# CFD Modelling of liquid film evaporation in regime of mixed convection

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**Abstract**

This work deals with numerical modelling of heat and mass transfer in mixed convection with evaporation from a liquid film in a rectangular channel. A numerical model to simulate the evaporation from the thin horizontal liquid film was adopted and developed further. The presented numerical model treats the temperature and vapour concentration as active scalars**,** was implemented into the commercial code Star-CCM+ and its validity is assessed based on the comparison with the experimental data. The experimental data of evaporation from water film were measured in low speed wind tunnel. Evaporation rate is evaluated based on conservation of mass and change of airstream's specific humidity. Presented numerical approach shows good agreement with conducted experiment.

*Keywords:* heat and mass transfer; liquid film; evaporation; CFD; experiment

## 1. Introduction

In the recent years, many researchers have been trying to solve the problems related to evaporation and condensation from thin liquid films. Despite the fact that various commercial CFD software offer models able to treat the evaporative process, its investigation is still topical in the academic field. Natural convection in presence of water evaporation and condensation plays a crucial role in a number of natural and industrial processes. A realistic and accurate numerical reproduction of such phenomena is a challenging problem from both numerical and theoretical perspectives. Moreover, from a numerical point of view, in most applications, flows cannot be considered laminar and an accurate reproduction of turbulence is required. The Boussinesq approximation for incompressible flow, cannot be applied in systems with high temperature and humidity differences.

The study of Yan and Tsay [1] is focused on the numerical investigation of laminar steady mixed convection flow between two vertical parallel plates covered with fluid film. In their work the fluid film is treated as a boundary condition, and it is defined as a Dirichlet boundary condition on the air-fluid interface. To enhance the natural convective flow, the Boussinesq approximation is introduced, $Gr_T$ and $Gr_M$, the *Grashof numbers* for *heat* and *mass transfer*, respectively, are implemented in axial-momentum equation. A similar approach can be seen in the study of Laaroussi [2] and Sosnowski [3].

In the work of Sosnowski on the air-fluid film interface the Dirichlet boundary condition is defined and the Boussinesq approximation reflecting both, solutal and thermal buoyancy effects is implemented into the incompressible form of Navier-Stokes equations. The change of the thickness of the fluid film in time is evaluated using the velocity of evaporation.

Concluding foregoing, in this study the evaporation from the horizontal liquid film inside a rectangular channel is investigated numerically based on the mathematical model used in [3] and its validation is assessed against the experimental results. This approach will be denoted as *two active scalars* based model, where solutal and thermal buoyancy effects are implemented into momentum equations, the air-fluid film interface is defined as a Dirichlet boundary condition and evaporation rate will be monitored by the *Boundary Species Flux* field function defined in Star-CMM+ code. The subject of this work is very interesting and actual, and yet very challenging. The chosen approach corresponds to the present state of knowledge.

## 2. Mathematical model

In order to solve the evaporation from a horizontal liquid film different physical phenomena have to be modelled by a suitable set of mathematical equations: natural convection in the fluid domain; the heat and mass transfer between the liquid phase and the surrounding gas through change of phase; and the exchange of heat between wetted solid bodies and the thin liquid film or drops laying on their surface. The latter one will not be treated in the present work.

### 2.1. Governing equations

The gas phase is governed by the incompressible Navier–Stokes equations with the Boussinesq approximation (3) to account for buoyancy effects [3].

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

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$$\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_j}{\partial x_j} = -\frac{1}{\rho_0}\frac{\partial p}{\partial x_i} + \nu \frac{\partial^2 u_i}{\partial x_j \partial x_j} - \\ -\frac{\rho}{\rho_0} g \delta_{1,2} \qquad (2)$$

$$\frac{\rho}{\rho_0} = 1 - \beta_T (T - T_0) - \beta_w (\omega - \omega_0) \qquad (3)$$

where $u$ is the fluid velocity, $p$ is pressure, $T$ is temperature, $\rho$ is space-time variable density in the fluid flow, $\omega$ is water vapour concentration. The zero subscript denotes the reference value of the corresponding variable. The vapour concentration is defined as:

$$\omega = \frac{m_v}{m_v + m_a} \qquad (4)$$

in which $m_v$ and $m_a$ are the vapour and air masses, respectively.
Temperature and vapour concentration in the fluid medium are both modelled as active scalars. They are diffused and transported by air according to the advection-diffusion equations as follows:

$$\frac{\partial T_a}{\partial t} + u_j \frac{\partial T_a}{\partial x_j} = \alpha_a \frac{\partial^2 T_a}{\partial x_j \partial x_j} \qquad (5)$$

$$\frac{\partial \omega}{\partial t} + u_j \frac{\partial \omega}{\partial x_j} = \alpha_\omega \frac{\partial^2 T_a}{\partial x_j \partial x_j} \qquad (6)$$

The turbulence modelling is based on the constant value of turbulent Prandtl number and turbulent Schmidt number.

### 2.2. Air-fluid film interface definition

The air-fluid film interface definition is treated similarly as in the work Yan and Tsay [1] and Sosnowski [3]. From the theory of evaporation, it is assumed a thin saturated layer of moist air above a fluid film, in which moist air and a liquid film are in thermodynamic equilibrium in this layer. The interfacial concentration of water vapour can be evaluated as follows [3]:

$$\omega_v = \frac{M_v}{M_a} \frac{p_v''(T_{ref})\varphi}{\left[p_{atm} - \left(1 - \frac{M_v}{M_a}\right) p_v''(T_{ref})\right]} \qquad (7)$$

where $M_a = 28.97\, g/mol$ and $M_v = 18.02\, g/mol$ are the values of molar mass of air and water vapour respectively; $p_{atm}$ and $p_v''$ are the vapour atmospheric pressure at actual and at saturation condition, respectively; $\varphi$ is relative humidity which equals unity at the water-air interface; $T_{ref}$ is the temperature at the air-water film interface.
The saturated pressure $p_v''$ can be evaluated as in [3]:

$$p_v'' = 611.85 exp\left[17.502 \frac{(T_{ref} - 273.15)}{(T_{ref} + 273.15)}\right] \qquad (8)$$

Note that expression of (8) is strictly valid for the water-air interface.

### 2.3. Flow regime

The treated case can be identified as a problem of internal flow over a horizontal plate. When assuming reference properties, those of ambient conditions and heated fluid film are assumed.
To determine whether the driving force is caused by natural or forced convection was used the Richardson number. The Richardson number means the relative intensity between natural and forced convection [4].
In case of ambient conditions given during measurements conducted in the summer, the calculated Richardson number is $Ri = \frac{Gr}{Re^2} > 65$, which implies that we are dealing with a case of mixed convection regime, where the *natural convection is dominant*; however, the role of the *forced convection is still assumed* since the conditions of natural and forced convection equality is exceeded by only a factor of 10.
Judging the flow from the perspective of natural convection, the value of Rayleigh number $Ra = 10^7$ suggests turbulent regime. Although the Reynolds number gives values of laminar flow, still it is convenient to consider the flow as turbulent – same approach is adopted in [5].

## 3. Numerical approach and simulations settings

Simulations are carried out using the commercial CFD tool of Star- CCM+ and were conducted as three dimensional with gravity effect considered. The continuum is assumed as non-reacting. Such formulation can be described as using a collocated variable arrangement (opposed to staggered) and Rhie-and-Chow type pressure-velocity coupling combined with a SIMPLE type algorithm. This model is more suitable for constant density flows but it can handle mildly compressible flows and low Rayleigh number natural convection [6]. Although, the simulated process is a combination of free and forced convection the solution might be assumed as steady. The flow is modelled as turbulent recalling the identification in section 2.3. *Realisable k-ε Two-Layer* turbulence model is selected since it is possible to activate *Buoyancy Driven Two-Layer Type model* correlating the turbulence parameters (turbulent kinetic energy *k* and turbulent dissipation rate *ε*) for flows where buoyancy forces dominate.
Radiation effect is not neglected due to its considerable influence. Since dry air and moist air do not participate in radiation heat transfer the *Surface-to-Surface model* is selected. Radiation properties of surfaces are considered same for all wavelengths, the *Gray Thermal Radiation model* is used, and their values are defined according to [7].

### 3.1. Boundary conditions

Figure 1 describes parts of the computational domain. The test rig is manufactured from different materials, therefore, the heat transfer coefficient distribution is not constant. The distribution of heat transfer coefficient along the test rig shows figure 2.

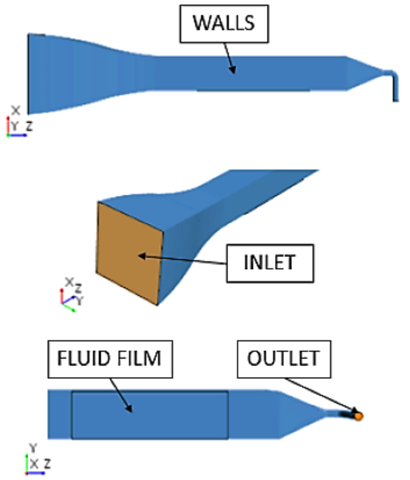


***Figure 1.*** *Parts of computational domain*

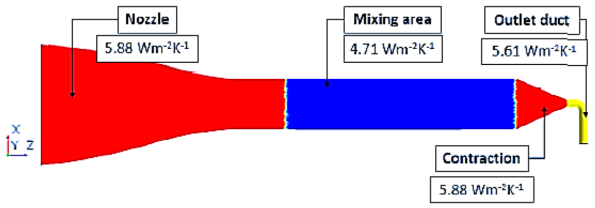


***Figure 2****. Heat transfer coefficient along the test rig.*

## 4. Experimental validation

A psychrometric method for measurement of evaporation rate is used. The evaporation rate is evaluated from the difference of specific humidity of passing air current over hot water film (schematically shown in fig. 3). Conservation of mass must be valid in the test section:

$$\dot{m}_{da}(1 + x_{in}) + \dot{m}_{ev} = \dot{m}_{da}(1 + x_{out}) \qquad (9)$$

from which, the evaporation rate $\dot{m}_{ev}$ can be evaluated:

$$\dot{m}_{ev} = \dot{m}_{da}(x_{out} - x_{in}) \qquad (10)$$

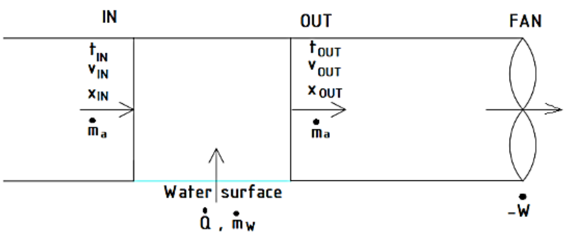


***Figure 3.*** *Schematic of the experiment*

The experimental setup used for the validation is shown in figure 4. At the inlet section of the test rig is a nozzle, which straightens the flow. Experiments were performed in 300mm x 300mm square horizontal test section. The test section is 1000mm long and the top and side walls are made from 8mm thick plexiglass and insulated by 25mm mirelon plates. At the bottom of the test section is water tank (at the bottom and side walls are power regulated heating foils) containing heated water for maintaining stable conditions. An aluminium plate is dipped in the tank to create the water film. The plate is equipped with 18 temperature sensors (digital thermometers Dallas Ds18b20, ±0.5°C accuracy) to monitor plate’s temperature and its uniformity. The air mass flow rate through the test section is measured by orifice plate. Differential pressure transducer Setra 265 with range ±125Pa is used for measuring the orifice pressure difference. The mass flow rate is evaluated according to standard CSN EN ISO 5176-2 [8]. Specific humidity is measured by psychrometers, each one consists of two sheathed RTD probes (PT1000) - wet and dry thermometer. The wet thermometer is wrapped by a moisten sock (by distilled water). Two psychrometers are located in outlet pipe for measurement of the outlet specific humidity. Psychrometer measuring inlet specific humidity is located above the entrance of the nozzle.

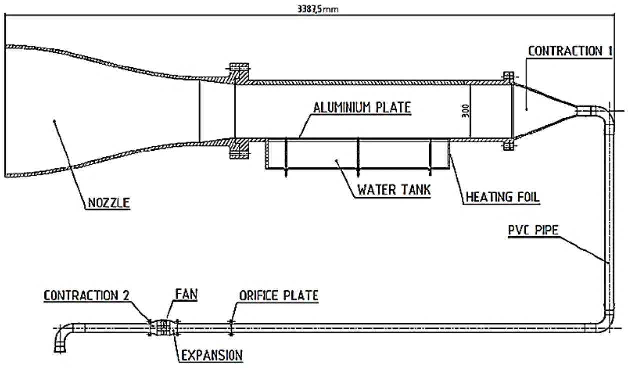


**Figure 4**. *Drawing of test rig*

### 4.1. Measurement procedure and evaluation

One single measurement took approximately from 5 to 10 minutes, so i tis worked with time-averaged values. From processed data, the specific humidity is evaluated as [8]:

$$\chi = \left[\frac{2501.6 - 2.3263\cdot(T_{wb} - 273.15)}{2501.6 + 1.8577\cdot(T - 273.15) - 4.184\,(T_{wb} - 273.15)}\right]\cdot\left[\frac{0.62509\cdot p_{wb}}{p_{atm} - 1.005\cdot p_{wb}}\right] - \left[\frac{1.00416\cdot(T - T_{wb})}{2501.6 + 1.8577\cdot(T - 273.15) - 4.184\,(T_{wb} - 273.15)}\right] \qquad (11)$$

Mass flow rate of humid air is evaluated according to [9]. From that, mass flow rate of dry air is calculated:

$$\dot{m}_{da} = \frac{\dot{m}_{ma}}{1+\chi_{out}} \tag{12}$$

After evaluation of previously mentioned properties, the evaporation rate is calculated by above written eq. 10. Uncertainty analysis is completely described in [10].

## 5. Results and discussion

Simulation results are compared with the experimental measurements from which the boundary conditions are determined. For the sake of brevity in this paper will be presented the results comparison for several boundary conditions, and it will be shown for the for water evaporation rate and the outlet humidity.
Dependencies presented in fig. 5 and fig. 6 are within ranges $\dot{m}_{in} \in <0.0104; 0.0106>$, and $\omega_{v_{in}} \in <0.0125; 0.0134>$ assumed as linearly dependent.
From the comparison of experimental data and simulation results of water evaporation rate (Figure 5) can be concluded that the implemented model, the *two active scalars* based model results deviation from the experiment results is within the range of 10%.

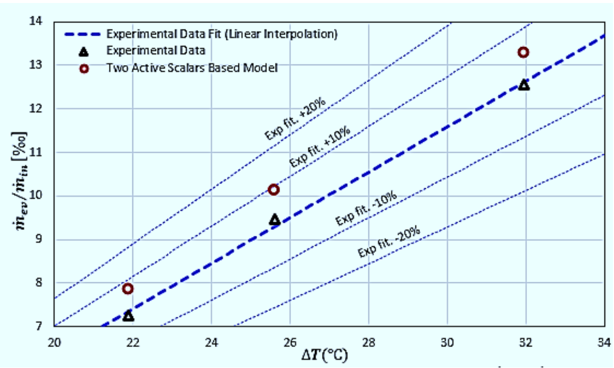


***Figure 5***. *Water evaporation rate dependency on temperature difference* ($T - T_0$)

Comparison of simulation results and experimental data of outlet humidity dependency on inlet humidity (Figure 6) shows good agreement and simulation results deviation is within the range of 10%.

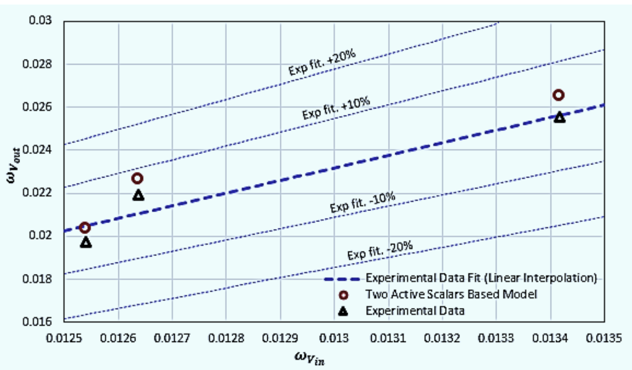


***Figure 6.*** *Outlet vapour mass fraction dependency on an inlet vapour mass fraction*

## 6. Conclusion

In this work, an attempt has been made to simulate evaporation from a thin horizontal liquid film. The model of interest, *the two active scalars* based model is firstly described theoretically, in order to highlight its main principles, and then its implementation into commercial Star-CCM+ code is shown. The validity of the developed model is evaluated by comparison with the experimental data, and the chosen experimental method was described. The simulation validity assessment was shwon for water evaporation rate and and outlet humidity. It can be concluded that the comparison of simulation results and experimental data of water evaporation rate shows a very good agreement; the difference of simulation results from the experimental data is on average less than 10%.
It was said that in this work the heat exchange between the solid substrate and the laying water film is not treated. Evaporation/condensation on solid surfaces absorbs/releases heat; in a wide range of applications, these effects strongly alter the temperature at the solid boundaries. In future work the exchange of heat between the three medias i.e. solid, water film and gaseous phase media respectively, it is recommended to be taken into consideration.

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## Symbols

| | |
|---|---|
| $\chi$ | specific humidity (kg·kg$^{-1}$) |
| $\dot{m}$ | mass flow rate ($kg \cdot s^{-1}$) |
| $p$ | pressure (Pa) |
| $\alpha$ | thermal diffusivity ($m^2 \cdot s^{-1}$) |
| $\beta, \beta_T$ | Volumetric thermal expansion coefficient ($K^{-1}$) |
| $\omega$ | Mass fraction (1) |
| $T$ | Temperature (K) |
| $\rho$ | density (kg·m$^{-3}$) |

## Subscripts

| | |
|---|---|
| T | Thermal |
| da | Dry air |
| ev | evaporation |
| in | inlet |
| out | outlet |
| atm | atmospheric |
| i | interface |
| v | vapor |
| ref | reference |
| a | air |

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