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CFD Study for the Optimization of the Drying Process of Foundry Moulds used in the Production of Wind Turbine Components

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ABSTRACT
In order to drive down the cost of wind turbine cast components, the optimization of each production step is necessary. In particular, foundry moulds used for the production of cast components undergo a process of drying needed to avoid quality problems in the final parts. In order to reduce drying times forced convection by the use of fans is needed. In this work we perform Computational Fluid Dynamic studies with the aim to optimize the drying process for mould geometries typically used for the production of wind turbine components. Representative geometries are modelled in a 3D software, imported in a fluid flow solver and complete Navier-Stokes equations coupled with energy transport equations are solved. Velocity profiles from shop floor measurements are used as boundary conditions for the problem. Finally surface heat exchange coefficients are determined and results analyzed. Results show that it is possible to use this methodology to optimize the drying process, and determine areas of the moulds that are more difficult to dry than others. Optimal fan arrangement for typical geometries are also provided.
Keywords – CFD Simulation, Drying, Foundry Moulds, Wind Turbine Components, Quality, CAE

I. INTRODUCTION

In order for wind turbines to become more and more competitive there is a need to lower cost by optimizing production times and quality [1]. Drying of coated sand moulds is a time consuming, as well as energy and capital intensive process. CFD simulations can be used as a tool to decrease the design time and lower the possible risk and costs related to the start- up of drying operations [2,3].

In this study we consider the case of moulds coated with water based foundry coatings [4,5] dried with forced air convection obtained by fans. The heat transfer coefficient (dependent on the air speed and temperature) plays a major role in the removal of water and therefore in the reduction of drying times [6,7].

Given the large dimensions of the moulds to be dried (typically above 3000x3000x3000mm) it is important to choose a fan layout that minimize the areas with low heat exchange coefficients and help to determine the area that will be the last to dry so that quality checks can be done with higher certainty [8].

The goals of this work can therefore be summarized as follows:

- Asses different fans types with CFD simulations
- Propose an optimal fan layout typical for wind turbine components moulds
- Determine areas that are difficult to dry in typical wind turbine component moulds
- Perform a qualitative validation for the use CFD methodology

II. METHODS

2.1 Fan Selection

The first step of the optimization consists in comparing the performances of different fans (Table 1) with the objective to choose a fan model to be used in the subsequent mould drying simulations.

CFD simulations of each fan model were carried out to compare the performance and make a preliminary selection. A representative test chamber of dimensions 8000x6000x5000 mm was used and the fans were placed at three different distances (1500, 2500, 3500 mm) from a surface at constant temperature. The test chamber was chosen in order to minimize the effect of walls on the enclosed flow field.

The incompressible Navier-Stokes equations and the energy equation are solved in a pressure-based segregated solver of SIMPLE type, with standard under relaxation of inner iterations. The buoyancy effects of the natural convection are accounted for by the Boussinesq approximation in the negative z direction. The turbulence model selected is the standard k-ω (k-omega) with wall functions. The boundary conditions in terms of the velocity field were obtained after technical data from the data sheets of the different fans. A fixed temperature increment is applied between inlet and outlet sections of the fan, in order to trigger the heat transfer effects on the target wall. This temperature difference allows to calculate heat transfer coefficient maps on the target wall, as in Figure B.

The simulations were run in order to compare the performances of three different fan models reported in Table A and to determine the effects of the distance between the fan and the target wall for each model.

Table 1. Fan models considered in this study.

	TFV 10 S	TFV 30 S	TTV 4500
Air flow rate max [m³/h]	525	3500	4500
Air pressure max [Pa]	80	500	80
Power input [kW]	0.1	1.2	0.23
Dimensions [mm x mm x mm]	300 x 275 x 330	545 x 515 x 490	210 x 510 x 510

2.2 Moulds Drying

For the mould drying simulations we start by modeling three representative geometries, namely hub, main foundation and main bearing house in a 3D software. These geometries are then imported into a fluid flow solver where complete Navier-Stokes equations coupled with energy transport equations are solved. Moisture transfer and evaporation are not modelled in order to reduce the modelling effort and reduce the overall computational time, while velocity field and heat exchange coefficient are used to evaluate the effectiveness of different fan layouts.

For the simulation we choose chamber dimensions big enough (typically 10x10x7m) to have a far field stagnation velocity smaller than 1 m/s (as in an ordinary room with still air) [9]. The typical number of mixed cells (tetrahedral, pyramids, hexahedral) in the simulation is around 1.000.000, with a dimensions refined towards the impinging jet and on the target surface. The heat transfer coefficient is computed by assigning a fixed temperature to the target surface and heating up the air with a $\Delta T = 5°$ C as the ambient air flows through the fan.

A qualitative validation of the results is done by comparing the results for the hub with a picture take by infrared camera (moulds areas with higher surface temperatures represent areas with higher heat exchange coefficients).

III. RESULTS

3.1 Fan Selection

A first qualitative evaluation of the fans can be done visually, by comparing the velocity contours as shown in Fig. 1 for the fan TFV 10 S and a wall distance of 3500 mm. It can be seen that, for this specific fan, the velocity drops to below 1 m/s before reaching the surface at 3500 mm distance.

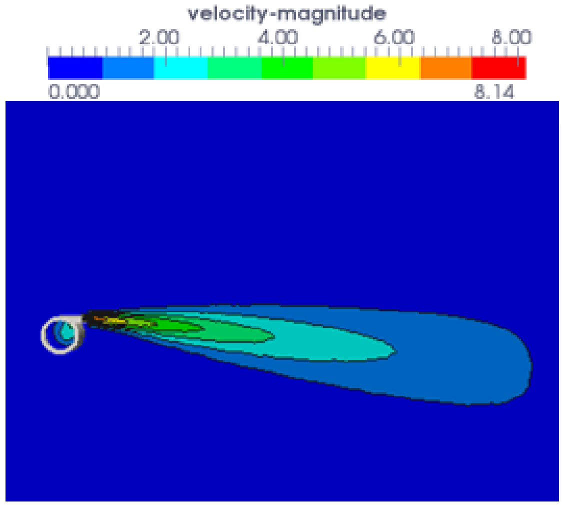

Figure 1. Velocity magnitude [m/s] contours over the central section plane for the fan TFV 10 S for a wall distance of 3500 mm.

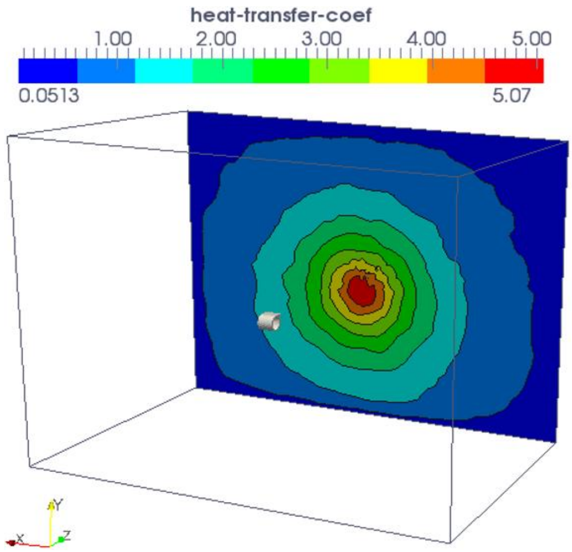

Figure 2. Convective heat transfer coefficient [W/(m^{2}K)] distribution over isothermal surface for TFV 10 S fan for a wall distance of 3500 mm.

As consequence we can see how the heat exchange coefficient field on the target surface is below 5 W/(m^{2}K) as shown in Fig. 2.

In order to compare the three different fans we plot the heat exchange coefficients on the x axis of the wall at the same y coordinate of the fan center of mass (Fig. 3). We can see how the TTV 4500 fans provides the highest heat exchange coefficient at a fraction of the power used by TFV 30 S. The TFV 10 S provides the smallest heat exchange coefficients.

Additionally, we notice that for all the distances form the surface, the TTV 4500 provides a heat transfer coefficient higher then 10 W/(m^{2}K) (well above the typical values achievable by natural convection [10]) on an approximately circular area of two meters in diameter.

Based on these results we choose the TTV 4500 for the mould simulations and decide to pre-dimension the number of required fans based on the area of the moulds and the area spanned by a heat transfer coefficient above 10 W/(m^{2}K).

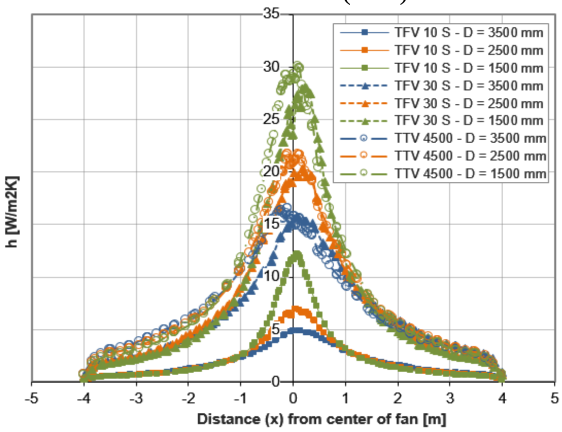

Figure 3. Convective heat transfer coefficient [W/(m^{2}K)] distribution for different fans and wall distances along the x axis.

3.2 Hub Mould Drying

Figures 4 and 5 show the optimal fan layout chosen for the hub mould using three fans.

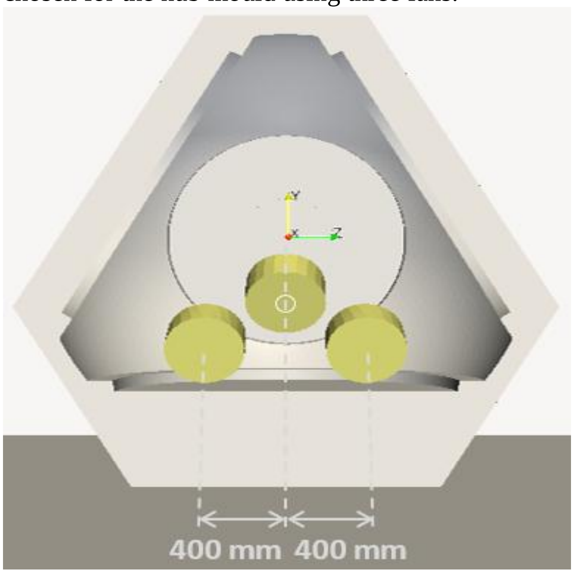

Figure 4. Optimal fan arrangement for the hub (front view).

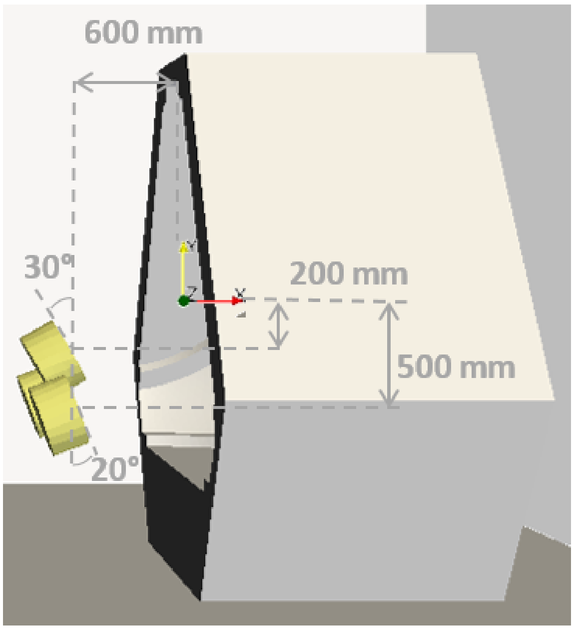

Figure 5. Optimal fan arrangement for the hub (side view).

Figure 6 shows how air is channelled in the bottom section of the cavity and exits from the top, thus ensuring a steady air velocity to be maintained above the mould surface. This has beneficial effects on the average heat transfer coefficient which is thus increased with respect to other arrangements simulated.

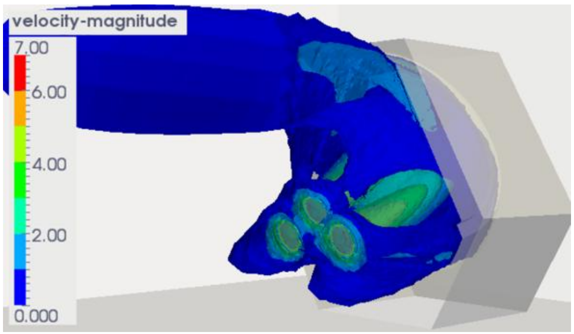

Figure 6. Velocity magnitude [m/s] isosurfaces. Air stream from the fans is channeled in cavity.

Figure 7 shows the heat transfer coefficient distribution on the mould surface. It is possible to appreciate that the areas with the lowest values are localized near the corners. This observation allows to determine where localized moisture measurements should be done in real production.

From Figure 8 we see that less than 2% of the total area of the mould is subjected to a heat transfer coefficient of less than 10 W/(m^{2}K).

In Figure 9 we see a surface temperature measurement done with infratred camera. The areas near corners have lower temperature due to lower heat exchange coefficient as shown by the simulation.

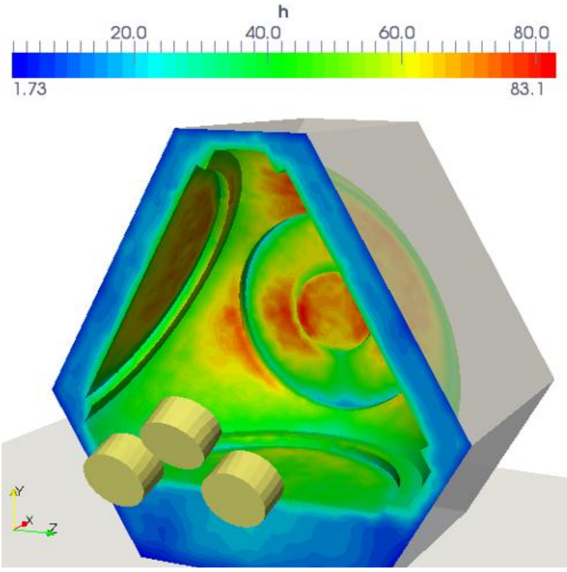

Figure 7. Convective heat transfer coeffient [W/(m^{2}K)] distribution on hub mould.

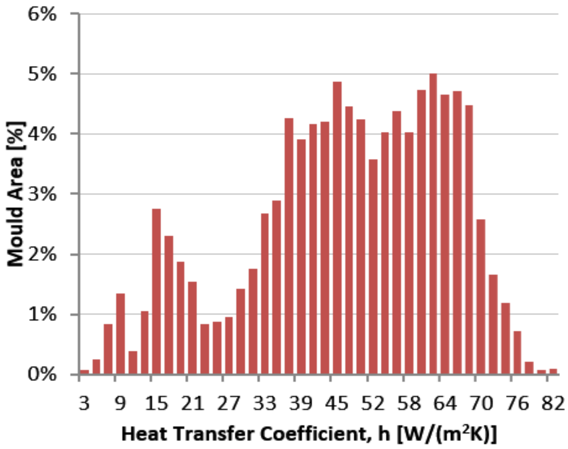

Figure 8. Histogram showing the distribution of the heat exchange coefficient on the hub mould surface.

Figure 9. Hub mould surface temperature measurement done with infratred camera. Areas near corners have lower temperatures due to lower heat exchange coefficient as shown by simulation.

3.3 Main Foundation Mould Drying

The optimal fan arrangement for the main foundation moulds requires six fans due to the larger mould dimensions. In particular we see that the fans are never placed perpendicular to the mould wall (Figure 10) in order to avoid formation of stagnation zones and therefore a reduction of the heat transfer coefficient. This is confirmed by Figure 11 where the jets exiting from the fans can be clearly seen.

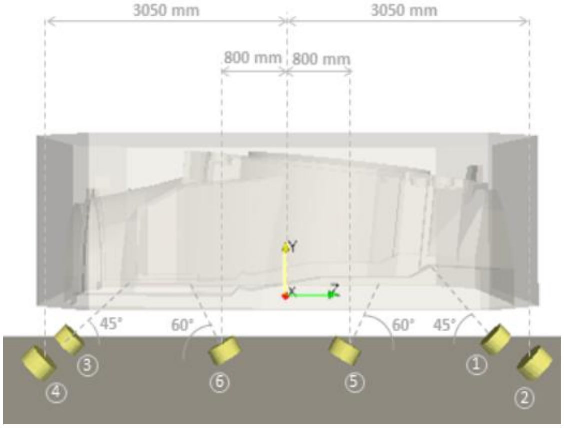

Figure 10. Optimal fan arrangement for main foundation mould.

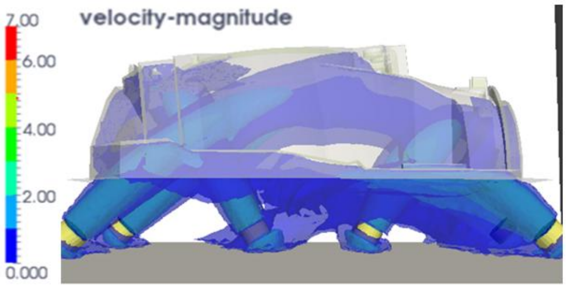

Figure 11. Velocity magnitude [m/s] isosurfaces. Air jets exiting from the fans are clearly visible below the drag.

The heat transfer coefficients obtained on the surface are concentrated in a lower range as compared to the hub (Figure 12). This means that the drying process of this mould will be necessary slower. However even for this case only less than 5% of the total area is subjected to heat exchange coefficients smaller than 5 W/(m^{2}K). These areas are clearly visible in dark blue in Figure 13 in correspondence of the top areas of the mould surface. Therefore it will be important to assess the moisture of these areas once the drying process is completed in order to ensure a complete drying and to avoid possible problems in quality.

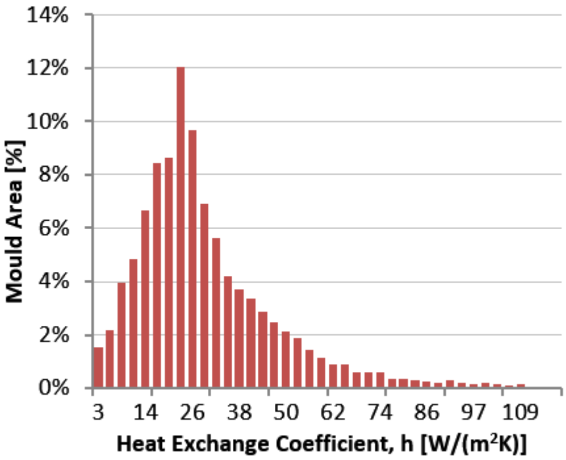

Figure 12. Histogram showing the distribution of the heat exchange coefficient on the main foundation mould surface.

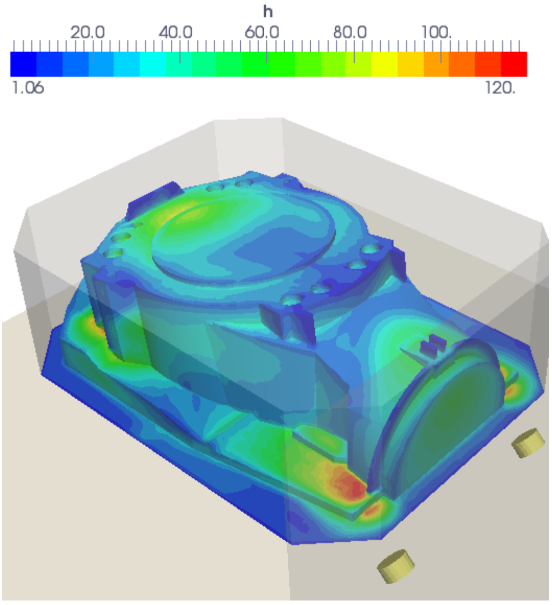

Figure 13. Convective heat transfer coeffient [W/(m^{2}K)] distribution for the main foundation mould.

3.4 Main Bearing House Mould Drying

The main bearing house requires two fans due to its smaller dimensions (Figure 14). Again the fans are positioned such that the air flow enters tangentially into the mould shape on one side and freely exits from the other as visualized in Figure 15.

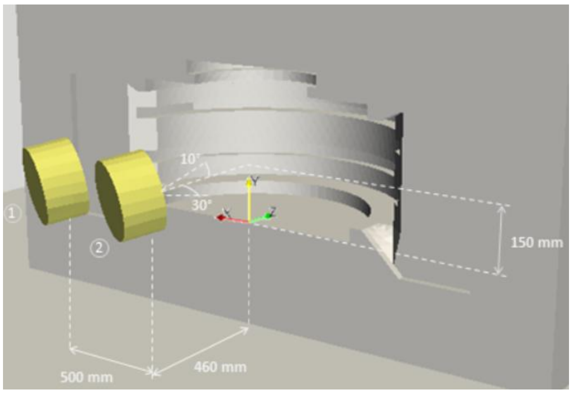

Figure 14. Optimal fan arrangement for main bearing house mould.

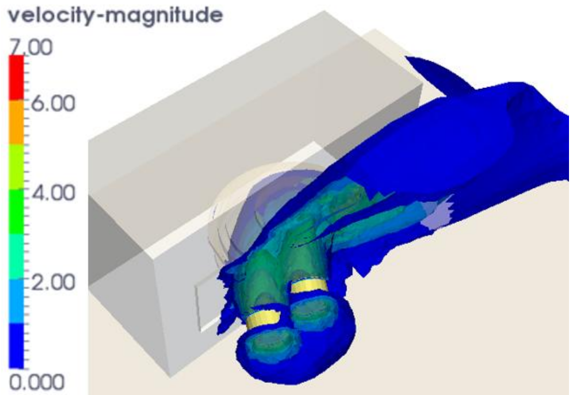

Figure 15. Velocity magnitude [m/s] isosurfaces. Air stream from the fans is channeled in the cavity.

With regard to heat transfer coefficients we see that the areas with minimum heat transfer coefficient are localized in the deeper pockets and cavities of the mould (Figure 16). Therefore these will be the areas to be checked for residual moisture after the drying process is completed.

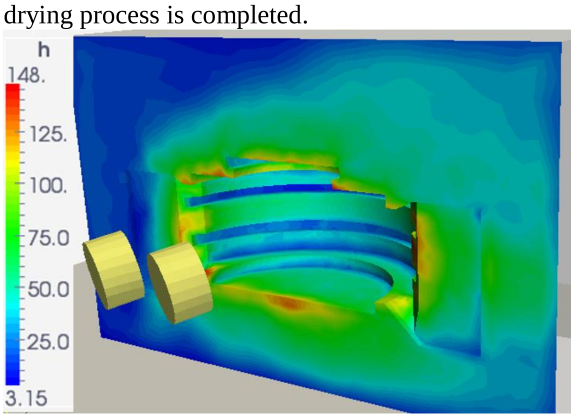

Figure 16. Convective heat transfer coeffient [W/(m^{2}K)] distribution for the main bearing house mould.

A final check of the heat transfer coefficient distribution (Figure 17) shows that only less than 0.5% of the mould area is subjected to a heat transfer coefficient smaller than 5 W/(m^{2}K). Also the overall distribution has higher values than the one of the other moulds previuosly analyzed. This suggests that the drying times for this items will be shorter.

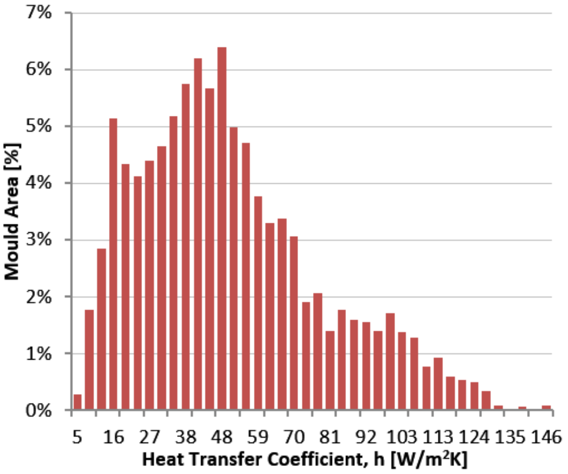

Figure 17. Histogram showing the distribution of the heat exchange coefficient on the main foundation mould surface.

IV. CONCLUSIONS

The main conclusions of this study can be summarized as below:

- Different fans type performances were assessed with the aid of CFD simulations and the TVV 4500 has been chosen due to the largest covered area with a reference level of the heat transfer coefficient around 10 W/(m^{2}K).
- Optimal fan layouts for the moulds of three typical components of wind turbines were proposed, in particular the hub will require 3 fans, the main foundation 6 fans and the main bearing house 2 fans.
- Areas which are particularly difficult to dry have been reduced to less than 2% of the mould surface and evidenced for moisture measurements during production process.
- Qualitative validation of the CFD model used was performed by comparison with infrared camera measurements for the hub mould and a satisfactory agreement was found.

V. ACKNOWLEDGEMENTS

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