similar behavior was observed in low frequency structures in a direct numerical simulation of the flow over the backward facing step without heat transfer by Ref. [28]. The cases of 5 Hz and 50 Hz represent similar transient behavior; however, the effect on the mean skin friction profile is minor. For frequencies of 100 Hz and above, the breaking up of the recirculation region is not observed. The instantaneous changes in the magnitude and location of the maximum skin friction coefficient do not affect significantly the mean value. The mean maximum peak in the skin friction coefficient has almost the same magnitude and position as the one in the steady state calculations, see Fig. 9.

For the 10 Hz excitation, the transient changes in the heat transfer coefficient are significant. The maximum peak in the Stanton number corresponds exactly to the maximum peak of the skin friction coefficient. Due to the recirculation bubble phenomena described above, the maximum peaks in skin friction and heat transfer are moving up and down, backward and forward, depending on the recirculation zone position, as presented in Fig. 10. As a consequence, the mean Stanton number profile for the 10 Hz excitation frequency is significantly changed. The maximum peak is higher and shifted upstream. The fast and transient changes in the velocity field at frequencies of 5 Hz and above 50 Hz affect only a little the mean temperature field near the wall, see Fig. 11. The velocity field at frequencies of 5 Hz and above 50 Hz affect only a little the mean temperature field near the wall, see Fig. 11. Hence, the mean values of the Stanton number profiles agree well with steady state numerical results for high frequency.

The enhancement of the mean heat transfer coefficient and skin friction coefficient at a pulsation frequency of 10 Hz can be related to the turbulent bursting frequency. The bursting frequency is a characteristic frequency at which the viscous sublayer is renewed. The increase and destruction of the viscous sublayer is a periodical process and matching it with the frequency of flow pulsation results in resonance behavior. This in consequence may increase the heat transfer process. For a pipe flow, increase or decrease of the heat transfer coefficient depending on the excitation frequency and Reynolds number was reported, see Refs. [36] and [37]. The match between the frequency of pulsation and bursting frequency, defined for pipes as $f_D = n_{burst}/5D$ [36,38], may result in peak heat transfer coefficient in pipe flows at pulsation frequency of 10 Hz.
in resonance interaction and lead to enhancement of the heat transfer coefficient, as presented in Ref. [36]. In the investigated case the turbulent bursting frequency, based on the mean inlet velocity, is in the frequency range of 11–14 Hz (depending on the downstream or upstream geometry, respectively), which is nearby the pulsation frequency of 10 Hz. For the Strouhal number in the range of 0.35–3.5 (100–1000 Hz), the frequency of pulsation is always at least one order of magnitude higher than the bursting frequency; therefore, no significant effect on the heat transfer is observed.

**Transient Flow—Oscillating Wall**

The transient calculations of the flow over the backward facing step with the oscillating bottom wall are performed for the same domain configuration as for investigated earlier cases. The inlet velocity is undisturbed. Instead, the heated bottom wall is oscillating with a frequency equal to 10 Hz (piston-like movement). The frequency is chosen based on phenomena observed for 10 Hz inlet velocity pulsation. The maximum amplitude of the vertical displacement is equal to 2 mm according to $\delta = 0.02 \sin(2\pi tf)$. This leads to a variation of the expansion ratio, based on the geometrical dimensions of 1.24–1.26. The time step is set to 5 ms and 20 samples per cycle are collected. Other flow and geometrical properties are left unchanged. For calculations with moving wall, a model with a moving grid is used, see Ref. [25]. The total number of elements is kept constant. The region near the heated wall is made with the use of prism elements; thus, any changes in the domain during wall movement have only minor influence on the element distortion.

For the computations with moving bottom wall, the governing equations are modified since the control volume depends on time. An new term $\nabla \cdot (\Gamma_{disp} \nabla \delta)$ is introduced to represent the velocity of the control volume boundary, see Ref. [25]. The regions of nodes with the same degrees of freedom are determined by the mesh stiffness, see

$$\nabla \cdot (\Gamma_{disp} \nabla \delta) = 0$$  \hspace{1cm} (1)

Since in this work, an uniform mesh stiffness is applied, the displacement is homogeneously spread through the mesh and the mesh topology is preserved.

The instantaneous changes in the position of the bottom wall have a significant effect on the reattachment region. The length of the recirculation zone varies according to the changes in the step height, as shown in Fig. 12. When the heated wall is moving upward and the step height is decreasing, the reattachment point is moving downstream of the domain. When the step height is increasing, the reattachment region is decreasing. Opposite behavior was observed for dimensionless reattachment length by Ref. [39] in the Reynolds number range from 545 to 745 and Ref. [40] for $Re < 400$ (based on the step height). However, research of Ref. [16] for backward facing step with heat transfer for $Re_h$ of 26,000–92,000 and Ref. [41] for $Re_h$ of 8000–32,000 show that larger step height to inlet channel height ratio results in faster grow of unstable shear layer and higher turbulence intensities. In consequence, the normalized reattachment length is shorter. This confirms observations made in the present study for $Re_h = 28,000$.

The instantaneous minimum in the skin friction coefficient is about twice as low as the steady state value. The minimum in the skin friction coefficient is moving downstream and upstream depending on the transient reattachment region length. However, the changes are not as significant as in the case of the pulsating inlet velocity. The secondary bubble is not growing to the dimensions, which could threaten the breaking up of the recirculation region. The mean velocity profile is not much different from the steady state, only the minimum mean skin friction coefficient peak is lower than the one observed during steady state calculations. The mean length of the recirculation region is also the same. The behavior of the mean skin friction coefficient profile in the recovery region matches the steady results.

The Stanton number follows the changes in the skin friction coefficient, as illustrated in Fig. 13. Because of the low frequency of excitation, the heat transfer is affected by changes in the recirculation length. The instantaneous, maximum peak in the Stanton number is located at the position, which corresponds to the transient location of the maximum peak of the skin friction coefficient. However, the general increasing or decreasing tendency of the heat transfer coefficient with respect to skin friction coefficient is affected by a time delay equal approximately to 30 ms. These changes are not high enough to change significantly the mean Stanton number profile. Only minor changes in the vicinity of the maximum peak of Stanton number are observed. At this position, the mean value of the transient Stanton number is slightly bigger with respect to the steady state profile. It can be concluded that despite the high amplitude of the wall vibration, the effect on the mean skin friction coefficient and heat transfer for 10 Hz oscillation frequency is minor.

**Conclusions**

For steady inlet flow conditions, the $k-\omega$ and SST turbulence models present good agreement with the experimental data of the flow over backward-facing step with heated wall. The reattachment length is predicted with error below 3%. The $k-\varepsilon$ model, however, revealed a 15% underprediction of the reattachment point. In the recovery region, the skin friction coefficient is significantly underestimated by all turbulence models, leading to failure of the Chilton and Colburn J-factor analogy.

The effect of a pulsating inlet velocity is especially important at the excitation frequency of 10 Hz. In this case, the secondary recirculation bubble formed in the corner of the step is broken up into two parts. This behavior is also observed for frequency of 5
and 50 Hz; however, for these frequencies the mean skin friction coefficient and Stanton number profiles are almost the same as the one obtained during the steady flow computations. For all investigated frequencies, the instantaneous minimum of the skin friction coefficient is several times lower than in the steady state calculations. In line with this, the maximum Stanton number is much higher, suggesting an increase of a risk of wall failure due to exposition to high temperature oscillations.

The variations in the pulsation frequency affected the transient values of both the skin friction coefficient and the Stanton number, but the time mean values are left almost unchanged for frequencies above 100 Hz. Only in the case of the 10 Hz excitation frequency, changes are observed. This is in agreement with the turbulence bursting frequency effect. At this frequency, the transient effect is maximal.

The oscillatory changes in the wall position affected the instantaneous values of the skin friction coefficient and the Stanton number. A certain delay of the peak position between instantaneous profiles of the skin friction coefficient and the Stanton number is observed. However, the influence of those changes on the position of the main peaks as well as on the main profiles is not significant.

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Nomenclature

\begin{enumerate}
\item \textbf{Cl} = Skin friction coefficient
\item \textbf{Cp} = Specific heat at constant pressure
\item \textbf{D} = Diameter
\item \textbf{H} = Step height
\item \textbf{Pr} = Prandtl number
\item \textbf{Re} = Reynolds number
\item \textbf{Sc} = Schmidt number
\item \textbf{Sr} = Strouhal number
\item \textbf{St} = Stanton number
\item \textbf{T} = Temperature
\item \textbf{U} = Velocity
\item \textbf{V} = Volume
\item \textbf{W} = Channel width
\item \textbf{Xr} = Reattachment length
\item \textbf{f} = Frequency
\item \textbf{f_c} = Bursting frequency
\item \textbf{h} = Heat transfer coefficient
\item \textbf{k} = Turbulent kinetic energy
\item \textbf{q_w} = Heat flux
\item \textbf{x}, \textbf{y}, \textbf{z} = Directions
\item \textbf{\Gamma_{disp}} = Mesh stiffness
\item \textbf{\delta} = Displacement
\item \textbf{\epsilon} = Turbulent dissipation rate
\item \textbf{\nu} = Kinematic viscosity
\item \textbf{\rho} = Density
\item \textbf{\theta_{m}} = Boundary layer thickness
\item \textbf{\tau_w} = Wall shear stress
\item \textbf{\omega} = Turbulent frequency
\end{enumerate}

References


