Low-Order Analysis of Conjugate Heat Transfer in Pulsating Flow with Fluctuating Temperature

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Abstract. Realizing enhanced heat transfer in any kind of thermodynamic process directly improves its efficiency or reduces its size. Therefore it is of fundamental interest that in periodic flows enhanced heat transfer has been reported repeatedly. Because of the huge number of different parameters in turbulent pulsating flows, e.g. perturbation amplitudes, flow regimes, it is difficult to identify relevant effects. To develop a more global insight into the dynamics of conjugate heat transfer in pulsating flows, a transient lumped model is formulated and analyzed in this study. It is derived from a one dimensional analysis of the conjugate heat transfer from the bulk of a pulsating flow through a wall of finite thickness. All hydrodynamic mechanisms of the pulsating flow are modeled through an harmonically oscillating heat transfer coefficient. Additionally an harmonically oscillating bulk temperature is considered. Two wall configurations of increasing complexity are analyzed: Firstly a lumped capacity model to scrutinize the effect of a dynamic behavior of the wall temperature. Finally, the most general case of a one dimensional finite volume discretization of the wall is presented and the dynamic of the wall temperature itself is examined. The results show that enhanced time averaged heat transfer is possible.

1. Introduction
Enhanced heat transfer in pulsating flows has been reported repeatedly, especially in devices exhibiting self-excited combustion oscillations. For example, Dec et al. [1] measured wall heat fluxes that locally exceeded expected values by up to 250% in the tailpipe of a pulse combustor. Perry and Culick observed an increased average heat transfer rate in a solid propellants rocket thrust chamber with large amplitude oscillations [2]. Even though explanations for this behavior have been suggested in the literature, the responsible mechanisms were not identified with certainty, in some cases contradictory predictions were made. The turbulent length scales present in these setups, introducing several flow regimes, increase the complexity of the problem considerably. Especially because a wide range of perturbation amplitudes and mean flow Reynolds numbers have been considered in previous investigations, it is difficult to separate pertinent effects and compare the various conclusions.

In the previously mentioned devices, the velocity fluctuations are induced by thermoacoustic instabilities. It can be thus expected that temperature fluctuations additionally occur. The impact of these temperature fluctuations on the wall heat flux in laminar pulsating pipe flow have been investigated by Marksten et al. [3]. They showed that considerably enhanced or decreased heat transfer can be achieved, depending on the relative phase lag between the oscillations.
However, their study is restricted to cases with ideal wall boundary conditions, where wall thermal effects can be neglected.

Other studies of laminar pulsating flow have demonstrated that both enhanced as well as decreased heat transfer are possible in pulsating flows. In an experimental study, Habib et al. [4] investigated the heat transfer in laminar pulsating pipe flow. In the thermally developing region, they found a spatial modulation of the local Nusselt number along the tube, with both higher and lower values compared to the stationary case. However, the investigations were restricted to flow without reversal. Hemida et al. [5] performed a theoretical study of the heat transfer characteristics of laminar pulsating pipe flow. They confirmed the existence of locally enhanced or decreased Nusselt numbers. Again, the effects of pulsation on heat transfer are mostly present in the thermally developing region. Additionally, they investigated the impact of thermal resistance and thermal inertia of the wall assuming an homogeneous temperature distribution in the wall normal direction (lumped capacity).

In a more general approach, Zudin [6] also treated the conjugate heat transfer problem analytically for the two dimensional case with spatio-temporal fluctuations of the heat transfer coefficient. Thus, the hydrodynamic effects causing the convective heat transfer are modeled through a heat transfer coefficient applied as a boundary condition of the problem. He concludes that under these conditions, overall heat transfer through the wall should decrease. However, the study is restricted to fluctuations of heat transfer coefficient with constant bulk temperatures.

To develop a more global insight into the system dynamics of heat transfer in pulsating flows, a transient low order model taking wall thermal effects into account is formulated and analyzed in this study. It is derived from a one dimensional analysis of the conjugate heat transfer from the core of a pulsating flow through a wall of finite thickness. Similar to the studies of Zudin, all hydrodynamic mechanisms of the pulsating flow are modeled through an harmonically oscillating heat transfer coefficient. In addition, fluctuations of the bulk temperature can be imposed. Two non-dimensional numbers are identified, i.e. the Biot- and the Fourier-Number. Depending on the fluid and wall properties, geometry and forcing frequency, two configurations of increasing complexity are analyzed: Firstly a lumped capacity model to scrutinize the effect of a dynamic behavior of the wall temperature. Secondly, the most general case of a one dimensional finite volume discretization of the wall is derived and the dynamic of the wall temperature itself is examined. Due to the inertia of the wall, the solution of the problem for the last two cases has been approximated asymptotically in the low and the high frequency regime, respectively.

2. Domain Setup
In this study, the heat transfer in pulsating flow with regularly distributed hot spots in axial direction [1] has been treated. At one fixed axial position, bulk velocity and temperature fluctuate harmonically in time. The spatial wavelength of the hot spots is assumed to be very large compared to the thickness of the wall and the thermal boundary layers. Furthermore, temperature gradients in axial direction are neglected compared to the ones in wall normal direction. This leads to the assumption that temperature profiles in the pulsating flow are quasi steady and the conjugate heat transfer problem reduces to a one dimensional heat conduction problem inside the wall with unsteady boundary conditions. The hydrodynamic effects that lead to the conjugate heat transfer are abstracted into heat transfer coefficients and bulk temperatures, regardless if the flow is in laminar, turbulent or transient regime. Luikov showed that the use of a heat transfer coefficient as boundary condition is not always meaningful or physical [7], specially in strong transient problems where the heat flux may locally change direction leading to negative heat transfer coefficients. In such a case, the full conjugate problem with boundary condition of the fourth kind should be used. Here, we remark the assumption of quasi steady temperature profiles in the fluid. Figure 1 shows a sketch of the model used in this investigation. A wall segment of thickness $l^*$ and constant properties interacts with a hot and a
cold flow on each side.

The investigation focuses on the response of the wall to harmonic perturbations of the hot bulk flow:

\[
T_h^* = T_{h0}^* + T_{h1}^* \cdot \cos(\omega^* t^*)
\]

where mean values are denoted with index 0 and the amplitudes of the oscillating parts with index 1. The oscillation frequency \(\omega^*\) of the heat transfer coefficient \(\alpha_h^*\) and the hot bulk temperature \(T_h^*\) are equal. However, a possible phase lag \(\phi\) between them is considered. To avoid nonphysical violations of the second law of thermodynamics, the heat transfer coefficient can only take positive values and the restriction \(0 \leq \alpha_{h1}^* \leq \alpha_{h0}^*\) has been assumed. On the other side, the cold flow conditions \(\alpha_c^*\) and \(T_c^*\) are kept constant.

The governing equation of the problem is derived from an energy balance of a finite wall segment in integral form:

\[
\frac{d}{dt} \int \int \int \rho_w c_w u^* dV^* = \int \int (\dot{q}_h^* - \dot{q}_c^*) dA^* .
\]

The internal energy can be expressed in terms of the local wall temperature \(u^* = c_w^* T^*(x^*)\) whereby density \(\rho_w^*\) and heat capacity \(c_w^*\) are assumed to be constant. Furthermore, Newton’s law of convective heat transfer \(\dot{q}^* = \alpha^*(T_a^* - T_b^*)\) is applied for the boundary heat fluxes \(\dot{q}_h^*\) and \(\dot{q}_c^*\). The wall surface temperatures, heat transfer coefficients and bulk flow temperatures of the corresponding sides are plugged in and the integrals have been simplified according to the one dimensional character of the problem:

\[
\rho_w^* c_w^* \frac{d}{dt} \int T_w(x)^* dx^* = \alpha_h^*(T_h^* - T_w^*(x^* = 0)) - \alpha_c^*(T_w^*(x^* = l^*) - T_c^*).
\]

Substitution of the harmonic perturbation relations Eq. (1) for the hot bulk flow and introducing the following set of nondimensionalizations \(T = \frac{T^* - T_{h0}^*}{T_{h1}^*}, \quad x = \frac{x^*}{l^*}, \quad t = \frac{(\alpha_{h0}^* + \alpha_{h1}^*)}{\rho_c^* c_w^*} \omega^* t^* = BiFo, \quad \omega = \omega^* \frac{\rho_c^* c_w^* l^*}{(\alpha_{h0}^* + \alpha_{h1}^*)}, \quad \beta_c = \frac{\alpha_{h0}^*}{\alpha_{h0}^* + \alpha_{h1}^*}, \quad \beta_{h0} = \frac{\alpha_{h0}^*}{\alpha_{h0}^* + \alpha_{c}^*}, \quad \beta_{h1} = \frac{\alpha_{h1}^*}{\alpha_{h0}^* + \alpha_{c}^*}, \quad \theta = \frac{T_{h1}^*}{T_{h0}^*} - T_c^*\) lead after some rearrangements to the following generalized nondimensional expression:

\[
\int_0^1 \frac{\partial T_w}{\partial t} dx = \frac{\beta_{h0} + \beta_{h1} \cos(\omega t + \phi)}{\beta_h} (1 + \theta \cos(\omega t) - T_{w[0]} - \beta_c T_{w[l]} .
\]

The choice for the nondimensionalization of the time \(t = BiFo\) corresponds to a characteristic time of the unit impulse response of the wall temperature distribution \(T_w(x, t)\) [8]. The restrictions imposed on the oscillations of the heat transfer coefficient can be translated into
Due to the harmonic behaviour, only one period needs to be considered for the time averaging:

\[
\int_0^1 \frac{\partial T_w}{\partial t} \, dx = -c_T \frac{\partial T_w(t)}{\partial t} + \frac{\beta_h T_w(t)}{q_h} + \beta_{h0}(1 - T_{w|0}) \cos(\phi) + \frac{\beta_{h1}}{2} \cos(\omega t + \phi) . \tag{5}
\]

At this point, two important features of the problem dynamics can already be displayed. Firstly, higher harmonics in form of a second order excitation term \( \frac{\beta_{h1}}{2} \cos(2\omega t + \phi) \) appear. Secondly, a time independent term \( \frac{\beta_{h0}}{2} \cos(\phi) \) leads to an offset of the average values. The sign of this offset depends on the phase lag \( \phi \) of the perturbations.

2.1. Time averaging of the Heat Equation

In order to evaluate the long term consequences of the perturbations on the heat transfer, a proper time averaging of the relations has to be defined and compared to a reference case. As a reference, the heat transfer \( \tilde{q}_{h,ref} \) obtained for the steady case without perturbations (\( \theta = 0 \) and \( \beta_{h1} = 0 \)) will be used:

\[
\tilde{q}_{h,ref} = \beta_{h0}(1 - T_{w|0,ref}) , \quad \tilde{q}_{c,ref} = \beta_c T_{w|l,ref} . \tag{6}
\]

Due to the harmonic behaviour, only one period needs to be considered for the time averaging:

\[
\frac{1}{\tau_p} \int_0^{\tau + \tau_p} \int_0^1 \frac{\partial T_w}{\partial t} \, dx \, dt = \frac{1}{\tau_p} \int_0^{\tau + \tau_p} (\tilde{q}_h - \tilde{q}_c) \, dt . \tag{7}
\]

Time averaged quantities are denoted with an over bar. In the steady-state oscillation the temperature distributions \( T_w(x,t) \) after a full period \( \tau_p \) are unknown but equal. Consequently the internal energy storage term on the left hand side drops out over the averaging operation:

\[
\int_0^1 (T_w(\tau + \tau_p) - T_w(\tau)) \, dx = 0 = \tilde{q}_h - \tilde{q}_c . \tag{8}
\]

This implies that the average heat flux on the cold and the hot side of the wall are equal \( \tilde{q} = \tilde{q}_c = \tilde{q}_h \). Resubstituting the heat flows from equation (5) and recognizing that the oscillating parts drop out over the averaging operation, we obtain the average nondimensional heat flux rate on the hot and the cold side of the wall

\[
\tilde{q}_h = \frac{1}{\tau_p} \int_0^{\tau + \tau_p} (\beta_{h0}(1 - T_{w|0}) + \frac{\beta_{h1}}{2} \cos(\phi)) \, dt , \quad \tilde{q}_c = \frac{1}{\tau_p} \int_0^{\tau + \tau_p} \beta_c T_{w|l} \, dt . \tag{9}
\]

Once the surface temperatures \( T_{w|0} \) on the hot and \( T_{w|l} \) on the cold side are known, the terms can be evaluated.

2.2. Relative Change in Average Heat Flux

The difference between the actual heat flux and the reference heat flux determines whether enhanced heat transfer takes place or not. As the average heat fluxes on hot and cold side are equal, the relative change in average heat flux may be evaluated on both sides of the wall:

\[
\Delta \tilde{q}_h = \frac{\tilde{q}_h - \tilde{q}_{h,ref}}{\tilde{q}_{h,ref}} = \frac{T_{w|0,ref} - T_{w|0}}{1 - T_{w|0,ref}} + \frac{\beta_{h1} \cos(\phi)}{2\beta_{h0}(1 - T_{w|0,ref})} . \tag{10}
\]
and

\[ \Delta \dot{q}_c = \frac{\dot{q}_c - \dot{q}_{c,ref}}{\dot{q}_{c,ref}} = \frac{T_{w[l]} - T_{w[l],ref}}{T_{w[l],ref}}. \] (11)

Consequences for the hot side are not obvious, but on the cold side it is clear that an elevated averaged wall temperature \( \bar{T}_{w[l]} \) with respect to the reference wall temperature \( \bar{T}_{w[l],ref} \) is equivalent to an increase in average heat flux rate.

3. Lumped Capacity Model

The simplest model for the wall that provides dynamic behaviour is a lumped capacity. The model assumes that temperature gradients inside the wall are negligible. In the steady case this condition can be expressed by a Biot number \( Bi = \frac{\alpha h \lambda}{\kappa_w} \leq 0.2 \) \[8\]. In a transient case, it must be additionally ensured that the time needed by the complete wall to adapt to the new boundary conditions must be shorter than a characteristic time of the excitation signal. By approximating the sinusoidal excitation by a square wave signal, we decided to set the characteristic time scale to one fourth of the excitation period \( t = \frac{T}{4} \). This time is compared to the time needed by the wall to achieve a 90% answer level to a step response, as described in \[8\]:

\[ 0.1 \bar{T}_w \leq \bar{T}_w e^{-t} \quad \Rightarrow \quad \tau_p \geq 10 \quad \Rightarrow \quad \omega \leq \omega_{crit} = 0.1 \cdot 2\pi. \] (12)

This leads to a critical frequency \( \omega_{crit} = 0.1 \cdot 2\pi \) that determines the upper frequency limit of the model. The wall temperature \( T_w = T_{w[0]} = T_{w[l]} \) is thus independent of the coordinate \( x \) and the differential equation of the system (5) reduces to:

\[ \frac{\partial T_w}{\partial t} + T_w (1 + \beta_{h1} \cos(\omega t + \phi)) = \beta_{h0} + \frac{\beta_{h1} \theta}{2} \cos(\phi) + \beta_{h0} \theta \cos(\omega t) + \beta_{h1} \cos(\omega t + \phi) + \frac{\beta_{h1} \theta}{2} \cos(2\omega t + \phi). \] (13)

Due to the term \( T_w \beta_{h1} \cos(\omega t + \phi) \) it is a nonlinear inhomogeneous differential equation. Two approximations for low and high frequencies will be investigated. The reference wall temperature is determined by setting the oscillation amplitudes to zero \( T_{w[ref]} = \beta_{h0} \).

3.1. Low frequency regime

For low frequencies a quasi steady \( \frac{\partial T_w}{\partial t} \approx 0 \) approximation is used. The simplified and refactORIZED heat equation (13) becomes:

\[ T_w = \frac{1 + \theta \cos(\omega t)}{1 + \frac{1-\beta_{h0}}{\beta_{h0} + \beta_{h1} \cos(\omega t + \phi)}}. \] (14)

At this point an analytical time integration of expression (14) would directly lead to a closed form solution for the average heat transfer. Unfortunately we have not succeed, but still some checks and conclusions can be derived from the equation. An adiabatic boundary on the cold side is described by \( \alpha_s^* = 0 \rightarrow \beta_{h0} = 1 \). In this case the wall temperature follows the temperature on the hot side exactly \( T_w = T_h \), which is correctly predicted by the approximation. Setting the oscillating parts to zero \( \theta = \beta_{h1} = 0 \) represents the standard case of steady heat transfer. The predicted solution of the wall temperature is then equal to the pecl et equations for heat resistances \( \alpha_{h0}^*, \alpha_{s}^* \). The maximal values of the wall temperature are between 0 and 2. Those values are reached for maximum amplitudes of the heat transfer coefficient \( \beta_{h1} = \beta_{h0} \) and temperature \( \theta = 1 \). The term \( \cos(\omega t + \phi) \) can be set to 1 by choosing the corresponding \( \phi \) and
the oscillations of $T_w$ will then be between 0 and $\frac{1+\cos(\omega t)}{2(1+\beta^{2}_h)}$. Considering the restriction $\beta_{h0} \leq 1$, the maximal wall temperature is obtained for $\beta_{h0} = 1$, which in return is the adiabatic boundary condition. The maximum wall temperature is thus 2 when $\cos(\omega t) = 1$. In order to obtain a high average temperature, the denominator should be minimal when the numerator is maximal and vice versa. This condition is fulfilled for phase angles $\phi = 0$.

3.2. High frequency regime
At high frequencies the time derivative dominates the dynamic behaviour. To obtain a linear inhomogeneous differential equation, the wall temperature in the nonlinear excitation term $T_w$$\beta_h \cos(\omega t + \phi)$ is approximated by the constant excitation part on the right hand side $T_w \approx \beta_{h0} + \frac{\beta_h \theta}{2} \cos(\phi)$:

$$\frac{\partial T_w}{\partial t} + T_w = \beta_{h0} + \frac{\beta_h \theta}{2} \cos(\phi) + \beta_{h0} \theta \cos(\omega t) + (1 - \beta_{h0} - \frac{\beta_h \theta}{2} \cos(\phi)) \beta_h \cos(\omega t + \phi) + \frac{\beta_h \theta}{2} \cos(2\omega t + \phi).$$

The ansatz for solving the differential equation consist of three linear independent parts $T_w(s,t) = T_{w0} e^{\lambda t} + T_{w1}$ with $s = i\omega + \lambda$ and $i = \sqrt{-1}$: the homogeneous solution, which is decaying in time due to $\lambda = -1$ (compare (12)); the oscillating excitation term for a given frequency $\omega$; and the constant term $T_{w0}$. By averaging over long times, the homogeneous and oscillating parts cancel out and the solution for the temperature and the corresponding increase in heat transfer (11) are:

$$\bar{T}_w = \beta_{h0} + \frac{\beta_h \theta}{2} \cos(\phi), \quad \Delta \dot{q} = \frac{\beta_h \theta}{2\beta_{h0}} \cos(\phi).$$

The high frequency approximation clearly shows that the parameter $\phi$ is determining the sign and the absolute value of the relative change in heat transfer. Similar to the low frequency approximation, the parameter value $\phi = 0$ maximizes the average wall temperature and thus the average relative enhancement in heat transfer. Large oscillations of both, the temperature $\theta$, and the heat transfer coefficient $\beta_h$, are favorable for a larger impact on the average heat transfer. Finally, it becomes clear that the restriction of $\beta_h \leq \beta_{h0}$ limits the maximum relative increase or decrease in heat transfer to $\frac{\beta_h}{\beta_{h0}}$. In the limit of an infinitely large heat transfer coefficient on the cold side of the wall $\beta_c \rightarrow 1$, the model converges to a constant wall temperature model, as the one analyzed by Lundgren et al. [3]. It is remarkably that the heat transfer coefficient on the cold side of the wall $\beta_c$ does not influence the relative change in heat transfer and our results are in agreement with Lundgren et al.

3.3. Results and Validation
In order to validate the conclusions drawn from the two approximations, they are compared to the numerical integration of the full nonlinear differential equation (5). The parameters are set to maximum oscillation amplitudes of the temperature $\theta = 1$ and heat transfer coefficient $\beta_h = \beta_{h0}$. Furthermore, the average heat transfer coefficient on hot and cold side are set equal $\beta_{h0} = \beta_c = 0.5$ to achieve maximum heatflux rates through impedance matching. The three figures 3, 4 and 5 show the relative change in heat flux rate for the low frequency approximation, the full differential equation at three different frequencies and the high frequency approximation at the highest corresponding frequency. Horizontal lines depict the average values.

As already mentioned, increased average heat transfer occurs for values of $\phi \approx 0$. For a phase angle of $\phi = \pi$, the relative heat transfer is reduced, whereas for an intermediate phase angle,
depending on the frequency, both, slightly enhanced or reduced heat transfer may occur. As can be clearly seen in figure 3, the average heat transfer of the numerical integrated model equation increases with rising frequency and approaches the value of the high frequency approximation. Unfortunately, the lumped capacity model is limited to \( \omega_{\text{crit}} = 0.1 \cdot 2\pi \), which is not covered by the highest frequency \( \omega = 0.2 \cdot 2\pi \) displayed in the figures above.

4. Finite Volume Model
A finite volume discretized wall model as shown in figure 6 is introduced to take into account temperature gradients inside the wall and extend the range to higher frequencies. The motivation was to discretize the wall into several lumped capacity segments. In analogy to the lumped capacity model, the condition for the frequency limit at a given discretization grade \( N \) is:

\[
0.1\tilde{T}_w \leq \tilde{T}_w e^{-tN} \quad \rightarrow \quad \tau_p \geq \frac{10}{N} \quad \rightarrow \quad \omega \leq 0.1N \cdot 2\pi.
\]  

(17)

For \( N = 1 \) the condition converges to the one of the lumped capacity (12). In this study a discretization grade of \( N = 35 \) was chosen, which gives an upper frequency limit of \( \omega \leq \omega_{\text{crit}} = 3.5 \cdot 2\pi \). Alternatively for a given frequency the required number of discretization points can be expressed by \( N \geq N_{\text{min}} = 10\frac{\omega}{2\pi} \). The system of equations describing the dynamic of the discretized wall temperature and the output equation are given in the form:

\[
\dot{\tilde{T}} = A \cdot \tilde{T} + \tilde{b}T_h, \quad T_{w|l} = \tilde{c} \cdot \tilde{T}.
\]  

(18)

A detailed description of the model is given in [9]. The matrix \( A \) is mainly a band matrix containing the coefficients of the centered point finite volume integration scheme. The output
\[ \vec{c} \cdot \vec{T} \] evaluates the temperature on the surface of the cold side of the wall. The input vector \( \vec{b}_T \) contains the nonlinear excitation due to the oscillations of the heat transfer coefficient \( \text{Bi}_{h} = \text{Bi}_{h0} + \text{Bi}_{h1} \cos(\omega t + \phi) \) and the bulk flow temperature \( T_h = 1 + \theta \cos(\omega t) \) on the hot side:

\[
\vec{b}_T = N^2 \begin{bmatrix} 
\frac{2\text{Bi}_{h1}}{\text{Bi}_{h0}+2N} & \ldots & 0
\end{bmatrix}^T T_h.
\] (19)

From a linearization of the differential equation of the first wall element the approximation for high frequencies is then derived. Here we will give just the result for the averaged relative change in heat flux rate:

\[
\Delta \dot{q} = \frac{\text{Bi}_{h1}\theta}{2\text{Bi}_{h0}} \cos(\phi).
\] (20)

This is in close analogy to the equation given for the lumped capacity high frequency approximation (16). In order to validate the result of the high frequency approximation, the finite volume model was implemented in openmodelica[10] with a discretization of \( N = 35 \) finite volume elements. Figure 7 shows the result for the numerical integration of the nonlinear system equation at a phase angle \( \phi = 0 \) and the frequencies \( \omega = 1 \cdot 2\pi \) and \( \omega = 10 \cdot 2\pi \) and the high frequency approximation for the former frequency. Compared to the results for the lumped capacity in figure 3, it becomes clear that also the dynamic behavior of the finite volume modeled system is very similar to the one of the lumped capacity. The dashed lines are representing the (averaged) heat transfer of a frequency \( \omega = 10 \cdot 2\pi \) and show that the solution of the the finite volume model converges against the high frequency approximation at rising frequencies.

5. Conclusion

The oscillation of temperature and heat transfer coefficient on the hot side of a wall modeled as a lumped capacity gives rise to a nonlinear coupling phenomenon. This interaction may introduce second order terms and a constant offset to the heat flux in pulsating flows. Depending on the relative phase lag between the oscillations, enhanced or degraded average heat flux occurs. The maximum average heat flux is obtained when temperature and heat transfer coefficient oscillate in phase. Large amplitudes of temperature and heat transfer coefficient fluctuations intensify the effect. For a given set of optimal parameters, the average relative heat transfer enhancement increases with rising non-dimensional frequencies, where the non-dimensional frequency is dependent on geometry and material properties. The enhancement in relative averaged heat flux rate is limited to a value smaller than 50% of the nondimensional temperature amplitude. For vanishing temperature fluctuations, the relative change in heat transfer is predicted to be smaller than 0, which is in accordance with the results of Zudin [6]. As the lumped capacity model is restricted to low frequencies, a finite volume discretization of the wall is introduced. The fundamental nonlinear coupling between the temperature and heat transfer coefficient oscillations is confirmed also for the finite volume model.

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