TIME-RESOLVED HEAT TRANSFER IN ENGINE INTAKE MANIFOLD

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ABSTRACT. In this paper the authors present the development and application of transient heat transfer correlations to the intake manifold of a spark ignition engine. During one engine operating cycle, the air motion in the intake manifold encompasses two phases which closely correspond to the open and closed durations of the intake valve. Phase I is associated with air induction and Phase II is associated with air oscillation-decay process. For Phase I, the authors develop an unsteady heat transfer correlation that is not only a function of Reynolds number, but also a function of the changing rates of velocity. For Phase II, the authors adopt the turbulence decay equation and its associated heat transfer correlation. Comparisons between the measured and predicted heat transfer show that the proposed approach corrects the phase and amplitude errors generated by using steady-state correlations.

NOMENCLATURE

D = hydraulic diameter of the intake manifold

L = integral length scale = D for intake manifold

$Nu = hD/k$, Nusselt number

$Pr = \nu/\alpha$, Prandtl number

$Re = \rho U D/\mu$, Reynolds number

$\Pi = (L/U^2)(dU/dt)$, Π number for normalized changing rate of velocity

$St = h/\rho c_p U = Nu/RePr$, Stanton number

$C_f = \tau_w/(\rho U^2/2)$, skin friction coefficient

$\dot{q}$ = heat flux

U = instantaneous gas velocity

u' = turbulence intensity

k = thermal conductivity

Greek symbols

τ_w = surface shear stress

ρ = density

c_p = constant pressure specific heat

α = thermal diffusivity

γ = ratio of the specific heat

μ = dynamic viscosity

ν = kinematic viscosity

ε_m = turbulence kinematic viscosity

λ = Taylor's micro-scale

Subscript

e = effective parameters

INTRODUCTION

Heat transfer is an important process in the intake manifold of engines. It increases the charge temperature which reduces the volumetric efficiency and the tolerance to engine knock, and also causes higher chemical reaction rates leading to increased NOx emissions. It also affects engine performance and emissions through enhancing the fuel evaporation and charge mixing process in the engine intake ports and cylinders. Therefore, many correlations have been developed to predict the heat transfer in the intake manifold of engines. These correlations are usually in the form of $Nu = a\,Re^b$ or $Nu = a\,Re^b Pr^n$ and differ only by empirically fitted constants a, b and n [1, to 6]. Some of these correlations are listed below.

Correlations Used for Engine Intake Manifolds

Dittus-Boelter correlation (1930) [1].

$$Nu = 0.023\,Re^{0.8} Pr^{n} \quad (n = 0.4 \text{ for } Tw > Tg \text{ and } n = 0.3 \text{ for } Tw < Tg) \tag{1}$$

Bauer, et al. correlation (1998) for intake [2].

$$Nu = 0.062\,Re^{0.73} \text{ for straight manifold, } \quad Nu = 0.14\,Re^{0.66} \text{ for curved manifold} \tag{2}$$

Depcik-Assanis correlation (2002) [3].

$$Nu = 0.07\,Re^{3/4} \tag{3}$$

These correlations provide reasonable agreement with experimental data in fully-developed steady pipe flows and acceptable agreement with time-resolved experimental data in unsteady flows with slow velocity variation, assuming they can be treated as quasi-steady. However, for highly unsteady flows with rapid velocity variations, such as flows in the engine intake manifolds, these correlations can produce large errors on both phase and amplitude on time-resolved heat transfer calculation [2]. Therefore, an unsteady heat transfer correlation needs to be developed to deal with the time-resolved unsteady convective heat transfer in the engine intake manifolds.

UNSTEADY HEAT TRANSFER CORRELATION

Dimensional Analysis of Boundary Layer Momentum Equation

The time-averaged, 2-D unsteady, turbulent, viscous and incompressible boundary layer (see Fig. 1) momentum equation for uniform pressure field can be written as,

$$\frac{\partial \bar{u}}{\partial t} + \bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{v}\frac{\partial \bar{u}}{\partial y} = \nu_e \frac{\partial^2 \bar{u}}{\partial y^2} \qquad (4)$$

where $\nu_e = \nu + \varepsilon_m$ is the effective kinematic viscosity for the turbulent flows. Eq. (4) can be normalized using the following dimensionless variables

$$u^* = \frac{\bar{u}}{U}, \quad v^* = \frac{\bar{v}}{U} \text{ where } U = U(t); \quad x^* = \frac{x}{L}, \quad y^* = \frac{y}{L}, \text{ and } t^* = \frac{t}{L^2/\alpha_e} = \text{Fourier number} \qquad (5)$$

The normalized momentum equation can be expressed as,

$$\left\langle \frac{L}{U^2}\frac{dU}{dt} \right\rangle u^* + \frac{1}{Re\,Pr}\frac{\partial u^*}{\partial t^*} + u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial u^*}{\partial y^*} = \frac{1}{Re}\frac{\partial^2 u^*}{\partial y^{*2}} \qquad (6)$$

From Eq. (6), three dimensionless variables that characterize the momentum boundary layer can be found as,

$$Re = \frac{\rho U L}{\mu}, \quad Pr = \frac{\nu}{\alpha} \text{ and } \boxed{\Pi = \frac{L}{U^2}\frac{dU}{dt}} \qquad (7)$$

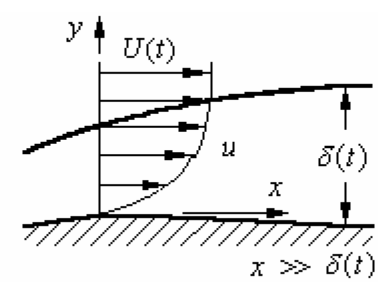

Figure 1. Unsteady momentum boundary layer with time varying velocity

where Reynolds and Prandtl numbers Re and Pr have been commonly used in steady heat transfer correlations, and Π is a new dimensionless variable that characterizes the unsteady nature of the momentum boundary layer. Its physical meaning may be expressed as,

$$\Pi = \frac{L}{U^2}\frac{dU}{dt} = \frac{\rho\,(dU/dt)}{\rho U(U/L)} = \frac{\text{Temporal gradient of inertial force}}{\text{Spatial gradient inertial force}} \qquad (8)$$

From the above dimensional analysis, it can be concluded that in addition to Reynolds and Prandtl numbers Re and Pr, Π also plays a role on heat transfer in unsteady flows with rapid velocity variation.

Dynamic Variables

In this section, the authors introduce the concept of dynamic variables to extend the steady correlations into unsteady correlations. A steady variable has only one value. An unsteady variable assumes a series of values, but at each time step, its value has information only related to that time step. A dynamic variable captures not only the information at a current time step, but also its normalized instantaneous rate of change. Using the dimensionless variable Π derived above, a dynamic velocity U_{dyn} for convective heat transfer can be constructed as,

$$U_{dyn} = U(1 + C_2 \Pi) = U\left(1 + C_2 \frac{L}{U^2}\frac{dU}{dt}\right) \tag{9}$$

where C_2 is a calibration constant. The dynamic velocity U_{dyn} contains information for both the instantaneous value of the velocity and its normalized rate of change. Based on the dynamic velocity U_{dyn}, the dynamic Reynolds number Re_{dyn} can be defined as,

$$Re_{dyn} = \frac{\rho U_{dyn} L}{\mu} = \frac{\rho U L}{\mu}(1 + C_2 \Pi) = Re\left(1 + C_2 \frac{L}{U^2}\frac{dU}{dt}\right) \tag{10}$$

Similarly, a dynamic skin friction concept can be developed by analogy to the steady-state friction. It is well known that in steady flows the skin friction coefficient can be expressed as:

$$\left(\frac{1}{2}C_f\right) = C_1\, Re^{-\beta} \tag{11}$$

where C_1 is a calibration constant. The value of β can be obtained from Moody chart for pipe flows. For smooth pipes, $\beta = 1$ for laminar flow, $\beta = 1/4$ for $Re \leq 2\times10^4$ and 1/5 for $Re \geq 2\times10^4$ turbulent flows. Then, in unsteady flows, the dynamic skin friction coefficient can be expressed as,

$$\left(\frac{1}{2}C_f\right)_{dyn} = C_1\, Re_{dyn}^{-\beta} = C_1\, Re^{-\beta}\left(1 + C_2 \frac{L}{U^2}\frac{dU}{dt}\right)^{-\beta} \tag{12}$$

Similarly, based on the dynamic variable concept, the Colburn analogy for convective heat transfer in steady flows, ie.

$$St\,Pr^{2/3} = \frac{Nu}{RePr^{1/3}} = \left(\frac{1}{2}C_f\right) \tag{13}$$

can be extended for unsteady flows as,

$$St_{dyn}\;Pr^{2/3} = \frac{Nu_{dyn}}{Re_{dyn}\,Pr^{1/3}} = \left(\frac{1}{2}C_f\right)_{dyn} \tag{14}$$

Substituting Eq. (12) into Eq. (14), the dynamic Nusselt number can be derived as,

$$Nu_{dyn} = C_1 Re_{dyn}^{1-\beta} Pr^{1/3} = C_1 Re^b Pr^{1/3}\left(1 + C_2\frac{L}{U^2}\frac{dU}{dt}\right)^b \tag{15}$$

where $b = 1-\beta$. For smooth pipe, b equals to zero for laminar flows, 0.75 for $Re \le 2\times10^4$ and 0.8 for $Re \ge 2\times10^4$ turbulent flows. For flows over a flat wall, b = 0.5 for both laminar and turbulent flows. Eq. (15) is the new correlation developed by the authors to predict the unsteady convective heat transfer for flows with rapid velocity variation. It indicates that the unsteady heat transfer is not only a function of Reynolds and Prandtl numbers but also a function of the changing rate of the velocity. In the limit of steady flows, the rate of change of velocity tends to zero, the unsteady velocity correction term tends to one, and the unsteady correlation collapses to the steady correlation. Eq. (15) also shows that the unsteady effect increases as the length scale L and changing rate of the velocity dU/dt increase, and the velocity U decreases.

HEAT TRANSFER CORRELATION FOR THE TURBULENCE DECAY PROCESS

When the valve in a flow system suddenly closes, the mean flow velocity drops to zero rapidly following some velocity oscillation and the turbulence in the flow system experiences a decay process. During this process, the decaying turbulence plays a dominant role in the convective heat transfer [2, 8]. Under the assumption of isotropic turbulence, the decaying turbulence intensity can be calculated using the turbulence decay law [9, pp. 79 to 80],

$$\frac{d(\nu/u'^2)}{dt} = \frac{10}{Re_\lambda^2} \tag{16}$$

where the Reynolds number Re_λ is based on the turbulence intensity u' and Taylor's micro-scale λ. The associated convective heat transfer can be calculated using the following correlation [8, pp. 99 to 100],

$$Nu = C_3 \, Re_t^{\,d} \tag{17}$$

where C_3 is a calibration constant and d is suggested to be 0.33. The Reynolds number Re_t is based on turbulence intensity and integral length scale.

TIME-RESOLVED HEAT TRANSFER IN ENGINE INTAKE MANIFOLD

Experimental Results and Analysis of Bauer et al.

Bauer et al. [2] used electric heaters on the outside of a specially designed engine intake manifold to produce heat transfer and carried out time-resolved heat flux measurements. Their experimental result shows that the air motion in the intake manifold encompasses two phases which closely correspond to the open and closed durations of the intake valve. The heat flux in Phase I is associated with air induction from 0 to ~180 degree CA (Crank Angle) when intake valves are opening. The heat flux in Phase II is associated with air oscillation-decay process from ~180 to 720 degree CA when the intake valve is closed. Bauer et al. compared their measurements with the steady turbulent flow correlation Eq. (1), and the CFD simulation results of Malan and Johnston [8] for heat transfer in a transiently growing shear free boundary layer (Fig. 2).

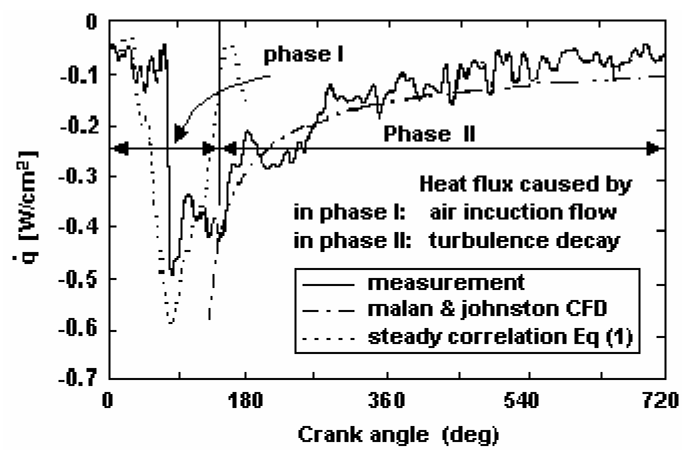

Figure 2. Measured heat flux after subtraction of pressure fluctuation induced component in comparison to steady correlation (0 to ~180 degree), and the CFD solution of Malan and Johnston (~180 to 720 degree), 2750 rpm /1 bar MAP.

The comparison results show that the heat flux in the turbulence decay process during Phase II can be modeled nicely using Malan and Johnston's CFD simulation result after matching the initial turbulence intensity. However, large errors in both phase and amplitude are produced in modeling the heat flux induced by the air induction during Phase I. From the comparison results, Bauer et al. concluded that it is inappropriate to assume quasi-steady heat transfer behavior in the engine intake manifolds. Nevertheless, since unsteady heat transfer correlations were not available at that time, Bauer et al. did not correct those prediction errors. In the next section, the authors will show that the new unsteady heat transfer correlation Eq. (15) derived in this paper can be used to correct those prediction errors.

Application of Unsteady Heat Transfer Correlation to the Air Induction Flow in Phase I

Using the velocity measurements of Bauer et al. (Fig. 3), the authors applied the unsteady heat transfer correlation Eq. (15) to the heat flux induced by the air induction flow in Phase I. In order to make closer comparison with the steady correlation used by Bauer et al., the authors set $b = 0.8$ and $n = 0.4$ in the Dittus-Boelter correlation. The calibration constants $C_1 = 0.018$ and $C_2 = -0.75$ were adjusted to match the measured heat flux. The length scale L in Eq. (15) was chosen to be the hydraulic diameter of the engine intake manifold. Then, Eq. (15) becomes,

$$Nu = 0.018\, Re^{0.8} Pr^{0.4} \left(1 - 0.75 \frac{D}{U^2} \frac{dU}{dt}\right)^{0.8} \tag{18}$$

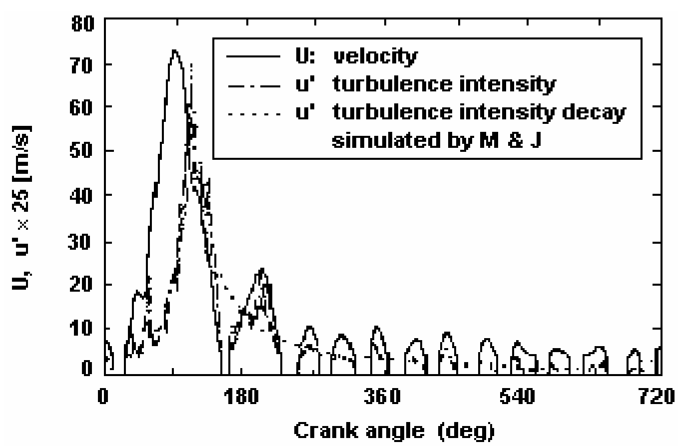

Figure 3. Velocity and turbulence intensity measurements by Bauer et al. using TSI hot wire; velocity measurements on centerline of the straight intake manifold, turbulence intensity measurement 2 mm from the wall. Turbulence intensity decay simulation result by Malan and Johnston for a transiently growing shear free boundary layer. (The absolute values below the calibration limit of 3.5 m/s are arbitrarily set to zero). 2750 rpm /1bar MAP

The comparison results are shown in figure (4). The results indicate that the unsteady heat transfer correlation developed in this paper is capable of capturing the transient heat flux induced by the highly unsteady air induction flow in the engine intake manifold. The proposed correlation improves the accuracy of predicting both the phase and the amplitude of the heat flux.

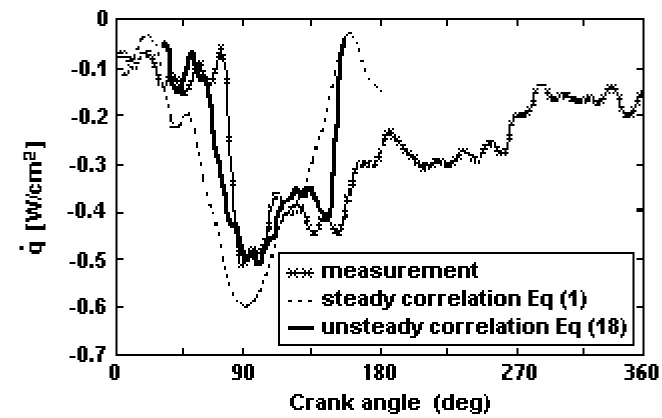

Figure 4. Comparison of heat flux predictions using steady and unsteady correlations to the measured heat flux, 2750 rpm /1bar MAP.

Results Analysis

During ~45 to 90 degrees CA, when the air induction velocity increases, the unsteady velocity correction term introduces a negative correction to reduce the heat flux predicted by the steady-state correlation. During ~90 to 180 degrees CA, when air induction velocity decreases, the correction term makes a positive correction to increase the heat flux predicted by the steady-state correlation. These corrections directly improve the phase prediction of the heat flux.

At lower air induction velocity, the unsteady velocity correction term makes a larger correction, and vice versa, at higher air induction velocity, the correction term results in a smaller correction. Around peak velocity where $dU/dt \approx 0$ and velocity is very large, the correction term ≈ 1, i.e., there is almost no correction. From the results of figure (4), it is clear that these kinds of corrections are physically correct and can be expected to improve the prediction accuracy of heat flux.

Furthermore, the unsteady heat transfer correlation captures the first and second humps of the heat flux induced by the air induction flow in the engine intake manifold. As indicated by Eq. (18), the dominant factor behind the first hump is the peak heat flux predicted by the steady-state correlation. The dominant factor behind the second hump is the unsteady velocity correction term. Because the term $(1/U^2)$ increases very fast when velocity decreases, the unsteady velocity correction term increases faster than the heat flux drop predicted by steady-state correlation during the rising phase leading to the second heat flux hump.

Because of the discontinuity of the measured velocity, the prediction of unsteady heat transfer correlation deviates from the heat flux measurements starting from ~ 146 degree CA. The measured velocity below the calibration limit of 3.5 m/s is arbitrarily set to zero, suddenly $U = 0$ and $dU/dt = 0$. The unsteady velocity term becomes unstable in this range and loses its corrective ability.

Application of Heat Transfer Correlation for the Turbulence Decay Process in Phase II

It has been shown in figure (2) that the heat flux in the turbulence decaying process (Phase II) can be modeled nicely using Malan and Johnston's CFD simulation result after matching the initial turbulence intensity. From their CFD results, Malan and Johnston proposed that the heat flux in the turbulence decaying process can be calculated using correlation Eq. (17).

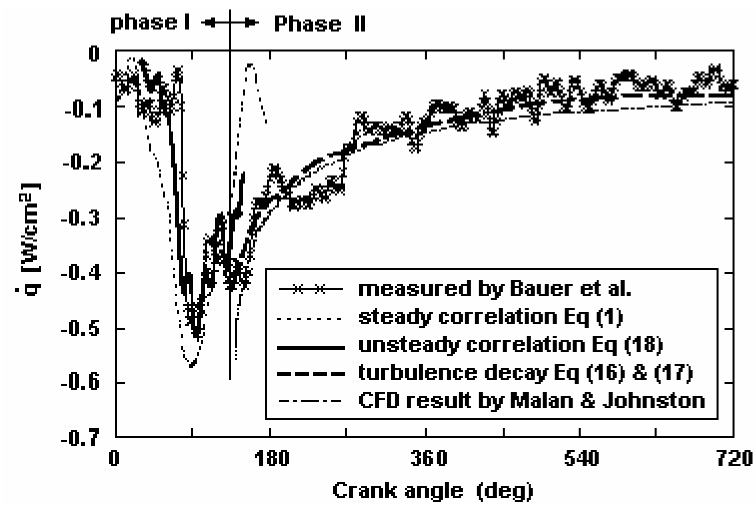

Figure 5. Comparison of heat flux predictions using different correlations to the measured result

In this section, the authors show that the heat flux in Phase II can also be modeled reasonably using the simple turbulence decay law Eq. (16) and correlation Eq. (17) when the turbulence intensity measurement is available. First, the authors smooth the Bauer et al.'s results of turbulence intensity measurement (Fig. 3), and calculate Reynolds number Re_λ that is based on the turbulence intensity u' and Taylor's micro-scale λ using the turbulence decay law Eq. (16). Then, the Reynolds number Re_λ and Eq. (17) with $C_3 = 15.0$ are used to calculate the heat flux in the turbulence decay process (Phase II). Figure (5) shows that Eq (16) and (17) predict the heat flux in Phase II well.

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CONCLUSIONS

The study in this paper concludes that the unsteady nature of flows in the engine intake manifold affects the heat transfer significantly in terms of both phase and amplitude. It is inappropriate to use steady-state correlations to predict the time-resolved heat flux in engine intake manifolds. For flows with rapid velocity variation, heat transfer is not only a function of Reynolds and Prandtl numbers, but also a function of the changing rate of velocity. The unsteady heat transfer correlation Eq (15) developed by the authors from dimensional analysis is capable of improving the prediction accuracy of the time-resolved heat flux in engine intake manifolds during the air induction process in Phase I. It captures the unsteady nature of heat transfer with unsteady velocity variation. The turbulence decay law Eq. (16) and heat transfer correlation Eq. (17) can be used to predict the heat flux during the turbulence decay process of Phase II. Since all model equations have been developed based on general mechanisms of unsteady turbulent flows, they should be applicable to estimating other engineering heat transfer processes in similar situations.

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