THERMAL INTERACTION MODEL BETWEEN A FLUID FLOW AND A SOLID

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***Abstract*— A thermal interaction model between a turbulent fluid flow and a solid was developed. The heat exchange is modeled by means of a Newton cooling law where the convective heat transfer coefficient is calculated using thermal wall laws. The finite element method was employed to solve the conservation equations. The model gives the possibility that the fluid and solid meshes are not connected; this generates a great flexibility in meshing and in geometry modification. The domains coupling algorithm was verified using simple problems. Finally, the model developed was validated and applied successfully to the simulation of the fluid dynamic thermal behavior of hot dip galvanizing bath.**

***Keywords*— fluid dynamic, thermal, finite element method, coupled problem, continuous galvanizing.**

I. INTRODUCTION

Numerical fluid dynamics is nowadays a powerful and reliable tool for simulating different thermo-fluid dynamic processes. Hence, it permits to analyze different operative variables and geometrical configurations to investigate technological windows of different processes in metallurgical industry. In some cases, the industrial process involves moving solid contours like rotating cylinders or circulating strips. This solid contours exchange momentum and heat with the surrounding fluid. In this paper a thermal interaction model between a fluid flow and a solid was presented.

In section II the governing equations and the used hypothesis are presented. In section III the numerical scheme used to solve the equations and the coupling method are described. The model was validated with simple cases. In section IV The model was validated and applied to the galvanizing process. Finally, in section V the conclusions are presented.

II. GOVERNING EQUATIONS

A. Turbulent fluid dynamic – thermal model

In order to obtain the field of velocities, pressures and temperature in a turbulent incompressible fluid flow the equations of Navier Stokes and energy using the Boussinesq approximation are solved.

$$\nabla \cdot \mathbf{v} = 0 \tag{1}$$

$$\rho_f \frac{\partial \mathbf{v}}{\partial t} + \rho_f \mathbf{v}.\nabla\mathbf{v} - \nabla \cdot \left[\left(\mu + \mu^t\right)\left(\nabla\mathbf{v} + \nabla\mathbf{v}^T\right) \right] \ldots \tag{2}$$
$$+\nabla P + \mathbf{F}_{bth} = \mathbf{0}$$

$$\rho_f Cp_f \left(\frac{\partial T}{\partial t} + \mathbf{v}.\nabla T_f \right) - \nabla \cdot \left[\left(\lambda_f + \lambda^t\right)\nabla T_f \right] + Q_f = 0 \tag{3}$$

$$\lambda^t = \frac{\mu^t \, Cp_f}{\Pr^t} \tag{4}$$

Where ρ_f is the fluid density, μ_f is the fluid laminar viscosity, μ^t is the turbulent viscosity, **v** is the time averaged velocity, p is the time averaged pressure, $\mathbf{F}_{bth} = \rho\,\mathbf{g}\,\beta_{th}\left(T - T^{ref}\right)$ is the buoyancy force, **g** is the acceleration due to gravity, β_{th} is the thermal expansion coefficient, Cp_f is the fluid specific heat, λ_f is the fluid thermal conductivity and $\Pr^t$ is the Prandtl turbulent number.

The mathematical description of the turbulent flows using mean quantities equations makes necessary the use of turbulence models to close the problem. For industrial problems modeling the mixl model or the k-ε model (Launder and Spalding 1974) are commonly used.

Due to the turbulence models cannot solve the flow in the zone near the solid contours, the wall functions method (Launder and Spalding 1974) is used. The finite element mesh is located at a wall distance Δ_{wall}. The friction velocity u^* is calculated solving the nonlinear equation

$$\frac{v_x}{u^*} = \frac{1}{\kappa}\ln\left(y^+ \; E\right) \qquad y^+ = \frac{\rho\, y\, u^*}{\mu} > 11.63 \tag{5}$$

And then, τ_w is applied in the corresponding fluid node.

$$\tau_w = \rho\, u^{*2} \tag{6}$$

In order to solve the thermal coupled problem the boundary layer temperature profile is considered to transfer the boundary condition as in the case of velocities. A similarity between the velocity profile and the temperature profile is assumed. The adimensional temperature is obtained like

$$T^+ = \begin{cases} \Pr y^+ & y^+ < y_0^{\theta+} \\ \sigma^\theta \left[\frac{1}{k}\ln y^+ + P_T\right] & y^+ > y_0^{\theta+} \end{cases} \tag{7}$$

$$P_T = \frac{1}{\sigma^\theta}\left[\Pr y_0^{\theta+} - \frac{\sigma^\theta}{\kappa}\ln y_0^{\theta+}\right] \tag{8}$$

Then, knowing the solid wall temperature T_s a heat flow is applied $Q = h\left(T_s - T_f\right)$, where $h = \rho\, Cp\, u_* / T^+$ is the convective heat transfer coefficient.

B. Solid thermal model

In order to obtain the field of temperatures in the solid,