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Numerical Simulation of Heat and Mass Transfer in a Natural Draft Wet Cooling Tower

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Abstract

A 2D axisymmetric two-phase simulation of the heat and mass transfer inside a natural draft wet cooling tower using the commercial CFD package *Fluent* has been conducted. The water droplets in the spray and rain zones are represented with droplet trajectories written in Lagrangian form with heat and mass transfer coupled with the continuous phase. The heat and mass transfer in the fill is represented using source terms controlled through user defined subroutines. These functions solve basic heat and mass transfer equations with the transfer coefficients written in terms of the Poppe equations. The functional dependence of the transfer coefficients are included in the empirical relationship. The model has the capability to model non-uniformities in fill layout and water distribution, which traditional analytical and empirical models are unable capture.

Introduction

Natural draft wet cooling towers (NDWCT) are commonly used in thermal power stations to cool the condenser feed water. Improving the performance of these structures can reduce turbine back-pressure and improve generation efficiency. Even a small reduction in cooling water temperature can result in a large savings in fuel costs. This leads to a desire to understand the operation of the cooling tower and to optimise its design parameters.

Current design procedures still make use of the traditional 1D analytical cooling tower models [1], relying on extensive empirical relationships for loss coefficients and transfer coefficients. The most widely used 1D model is the original Merkel model [1]. By dropping a few terms in the heat balance Merkel was able to reduce the driving for heat and mass transfer down to an enthalpy difference, allowing the calculation to be performed by hand. Poppe [1, 2, 3] later proposed a complete and more accurate set of equations accounting for the eveporation of water. Either of these models can be quite accurate and incorporate a great deal of detail, however they cannot capture the non-uniformities through the tower. Numerical models are required to assess the effects of major design layout changes [1].

The model developed here attempts to bring together the two approaches with the goal of understanding the limitations of the traditional design process and to optimise the cooling tower design with respect to these non-uniformities. This desire for a comprehensive comparison has influenced the development of the model, in particular the representation of the fill.

No numerical models reported on to date explicitly model the fill, instead researchers have employed source terms to model the effect of the fill on the continuous phase [4, 5, 6, 7]. Recently numerical models of NDWCTs have been developed using this approach supported with a variety of experimental data. Fournier and Boyer [4] presented a proprietary code with the capability to employ either Merkel or Poppe [1] transfer coefficients. Hawlader and Liu [5] developed a 2D axisymmetric model employing the Merkel fill characteristics.

Recently Kloppers and Kröger [8, 2] suggested a new empirical form of the equations for fill loss coefficients and fill transfer coefficients with full functional dependence. These have been found to provide a much better fit to experimental data than traditional forms of the equations and are valid under a wider range of operating conditions. This is essential for cooling tower modeling where the dependent conditions can vary considerably from the tower center to the tower outer edge.

Most previous studies have been performed on very coarse grids and have employed an algebraic turbulence model [5, 6, 7]. This study presents a more detailed model of the cooling tower fill within a commercial CFD package *Fluent*. It is the future goal of this investigation to create a numerical model and use it to optimise the design of cooling towers by varying the fill depth and water flow rate in the tower.

Fluent Numerics

Fluent is a general purpose CFD code. The code has been used to solve the steady Reynolds Averaged Navier-Stokes Equations closed with the standard k-epsilon turbulence model with buoyancy terms included. The semi-implicit method for pressure linked equations (SIMPLE) was employed with second order upwind discretisation employed for the spatial derivatives. A segregated implicit solver was used.

A cooling tower is a cylindrical structure so a 2d axisymmetric steady representation is valid under no wind conditions. This representation was used to reduce computational requirements. The steady 2D axisymmetric momentum and transport equations can be written in general form as follows:

$$\nabla \cdot (\rho u \phi - \Gamma_\phi \nabla \phi) = S_\phi \tag{1}$$

where (ρ) is the density, (u) is the velocity vector, (ϕ) is the flow variable (k, ε, ω (h_2o species concentration), u, v, h (enthalpy)). Γ_ϕ is the diffusion coefficient and S_ϕ is the source term.

The humid air-water mixture is taken to be an ideal gas and an incompressible fluid. The density is computed using,

$$\rho = \frac{p_{\mathrm{op}}}{\frac{R}{M_w} T} \tag{2}$$

where p_{op} is the user specified constant operating pressure for the entire domain and is set at 101kPa.

The buoyancy term in the momentum equation is given as $(\rho - \rho_0)g$, where g is the gravitational acceleration and ρ_0 is the constant operating density specified at the inlet fluid conditions of 295k and 50% humidity.

Simulation

The geometry of the tower is that of the unit 1 cooling tower at Mt. Piper Power Station in Lithgow NSW. The computational domain is discretised with approximately 250,000 2D unstructured mesh elements. The computational domain extends 90

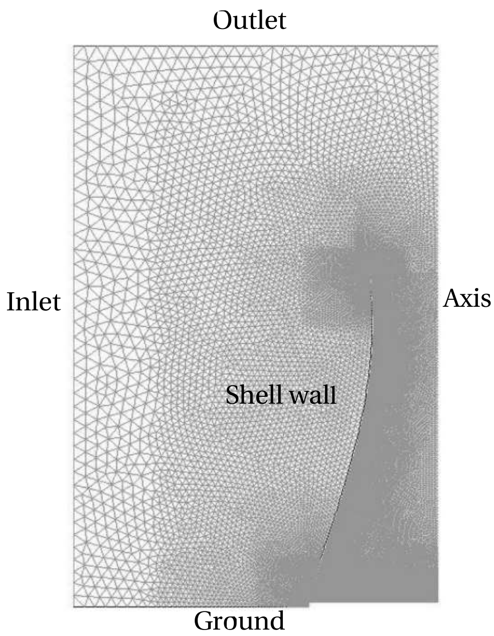

Figure 1: Computational domain with bounrdy conditions.

meters beyond the cooling tower inlet and 90 meters above the cooling tower outlet allowing for the determination of the inlet velocity profile and the effect of the plume. The tower has a total water flow rate of 17,000kg/s at 313K. The tower has a height of 131m and a base diameter of 49m.

Modelling

In a NDWCT there are three heat and mass transfer regions that need to be modeled, the spray region below the spray nozzles, the fill (with compact film type fill) and the rain region below the fill.

Droplet Modelling

In the spray and rain regions the water flows in droplet form. This has been represented with Lagrangian particle tracking with coupled heat and mass transfer between the droplets and the continuous phase. Most previous models [4, 5, 6, 7] assumed the 1D motion of the droplets.

The change in droplet temperature and mass are found through equations (3) and (4), where N_i, the molar flux of mass and h the heat transfer coefficient are evaluated through empirical correlations [9]. Variables m_p, c_p, A_p, T_p are the particle mass (kg), specific heat (J/kgK), surface area (m^2) and temperature (K) respectively. h_{fg} is the latent heat of vapourisation (J/kg)

$$m_p c_p \frac{dT_p}{dt} = hA_p(T_\infty - T_p) + \frac{dm_p}{dt} h_{\text{fg}} \tag{3}$$

$$m_p(t+\Delta t) = m_p(t) - N_i A_p M_{w,i} \Delta t \tag{4}$$

The energy (Q) and mass (M) transfer are coupled with the continuous phase through equations (5) and (6) [9].

$$Q = \left[\frac{\overline{m}_p}{m_{p,0}} c_p \Delta T_p + \frac{\Delta m_p}{m_{p,0}} \left(-h_{\text{fg}} + \int_{T_{\text{ref}}}^{T_p} c_{p,i} dT \right) \right] \dot{m}_{p,0} \tag{5}$$

$$M = \frac{\Delta m_p}{m_{p,0}} \ \dot{m}_{p,0} \tag{6}$$

$m_{p,0}$, $\overline{m}_p$ and $\dot{m}_{p,0}$ are the initial mass of the particle, the average mass of the particle in the cell and the initial mass flow rate of particles in the trajectory respectively.

Lagrangian particle trajectories are initiated from spray nozzle locations. At the surface of the fill these droplet trajectories are terminated and the droplet temperature and mass flow rate are stored. In the rain region the droplets are initiated from the center of each face on the bottom surface of the fill. The temperature and water mass flow rate of the droplets are determined by the subroutine that describes the heat and mass transfer on the fill. The droplets are given a uniform distribution of 2.5mm in the rain zone.

Fill Modelling

In the fill, the water flows in complex film type motion across the closely packed parallel wavy plates in the counter direction to air flow. It would be computationally prohibitively expensive to model the fill explicitly so the effect of the fill on the continuous phase is represented using source terms. The change in water temperature is calculated through the fill using a user defined subroutine which tracks the water properties through the fill to balance the heat and mass transfer to the continuous phase.

The water flow through the fill is physically one-dimensional as it is constrained to film flow descending along the vertical plates. This requires that the heat and mass transfer in the fill to also be a 1 dimensional process. This simplification allows the water flow to be represented solely by two variables at each point, its temperature and mass flow rate. The fill region in the tower is considered as a number of discrete columns, each one being equivalent to a 1D grid, overlaying the computational domain. Across each layer in these columns, or between points on the 1D grid, the change in water temperature and mass are computed based on the traditional analytical methods. This approach is depicted in figure (2). The water flow through the tower fill is represented by 87 of these columns with each one discretised into 10 layers or nodes.

Momentum Source Terms

The pressure loss through the fill is modeled using source terms in the momentum equation. This momentum sink is given as:

$$S_v = -K_{fi} \times \frac{\rho_m V^2}{2} \tag{7}$$

where K_{fi} is the fill loss coefficient per meter depth of fill. The empirical correlation for K_{fi} is expressed as a function of the air (m_a) and water (m_w) flow rates through the fill and the depth of the fill (L_{fi}) (equation 8) as described by Kloppers [2].

$$\begin{aligned} K_{fdm1} \quad = \quad & (5.154914 m_w^{0.877646} m_a^{-1.462034} \\ & + 10.806728 m_w^{0.226578} m_a^{-0.293222}) \\ & \times L_{fi}^{-0.236292} \end{aligned} \tag{8}$$

The pressure loss due to the cooling tower shell supports, the water distribution piping network and the drift eliminators was modeled in a manner similar to the fill. The loss coefficients used were $K_{cts} = 0.5$, $K_{wdn} = 0.5$ and $K_{de} = 3.5$ respectively as taken from [1].

Heat and Mass Transfer in the Fill

The heat and mass transfer characteristics are governed by the volumetric mass transfer coefficient and the wetted contact area between the phases. The product of these two values $h_d A$ can be found from the Merkel number [1] for a particular fill type. The transfer coefficients used are in the Poppe form, which means that the Poppe equations [3] are used to interpret the experimental data and form the empirical equation for the coefficient.

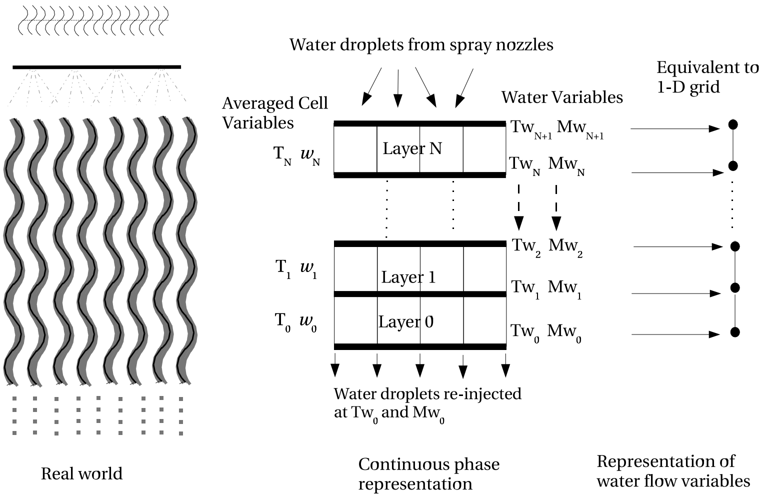

Figure 2: Incremental control volume of the fill.

Kloppers [2] gives a detailed analysis of the Poppe model. The methods used to derive the coefficients are replicated here to find the change in water properties.

The transfer coefficient used here is written in terms of inlet flow rates for water and air as shown in eqaution (9). The functional dependence on air wet and dry bulb inlet temperature was shown by Kloppers [2] to be insignificant so this dependency can be excluded. Although the coefficient is a function of both water inlet temperature and fill depth [2], these are both held consant in this investigation so a more general relation was not sought.

$$\frac{Me_{Poppe}}{L_{fi}} = \frac{h_d A}{m_w} = 1.380517 m_w^{0.112753} m_a^{0.698206} - 0.517075 m_w^{0.461071} m_a^{0.681271} \quad (9)$$

The heat transfer coefficient can then be found using the Lewis factor relationship given in equation (10).

$$Le_f = \frac{h}{h_d C_{pa}} \quad (10)$$

where h_d is the mass transfer coefficient for the fill with units $kg/m^2 s$ and h is the heat transfer coefficient for the fill with units $w/m^2 k$. The Lewis factor is determined using Bosjnakovics formula [10] given in equation (11). Under saturation conditions, the formula is modified to equation (12).

$$Le_f = 0.865^{2/3} \cdot \frac{\left(\frac{\omega_{sw}+0.622}{\omega+0.622} - 1\right)}{ln\left(\frac{\omega_{sw}+0.622}{\omega+0.622}\right)} \quad (11)$$

$$Le_{f,sat} = 0.865^{2/3} \cdot \frac{\left(\frac{\omega_{sw}+0.622}{\omega_{sa}+0.622} - 1\right)}{ln\left(\frac{\omega_{sw}+0.622}{\omega_{sa}+0.622}\right)} \quad (12)$$

Coupling Procedure

The calculations across each layer take place between the averaged continuous phase flow variables and the upstream (with respect to water flow) water flow variables. The heat and mass transfer from the water phase is computed first and then the energy and mass source terms which balance this change in water temperature and mass flow rate are evaluated. The source terms calcualted are identical for all the cells across the layer. The maximum column width in the model is 1m, the smallest being 0.1m.

The water evaporated $m_{evap}[n]$ across fluid layer n is determined using equation (13), where $\omega_{sat,Tw}$ is the specific humidity of saturated air evaluated at the water temperature (kg/kg) and $\omega_{ave,fluid}$ is the average specific humidity in the fluid zone.

$$m_{evap} = h_d A(\omega_{sat,Tw} - \omega_{ave,fluid}) \quad (13)$$

The volumetric transfer coefficients are specified per meter depth of the fill and the heat and mass transfer is being evaluated across a small increment in the fill ΔL_{fi}, so the transfer coefficients must be reduced to allow for the smaller area over which the heat transfer is taking place using,

$$h_d \cdot A = (h_d \cdot A)_{calc} \cdot \Delta L_{fi} \quad (14)$$

The new water mass flow rate is found using equation (15).

$$m_w[n] = m_w[n+1] - m_{evap}[n] \quad (15)$$

The latent and sensible heat transfer is evaluated using equations (16) and (17) respectively.

$$q_{latent} = m_{evap} \cdot h_{fg} \quad (16)$$

$$q_{sensible} = hA \cdot (Tw[n+1] - T_{ave,air}) \quad (17)$$

where $T_{ave,air}$ is the average temperature of the continuous phase in the layer. The water temperature at the inter-facial layer n is determined using equation (18).

$$Tw[n] = Tw[n+1] - \frac{\left(q_{sensible}[n] + q_{latent}[n]\right)}{C_{pw} m_w[n]} \quad (18)$$

where $Tw[n+1]$ is the water temperature corresponding to the fluid boundary above the fluid layer n, m_w is the mass flow rate of water in the column in kg/s, C_{pw} is the specific heat of water J/kgK.

When the flow becomes super-saturated then additional energy is released in the flow, as the latent heat of vapourisation is released when the water vapour condenses as mist. It has been

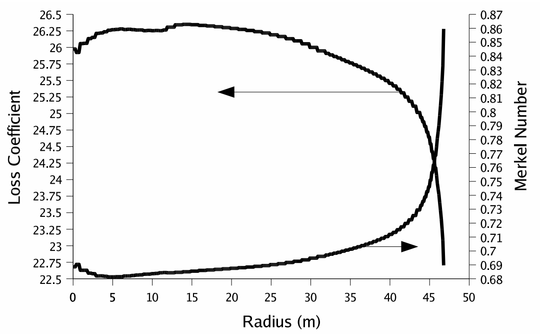

Figure 3: Merkel number and loss coefficient with radial location in the fill.

assumed for this investigation, as in the Poppe model [3], that vapour condenses as mist when the vapour pressure rises above the saturation vapour pressure although in reality it may reach very high levels of supersaturation before this occurs.

The mass source M_{source} (kg/m^3s) and enthalpy source Q_{source} (W/m^3) per unit volume are given by equations (19 and 20):

$$M_{source} = \frac{m_{evap}}{\Delta L_{fi}} \tag{19}$$

$$Q_{source} = \Big(m_w \cdot C_{pw} \cdot \Delta T_w + m_{evap} \cdot (C_{pv} \cdot (T_w - T_{ref}) - h_f g)\Big)/\Delta L_{fi} + m_{condense} \cdot h_{fg} \tag{20}$$

where C_{pw} is the specific heat of water and has units J/kg, (h_{fg}) and (C_{pv}) are the latent heat of vapourisation and the specific heat of saturated water vapour respectively.

Preliminary Results

The implementation of the cooling tower fill subroutine was validated against the traditional analytical models used and also against the experimental data obtained from Kloppers [2]. Full validation of the model has not yet been performed so the following results are therefore deemed preliminary.

The results indicate a significant deviation from the assumptions of 1D analytical models. A radial cut through the tower exhibits significant temperature, velocity, h_2o species concentration, and pressure gradients. Errors in the determination of the outlet condition will lead to erroneous computation of the tower draft and therefore heat transfer through the fill.

The Merkel number varys between 0.85 to 0.75 from the center to the outer edge of the tower as depicted in figure (3). Loss coefficients also vary throughout much of the tower, from 22.6 in the center to 24 near the outer edge where there is a low air to water flow rate ratio.

These results are of course numerically obtained and therefore do not represent any blockages or non-uniformities in the fill. Some published data [11] indicates that these non-uniformities can partially eliminate any variation in the temperature and humidity profile across the fill.

Conclusions

A commercial package can be successfully implemented with user defined subroutines to model a natural draft wet cooling tower. Preliminary results show room for improvement in cooling tower design and highlight the non-uniformities that exist in the fill inlet conditions. The future goal of this work is to quantify the effect of these non-uniformities on the accuracy of traditional cooling tower design and specification and to also determine where improvements can be made in cooling tower design.

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