Advanced near-wall modeling for engine heat transfer

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ARTICLE INFO

ABSTRACT

Recent developments in the engine heat transfer modeling tend to improve existing wall heat transfer models (temperature wall functions) which mostly rely on the standard or low-*Re* variants of k-ε turbulence model. Presently applied mesh resolutions already allow for first near-wall computational cells reaching the buffer or locally even viscous/conductive sub-layer, thus increasing the importance of more sophisticated modeling approach. As temperature gradient-induced density and fluid property variations become significant, wall heat transfer is strongly influenced by property variations (viscous/conductive sub-layer) and predictive capability of the turbulence model (buffer region), standard wall laws being inadequate anymore, even for attached boundary layers. The present approach relies on the k-ζ-f turbulence model and formulates a compressible wall function of Han and Reitz in the framework of hybrid wall treatment. The model is validated against spark ignition (SI) engine heat transfer measurements. Predicted wall heat flux evolutions on the cylinder head exhibit very good agreement with the experimental data, being superior to similar numerical predictions available in the published literature.

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1. Introduction

Prediction of heat transfer plays an important role in engine development as heat losses influence overall engine efficiency, exhaust emissions and component thermal stresses. Due to prohibitive computational costs, internal combustion (IC) engine simulations are still mainly limited to the Reynolds-averaged Navier–Stokes (RANS) framework and application of the standard wall functions. At the same time, due to continuous increase in computational power, more sophisticated modeling approach will eventually become inevitable as applied mesh resolutions already allow for first near-wall computational cells reaching the buffer or locally even viscous/conductive sub-layer. Improvements of the existing wall heat transfer models (temperature wall functions) for in-cylinder flows are mostly based on the standard or low-Reynolds number variants of k-ε turbulence model, which are known to perform poorly in engine relevant configurations such as impinging jets with heat transfer (Bovo, 2014). Irrespective of complexity of the heat transfer model, its performance strongly relies on capability of the underlying turbulence model to capture near-wall transport phenomena. Numerous engine simulations, however, still employ 'standard' approach for turbulence (e.g. standard k-ε) and wall heat transfer (e.g. temperature wall function of Jayatilleke, 1969) models which do not account for

near-wall effects (viscous and non-viscous), variable properties and increase of the turbulent Prandtl number. Consequently, this results in substantial under-predictions (log-law region) or over-predictions (viscous/conductive sub-layer) of wall heat transfer. The previous work pertinent to engine heat transfer modeling is scrutinized in the publications of Rakopoulos et al. (2010) and Nuutinen et al. (2014). Rakopoulos et al. (2010) have evaluated the most popular heat transfer formulations used in commercial and research computational fluid dynamics (CFD) codes. Along with a detailed review of research on heat transfer in internal combustion engines, they reported the extensive computational investigation of engines running under motoring conditions. The authors proposed a comprehensive temperature wall function that includes unsteady pressure term and performs good during the compression stroke. Under-predictions of the measured heat flux peak values by 35–50% revealed weakness of incompressible temperature wall functions, whereas the model of Han and Reitz (1997) was found to be the best compromise between simplicity and accuracy. Apart from variable density effects already observed by Han and Reitz (1997) and Angelberger et al. (1997), Nuutinen et al. (2014) included combined variable properties effects on heat transfer and near-wall turbulence modifications in their imbalance wall function. They solved simplified boundary layer equations for enthalpy, momentum, turbulent kinetic energy and dissipation in wall adjacent cells. The equations include temperature gradient-induced density and property variations and complete imbalance contributions such as convection, pressure gradient and external sources in compact forms. The resulting model is valid

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Table 1
Model coefficients.

C_1	C_2	$C_{\varepsilon 1}$		$C_{\varepsilon 2}$	C_μ	
0.4	0.65	$1.4(1+0.045/\zeta^{0.5})$		1.9	0.22	
σ_k	σ_ε	σ_ζ		C_τ	C_L	C_η
1.0	1.3	1.2		6.0	0.36	85

with near-wall grid resolution ranging from viscous sub-layer to fully turbulent region, yielding improved heat transfer predictions compared to the wall function of Angelberger et al. (1997).

The present work is based on more advanced, k-ζ-f turbulence model which allows integration to the wall, with incorporated molecular and wall-blocking modifications (Hanjalić et al., 2004). Consequently, the model is capable of capturing turbulent stress anisotropy near wall and predicting heat transfer with more fidelity. Hybrid wall treatment in AVL FIRE® (2013) is extended to the temperature wall function of Han and Reitz. The resulting hybrid formulation is validated against the spark-ignition (SI) engine heat transfer measurements of Alkidas and Myers (1982).

2. Turbulence model

The k-ζ-f RANS model employed in the present work relies on the elliptic relaxation concept providing a continuous modification of the homogeneous pressure-strain process as the wall is approached to satisfy the wall conditions, thus avoiding the need for any wall topology parameter. The variable ζ represents the ratio $\overline{v^2}/k$ ($\overline{v^2}$ is a scalar property in the Durbin's $v2 - f$ model (1991), which reduces to the wall-normal stress in the near-wall region) providing more convenient formulation of the equation for ζ and especially of the wall boundary condition for the elliptic function f. Hanjalić et al. (2004) demonstrated that the model is numerically very robust and more accurate compared to the simpler two-equation eddy viscosity models. The set of equations constituting the k-ζ-f model reads:

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = (P - \varepsilon) + \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] \tag{1}$$

$$\frac{\partial \varepsilon}{\partial t} + U_j \frac{\partial \varepsilon}{\partial x_j} = \frac{C_{\varepsilon 1}P - C_{\varepsilon 2}\varepsilon}{T} + \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] \tag{2}$$

$$\frac{\partial \zeta}{\partial t} + U_j \frac{\partial \zeta}{\partial x_j} = f - \frac{\zeta}{k}P + \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_\zeta}\right)\frac{\partial \zeta}{\partial x_j}\right] \tag{3}$$

$$L^2\nabla^2 f - f = \frac{1}{T}\left(C_1 + C_2\frac{P}{\varepsilon}\right)\left(\zeta - \frac{2}{3}\right) \tag{4}$$

with the wall boundary condition for f:

$$f_{wall} = \lim_{y \to 0}\left(-2v\zeta/y^2\right) \tag{5}$$

Here, T represents a switch between the turbulent time scale $\tau = k/\varepsilon$ and the Kolmogorov time scale $\tau_\kappa = (v/\varepsilon)^{1/2}$:

$$T = \max\left[\frac{k}{\varepsilon}, C_\tau\left(\frac{v}{\varepsilon}\right)^{1/2}\right] \tag{6}$$

The corresponding length scale L is obtained as a switch between the turbulent and Kolmogorov length scales:

$$L = C_L \max\left(\frac{k^{3/2}}{\varepsilon}, C_\eta\left(\frac{v^3}{\varepsilon}\right)^{1/4}\right) \tag{7}$$

Values of the coefficients appearing in the model equations are outlined in Table 1.

Popovac and Hanjalić (2007) proposed the so-called compound wall treatment with a blending formula following the work of Kader (1981), for the flow properties for which boundary conditions are required at the first near-wall grid node P (wall shear stress, kinetic-energy production and dissipation rate):

$$\phi_P = \phi_v e^{-\Gamma} + \phi_t e^{-1/\Gamma} \tag{8}$$

where 'v' denotes the viscous and 't' the fully turbulent value. In the present approach, variable ϕ represents the wall shear stress, with the blending coefficient Γ as a function of the normalized distance to the wall:

$$\tau_w = \mu \frac{U_p}{y_p}e^{-\Gamma} + \frac{\rho\kappa c_\mu^{1/4}k_p^{1/2}U_p}{\ln(Ey^+)}e^{-1/\Gamma}, \ \Gamma = 0.01\frac{(Pry^+)^4}{1 + 5Pr^3y^+} \tag{9}$$

Using the wall shear stress defined by Eq. (8), the kinetic energy production (P_p) is calculated employing a combined velocity scale:

$$P_p = \frac{\tau_w c_\mu^{1/4}k_p^{1/2}}{\kappa y_p} \tag{10}$$

Along with the dissipation rate as proposed by Basara (2006), these expressions provide the boundary conditions that ensure numerical robustness, which is required in industrial computations such as engine flows. Readers are referred to the original publications of Hanjalić et al. (2004), Popovac and Hanjalić (2007) and Basara (2006) for more specific details about the model developments.

3. Hybrid wall heat transfer model

Han and Reitz (1997) derived a temperature wall function formulation for variable-density turbulent flows. Whereas the effects of unsteadiness and heat release due to combustion were minor for the cases considered (a pancake-chamber gasoline engine and a heavy duty diesel engine), gas compressibility affected engine convective heat transfer prediction significantly. Neglecting the tangential derivatives, pressure gradient, radiation heat transfer and other sources, under assumption of the ideal gas with constant properties, Han and Reitz (1997) integrated the simplified boundary layer equation for energy:

$$-\frac{\rho c_p u^*}{q_w}dT = \frac{1}{\left(\frac{1}{Pr} + \frac{v^+}{Pr_t}\right)}dy^+ \tag{11}$$

u^* being friction velocity and

$$y^+ = \frac{yu^*}{v}; \ v^+ = \frac{v_t}{v} \tag{12}$$

Upon integration of the left hand side of Eq. (11), the non-dimensional temperature profile reads:

$$T^+ = \frac{\rho c_p u^* T \ln\frac{T}{T_w}}{q_w} \tag{13}$$

Based on numerous experimental data (Kays, 1994), the simplified expressions describing turbulent Prandtl number variation were used for the integration of the right hand side of Eq. (11):

$$\frac{v^+}{Pr_t} = a + by^+ + cy^{+2}, \ y^+ < y_0^+$$
$$\frac{v^+}{Pr_t} = my^+, \quad y^+ > y_0^+ \tag{14}$$

with the constants set to be $a = 0.1$, $b = 0.025$, $c = 0.012$ and $m = 0.4767$ for $Pr = 0.7$ and transition value y_0 chosen as 40 (Han and Reitz, 1997). Splitting integration into two parts and neglecting Pr^{-1} in the second part of the integration:

$$T^+ = \int_0^{y_0^+} \frac{1}{Pr^{-1} + a + by^+ + cy^{+2}}dy^+ + \int_{y_0^+}^{y^+} \frac{1}{my^+}dy^+ \tag{15}$$