

Immersion Quenching Simulation of Realistic Cylinder Head Geometry

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ABSTRACT

In this paper, a recently improved Computational Fluid Dynamics (CFD) methodology for virtual prototyping of the heat treatment of cast aluminum parts, above most of cylinder heads of internal combustion engines (ICE), is presented. The comparison between measurement data and numerical results has been carried out to simulate the real time immersion quenching cooling process of realistic cylinder head structure using the commercial CFD code AVL FIRE®. The Eulerian multi-fluid modeling approach is used to handle the boiling flow and the heat transfer between the heated structure and the sub-cooled liquid. While for the fluid region governing equations are solved for each phase separately, only the energy equation is solved in the solid region. Heat transfer coefficients depend on the boiling regimes which are separated by the Leidenfrost temperature. The objective of the present research work is to present an update of the quenching model where instead of constant, variable Leidenfrost temperature is applied. Furthermore, simulation results are compared with available measurements results of different solid part orientations. The temperature histories predicted by the presented model correlate very well with the provided measurement data at different monitoring locations. In terms of methodology presented in previous publications both, measurement and simulation techniques have been further improved.

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1. INTRODUCTION

Modern powertrain development is driven in the direction of reducing the weight ratio by replacing heavier metals with light low-cost alloys to reduce fuel consumption and emissions values. The immersion quenching cooling process, among all other heat treatment techniques, has been long identified as one of the most important ways to achieve the desirable microstructure and mechanical properties of the metal piece. This process provides even temperature distribution during the cooling process, leads to reduction of the residual stress levels, and consequently, prevents distortion and cracking of the cast parts.

In the immersion quenching process, a metal piece is heated up to a microstructure dependent temperature, stays there for a while, and is then immediately submerged into a sub-cooled liquid, such as water, oil or polymers. Therefore, all three boiling regimes appear during the immersion quenching cooling process. The film boiling regime occurs immediately after the heated piece is dipped into the sub-cooled liquid domain. The heat transfer of the cooling rate in this regime is relatively small, since the film is stable and the vapor blanket acts like an insulator. As soon as the surface temperature of the heated piece drops below the minimum film collapse temperature, also known as the Leidenfrost temperature, transition boiling starts, and the hot surface is partly wetted due to the collapse of the vapour film. With the time processing, the hot surface temperature drops and the heat flux increases.

When the wall heat flux reaches the maximum heat flux, or the so-called critical heat flux, the nucleate boiling regime is entered. There the vapor film becomes unstable and starts to disappear. This results in a higher heat transfer rate between the metal and the fluid, and leads to faster cooling. As soon as the surface temperature of the heated part drops below the boiling temperature of the fluid, the convective boiling regime occurs, and the cooling flow rate becomes stable and low.

In recent years, many analytical approaches for immersion quenching have been investigated and phenomena such as vapor pocket generation, stagnation region, flow dynamics etc. have been successfully resolved by several researchers [1, 2, 3, 4, 5]. Based on the complexity and phenomenon of multi-phase flows in immersion quenching applications, heat and mass transfer empirical models were proposed and developed. Krause *et al.* [5] employed the Eulerian multiphase model to simulate boiling flow, heat and mass transfer during the quenching process. The derived flow boiling model was based on a mixture model assumption and bubble crowding model approach for the two phase flow. Similar model for mass transfer prediction was proposed by Wang *et al.* [6], who assumed that the mass transfer rate due to boiling is proportional to the heat transfer rate in the fluid system. The model was later improved by Srinivasan *et al.* [1], and it has been already implemented into the commercial CFD code AVL FIRE® where it is applied for the numerical simulation of quenching process [1, 2, 3].

A new approach for modeling the variable Leidenfrost temperature threshold is presented in this work. As long as the surface temperature is above the Leidenfrost threshold, film boiling happens, while transition boiling occurs, when the surface temperature drops below the Leidenfrost temperature. Thus, the Leidenfrost threshold is a very important parameter for the numerical simulation of the quenching process, since it determines which regime is responsible for the calculation of the heat transfer coefficient. In previous published works [1, 2, 3, 4, 5, 6, 7], a constant Leidenfrost temperature threshold is assumed over the entire domain. This assumption has only limited validity, as discussed in the experimental work of Lübben *et al.* [8]. It can be summarized that the Leidenfrost temperature for the quenching application depends on several factors, such as the pool temperature (sub-cooling effect) [9], dipping and wetting velocity of the heated work piece [10], geometry and shape of the work piece, system pressure [11], surface roughness [12], etc. From previous work [13], it can be observed, that due to rising vapor from the bottom to the top of the quenched work piece, there is a higher vapor concentration on the top. This leads to the effect that transition boiling occurs at a lower temperature level, and consequently the Leidenfrost temperature is lower at the upper part of the quenched surface. Therefore, a simple new model for spatially changing Leidenfrost temperature is presented in this paper. The aim of this approach is to achieve better agreement between the predicted results and the available measurement data.

The paper is organized as follows. In [section 2](#) the physical model based on the previous work of Srinivasan *et al.* [1] is described. Two new models, lift force and wall lubrication force models, have been implemented for the interfacial momentum exchange between vapor and liquid. These models are discussed in [section 2](#). A brief introduction of the simulation set-up is performed in [section 3](#). Simulations of the heat treatment of cast aluminum parts, i.e. cylinder heads of internal combustion engines (ICE), are presented in this section. Two different quenching orientations have been conducted. The comparisons of the simulated results of a real-time quenching process with the available measurement data are presented and discussed in [section 4](#).

2. MATHEMATICAL MODEL

The Eulerian multi-fluid model considers each phase as interpenetrating continua coexisting in the flow domain, with inter-phase transfer terms accounting for phase interactions where conservation laws apply [14]. From the theoretical work of Drew *et al.* [15] the ensemble averaged continuity and momentum equations are presented as:

2.1. Continuity Equation

$$\frac{\partial \alpha_k \rho_k}{\partial t} + \nabla \cdot (\alpha_k \rho_k \mathbf{v}_k) = \sum_{l=1, l \neq k}^N \Gamma_{kl}, \quad k = 1, \dots, N \quad (1)$$

subjects to the compatibility condition

$$\sum_{k=1}^N \alpha_k = 1, \quad (2)$$

where α , ρ and $\mathbf{v}$ stand for volume fraction, density and velocity vector. The phase change rate (in this particular case, boiling) is Γ_k and the subscript k is the phase indicator ($k = l$ or $k = v$).

2.2. Momentum Equation

$$\frac{\partial \alpha_k \rho_k \mathbf{v}_k}{\partial t} + \nabla \cdot (\alpha_k \rho_k \mathbf{v}_k \mathbf{v}_k) = -\alpha_k \nabla p + \nabla \cdot \alpha_k \boldsymbol{\tau}_k + \alpha_k \rho_k \mathbf{g} + \sum_{l=1, l \neq k}^N M_{kl} + \mathbf{v}_{int} \Gamma_k, \quad (3)$$

with p , $\mathbf{g}$, $\boldsymbol{\tau}$ and $\mathbf{v}_{int}$ are respectively the pressure, gravity vector, stress tensor and interfacial velocity. Interfacial momentum term is given by

$$M_{kl} = F_D + F_{WL} + F_L, \quad (4)$$

where F_D indicates the drag force, F_{WL} the wall lubrication force and F_L the lift force.



2.2.1. Drag Force

The interfacial momentum transfer term for drag force is given as

$$F_D = C_D \frac{1}{8} \rho_c A_{\text{int}} |\mathbf{v}_r| \mathbf{v}_r, \quad (5)$$

where C_D is the drag coefficient, and A_{int} stands for the interfacial area density. The relative velocity $\mathbf{v}_r$ is defined as:

$$\mathbf{v}_r = \mathbf{v}_v - \mathbf{v}_l. \quad (6)$$

Subscripts v and l denote the vapor and liquid phases in the given flow. The drag coefficient C_D is given by

$$C_D = \begin{cases} \frac{24}{\text{Re}_b} (1 + 0.15 \text{Re}_b^{0.678}) & \text{Re}_b \leq 1000 \\ 0.438 & \text{Re}_b > 1000 \end{cases}, \quad (7)$$

where the bubble Reynolds number is defined as:

$$\text{Re}_b = \frac{\mathbf{v}_r D_b}{\nu}. \quad (8)$$

2.2.2. Lift Force

When the continuous phase field is non-uniform or rotational, the bubbles experience the lift force perpendicular to the relative velocity. This force F_L is described as:

$$F_L = -\alpha_v \rho_l C_l (\tilde{\mathbf{v}}_g - \tilde{\mathbf{v}}_l) \times (\nabla \times \tilde{\mathbf{v}}_l). \quad (9)$$

Tomiyama's lift model is employed to determine the lift coefficient as described in [16]:

$$C_L = \begin{cases} \min[0.288 \tanh(0.121 \text{Re}_g), f(Eo_d)] & Eo_d < 4 \\ f(Eo_d) & 4 \leq Eo_d \leq 10 \\ -0.27 & Eo_d > 10 \end{cases}. \quad (10)$$

The function of the bubble Eötvös number is given by

$$f(Eo_d) = 0.00105 Eo_d^3 - 0.0159 Eo_d^2 - 0.0204 Eo_d + 0.474, \quad (11)$$

and the modified Eötvös number Eo_d based on a deformable bubble is defined as

$$Eo_d = \frac{Eo}{E^2} = Eo \cdot (1 + 0.163 \cdot Eo^{0.757})^2, \quad E = \frac{d_h}{d_b}, \quad (12)$$

where E denotes the empirical correlation of the aspect ratio and d_v maximum vertical dimension of the bubble.

2.2.3. Wall Lubrication Force

The primary effect of the wall lubrication force is to push the bubbles away from the wall. This assures the experimentally found zero void condition near vertical walls, [17]. It is described by

$$F_{WL} = \alpha_v \rho_l C_{WL} \left\{ (\tilde{\mathbf{v}}_v - \tilde{\mathbf{v}}_l) - [(\tilde{\mathbf{v}}_v - \tilde{\mathbf{v}}_l) \cdot \tilde{\mathbf{n}}_w] \tilde{\mathbf{n}}_w \right\}^2 \tilde{\mathbf{n}}_w, \quad (13)$$

where $\tilde{\mathbf{n}}_w$ is the unit wall normal vector pointing from the wall into the fluid, and C_{WL} stands for wall lubrication coefficient. Based on Tomiyama's work [16], Frank *et al.* [17] proposed an extended and generalized wall lubrication force model for the coefficient C_{WL} , described by

$$C_{WL} = C_w(Eo) \max \left[0, \frac{1}{C_{w,D}} \frac{1 - \frac{y_w}{C_{w,c} d_b}}{y_w \left(\frac{y_w}{C_{w,c} d_b} \right)^{p-1}} \right], \quad (14)$$

where exponent $p = 1.7$, $C_{w,c} = 10$ and $C_{w,d} = 6.8$ are model coefficients. According to Tomiyama [16] C_w is a function of the Eötvös number, and it yields

$$C_w(Eo) = \begin{cases} e^{-0.933 Eo + 0.179} & 1 \leq Eo \leq 5 \\ 0.007 Eo + 0.04 & 5 \leq Eo \leq 33 \\ 0.179 & 33 < Eo \end{cases}. \quad (15)$$

The effects of the aforementioned forces are presented in [Figure 1](#). It can be seen that the vapor bubbles are pushed away from the heated surface ([Fig. 1](#), right), which results in a faster cooling rate. As the Euler-Euler model is used, vapor bubbles are treated as dispersed phase. Under this assumption, both lift force and wall lubrication force are applied. Vapor film is assumed as vapor phase with a volume fraction of 1 or less. If the vapor volume fraction is higher than 0.8, than the wall lubrication and the lift force are neglected. Detailed information concerning modeling the interfacial forces is described in the work of Kopun *et al.* [18].