

A Hybrid Wall Heat Transfer Model for IC Engine Simulations

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ABSTRACT

The present work improves performance of the wall heat transfer model of Han and Reitz employing advanced turbulence modeling and formulating a compressible wall function in the framework of hybrid wall treatment. Some ambiguities related to the originally published model of Han and Reitz are discussed in order to provide a basis for the present modeling approach. A hybrid heat transfer model formulation relies on the $k\text{-}\zeta\text{-}f$ turbulence model which is capable of capturing turbulent stress anisotropy near wall and predicting heat transfer with more fidelity. 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. In particular, extension of the flame-wall interaction effects, usually considered only in the first near-wall cell, to some of wall-adjacent cells, is found to be important for capturing evolution of the wall heat fluxes and improving model performance with respect to mesh sensitivity.

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INTRODUCTION

While importance of numerical simulations in engine development is rapidly increasing, most of the research is focused on the combustion process. However, one of significant limitations in conversion of the resulting thermal energy into mechanical energy is the thermal loss through the combustor chamber walls. Reliable predictions of the wall heat transfer are essential for the overall accuracy of engine combustion simulations with respect to engine efficiency, exhaust emissions and component thermal stresses. Recent developments in the engine heat transfer modeling tend to improve existing wall heat transfer models (temperature wall functions) while typically employing standard or low-Reynolds number variants of $k\text{-}\epsilon$ turbulence model. At this point, one should recall that 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. The previous work pertinent to engine heat transfer modeling is scrutinized in the publications of Rakopoulos et al. [1] and Nuutinen et al. [2]. Rakopoulos et al. [1] 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 [3] was found to be the best compromise between simplicity and accuracy. Apart from variable density effects already observed by Han and Reitz [3] and

Angelberger et al. [4], Nuutinen et al. [2] incorporated 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 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. [4].

Due to prohibitive computational costs, 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. In such cases, accounting for the flame-wall interaction (wall quenching) only in the first near-wall cells becomes insufficient. Furthermore, most commonly used turbulence models are known to perform poorly in engine-relevant configurations such as impinging jets with heat transfer [5].

This work is based on more advanced, $k\text{-}\zeta\text{-}f$ turbulence model which allows integration to the wall, with incorporated molecular and wall-blocking modifications [6]. Consequently, the model is capable of capturing turbulent stress anisotropy near wall and predicting heat transfer with more fidelity. Adopting $k\text{-}\zeta\text{-}f$ and the 2-zones extended

coherent flame (ECFM3Z [7]) models to model turbulence and combustion, respectively, the focus will be on the heat transfer model of Han and Reitz [3] in the framework of hybrid wall treatment. Some ambiguities in the original publication that are relevant to the model implementation details are addressed as well. The hybrid wall treatment in AVL FIRE® [7] is extended to the temperature wall function of Han and Reitz. Predictive capability of the hybrid approach will be validated against the spark-ignition (SI) engine heat transfer measurements of Alkidas and Myers [8], taking into account the flame-wall interaction also in some of wall-adjacent cells.

TURBULENCE MODEL

The k - ζ - f (or simply 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 v^2 / k (v^2 is a scalar property in the Durbin's $v^2 - f$ model [9], 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 conditions for the elliptic function f . Hanjalić et al. [6] 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:

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = (P - \varepsilon) + \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] \quad (1)$$

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

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

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

with the wall boundary condition for f : $f_{wall} = \lim_{y \rightarrow 0} (-2\nu\zeta / y^2)$.

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

$$T = \max \left[\frac{k}{\varepsilon}, C_\tau \left(\frac{\nu}{\varepsilon} \right)^{1/2} \right] \quad (5)$$

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

$$L = C_L \max \left(\frac{k^{3/2}}{\varepsilon}, C_\eta \left(\frac{\nu^3}{\varepsilon} \right)^{1/4} \right) \quad (6)$$

Values of the coefficients appearing in the model equations are outlined in Table 1. Readers are referred to the original publications of Hanjalić et al. [6], Popovac and Hanjalić [10] and Basara [11] for more specific details about the model developments.

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

HYBRID WALL HEAT TRANSFER MODEL

In order to facilitate a proper model implementation, the published model of Han and Reitz [3] will be briefly discussed. Neglecting the tangential derivatives, pressure gradient, radiation heat transfer and other sources, under assumption of the ideal gas with constant properties, the authors integrated the simplified boundary layer equation for energy:

$$-\frac{\rho c_p u^*}{q_w} dT = \frac{1}{\left(\frac{1}{Pr} + \frac{v^+}{Pr_t} \right)} dy^+ \quad (7)$$

u^* being friction velocity and

$$y^+ = \frac{y u^*}{\nu}; \quad v^+ = \frac{V_t}{\nu} \quad (8)$$

Upon integration of the left hand side of Eqn. (7), the non-dimensional temperature profiles reads:

$$T^+ = \frac{\rho c_p u^* T \ln \frac{T}{T_w}}{q_w} \quad (9)$$

Based on numerous experimental data, the simplified expressions describing turbulent Prandtl number variation were used for the integration of the right hand side of Eqn. (7):

$$\frac{\nu^+}{Pr_t} = a + by^+ + cy^{+2} \quad y^+ < y_0^+$$

$$\frac{\nu^+}{Pr_t} = my^+ \quad y^+ > y_0^+$$
(10)

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. Splitting integration into two parts and neglecting Pr^{-1} in the second part of the integration

$$T^+ = \int_0^{y_0^+} \frac{1}{Pr^{-1} + a + by^+ + cy^{+2}} dy^+ + \int_{y_0^+}^{y^+} \frac{1}{my^+} dy^+$$
(11)

Han and Reitz derived the wall function formulation

$$T^+ = 2.1 \ln y^+ + 2.5$$
(12)

used to model the wall heat flux as

$$q_w = \frac{\rho c_p u^* T \ln \frac{T}{T_w}}{2.1 \ln y^+ + 2.5}$$
(13)

Note that the published model formulation was recommended for both turbulent and laminar regimes, i.e. irrespective of y^+ . Accordingly, among other authors, Rakopoulos et al. [1] in their evaluation of the existing heat transfer models, point out validity of the wall function of Han and Reitz for all y^+ . Actually, Eqn. (12) is strictly valid for fully turbulent region, or precisely only for $y^+ > y_0^+$. Consistent implementation of the model would require consideration of the first part of the integral for $y^+ < y_0^+$, i.e. the near-wall formulation of the temperature wall function:

$$T_{nw}^+ = \int_0^{y_0^+} \frac{1}{Pr^{-1} + a + by^+ + cy^{+2}} dy^+ = \frac{2}{\sqrt{4c(Pr^{-1} + a) - b^2}} \arctan \left(\frac{2cy^+ + b}{\sqrt{4c(Pr^{-1} + a) - b^2}} \right) \Big|_0^{y_0^+}$$
(14)

Careful inspection of the resulting wall function reveals that the log-law constant 2.5 in Eqn. (12) cannot be determined following the derivation described in the original publication. If the integration constant in the temperature wall function is determined by its intersection with the near-wall formulation (at $y^+=40$), it turns to be 1.234, thus being inconsistent with Eqn. (12). There are two possible explanations for this as depicted by Figure 1. The mismatch could originate from a typo regarding constant b (0.025 instead of -0.025),

which seems to be unlikely. Alternatively, the same integration constant 2.5 can be obtained by intersection of the logarithmic region with the viscous layer formulation ($y^+=10.63$), as well.

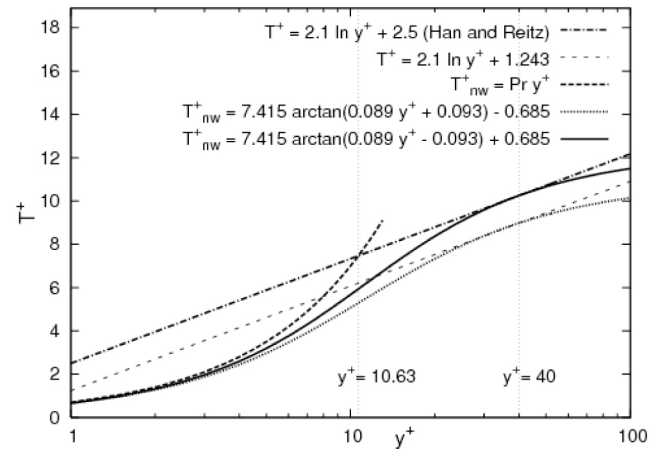


Figure 1. Various interpretations of the Han and Reitz wall model

The following two-layer formulation

$$T_{nw}^+ = 7.415 \arctan(0.089 y^+ - 0.093) + 0.685$$
(15)

will serve as a reference (Figure 3) for the model implementation in the framework of hybrid wall treatment. This method blends the integration up to the wall (exact boundary conditions) with the high-Reynolds number wall functions, enabling well-defined boundary conditions irrespective of the position of the wall-closest computational node. It is especially attractive for computations of industrial flows in complex domains where higher grid flexibility, i.e. weaker sensitivity against grid non-uniformities in the near-wall regions, featured by different challenging mean flow and turbulence phenomena (impinging, accelerating and decelerating flows, streamline curvature effects, separation, etc.), is desirable. Popovac and Hanjalic [10] proposed the so-called compound wall treatment with a blending formula (following the work of Kader [12]) for the quantities specified at the central node of the wall-closest grid cell as:

$$\phi_P = \phi_\nu e^{-\Gamma} + \phi_t e^{-1/\Gamma}$$
(16)

where ' ν ' denotes the viscous and ' t ' the fully turbulent value. The variable ϕ represents the wall shear stress, production and dissipation of the turbulence kinetic energy. A single expression for the non-dimensional temperature profile initially proposed by Kader [12] uses the following blending coefficient as a function of the normalized distance to the wall:

$$\Gamma = 0.01 \frac{(Pr y^+)^4}{1 + 5 Pr^3 y^+}$$
(17)

The evolution of laminar and turbulent contributions of the blending function for $Pr=0.7$ is illustrated in Figure 2