Thomas Bewer, Cham, CH

3D Simulation of the Plunger Cooling during the Hollow Glass Forming Process – Model, Validation and Results

A steady state model to describe the flow and temperature distribution in plunger and cooler set-ups for the hollow glass industry was developed. Applying the model parametric studies and design comparisons of existing designs were done. The model applicability was proven in field trials.

1. Introduction

An important step in the hollow glass forming process is the pressing of glass with a so called plunger. As the plunger is in direct contact with the about 1100°C hot glass, it is exposed to a significant heat input and therefore has to be cooled to avoid sticking of the glass. In practice the cooling is done with air, which is fed to plunger inside by a bored cooling tube (cooler). This study describes the development and validation of a 3D model that predicts the flow and temperature distribution in plunger and cooler set-ups. With this tool the governing parameters for the plunger/cooler set-up are determined. The work was done as an IPGR (International Partners in Glass research) project. The IPGR is a worldwide research consortium of today seven glass plants. Besides very close co-operations in benchmark projects, fundamental R & D projects are funded. The scope of the IPGR project was to develop a steady state and transient model. In this report only the steady state model is described.

As a first step of the model development the necessary boundary conditions are formulated. These boundary conditions are implemented in the commercially available CFD (Computational Fluid Dynamics) Code Fluent that allows solving the coupled momentum and energy conservation equations. In trials the assumed boundary conditions are validated and verified.

With the validated model the most important parameters influencing the temperature and flow distribution are identified using the Design of experiment method. Based on this experience predictions and optimisation recommendations for a plunger and cooler design were generated and compared to measurements in field trials.

2. Calculation model

2.1. General calculation procedure

As stated above the model bases on the commercially available CFD (Computational Fluid Dynamics) Code Fluent. This code is a general-purpose solver to solve coupled thermal and fluid dynamics problems. It incorporates all the necessary solution algorithms and therefore saves time in setting up the calculation model. Furthermore it reduces the risk of faulty programming significantly. In order to set up a problem with Fluent it is first of all necessary to define the geometry and the calculation grid. For this purpose a special program that is part of FLUENT called Gambit is used. It allows defining the geometry and after that meshes this geometry model. As the geometry creation and the meshing is a time consuming step the geometry creation is automised.

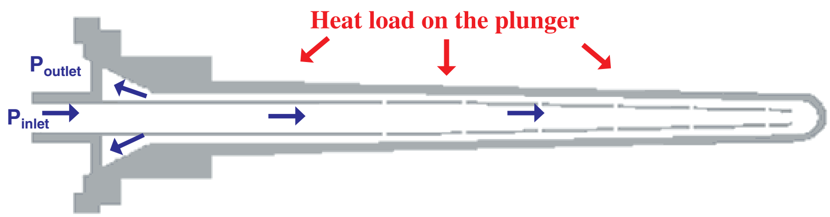


Figure 1: Geometry and necessary boundary conditions for the steady state model

This calculation model is transferred to the Fluent solver. In order to solve the coupled equations of momentum (Navier-Stokes), mass and energy conservation it is necessary to know the conditions on the boundaries of the calculation model. These conditions have to be specified by the user. Basically these are the heat load on the plunger and the pressure at inlet and outlet as illustrated by Figure 1. In order to save calculation time only one quarter of the axis-symmetric geometry is used. At the cutting area symmetry boundary conditions are applied.

2.2. Used boundary conditions

In order to define the flow region (that means the equations for conservation of mass and momentum) the pressure values at the inlet and outlet of the calculation model have to be set. In the model the outlet is located at the outlet of the plunger. The inlet is determined to be the inlet of the cooling tube.

The conservation equation of energy needs boundary conditions for heat input at all outer surfaces.

2.2.1. Mass and Momentum

As described above the pressure condition at the inlet and outlet have to be set. On the outlet it is assumed that the static pressure equals the ambient pressure.

The problem with the inlet pressure is that only the pressure at the valve is known. In order to reduce the model to a manageable size, the calculation starts at the entrance of the cooler. Therefore the pressure drop between the valve and the entrance of the cooler has to be determined. The pressure at the cooling tube entrance is then the pressure at the valve reduced by the pressure drop.

In order to determine the pressure drop between valve and cooling tube two measurements were conducted. In both cases a pressure tank (p_{max} = 5bar) with a capacity of 1.5 m³ was used to supply a plunger mechanism with air. The mass flow through the system was monitored. In the first experiment plunger and cooler were mounted, whereas in the second only a plunger was installed. For the measurements a typical plunger and cooler set-up was used.

The pressure drop can be determined as follows: With the known pressure at the valve the air flow through the entire system (Mechanism, plunger and cooler) can be determined from the first experiment. With this airflow, the pressure loss through the mechanism can be deduced from the second experiment. This was done for various air flows. The mass flow vs pressure drop characteristic was implemented in Fluent via a "User defined function".

2.2.2. Energy

In order to solve the conservation equation for the energy it is necessary to know what the heat input and output out of the system is. Therefore the thermal conditions on every surface of the model have to be specified.

For the air inlet temperature a value of 40°C is assumed. This accounts for the warming of the air on its way through the mechanism. The outlet temperature is automatically calculated by Fluent depending on all the given equations and boundary conditions.

The heat input on the plunger surface that is in contact with the glass depends on the contact time of plunger and glass. In former experimental investigations done by Emhart Glass the following equation fitted the heat input from glass to a blank mould very good. As a measurement of the heat transferred from glass to plunger is very complex for this study it is assumed that the results obtained from the blank mould / glass contact can be transferred to the plunger / glass contact situation. The following equation applies.

$$Q = \frac{T_{gob} - T_{plunger}}{C_1} \cdot 10^{C_2} \cdot \sqrt{t_{contact}}$$

Equation 1

with

Q	Transferred heat [J]
T_{gob}	Temperature of gob [°C]
$T_{plunger}$	Plunger temperature [°C]
$t_{contact}$	Contact time of glass and mould [s]
C1, C2	Constants

In order to reduce the necessary calculation time a steady state examination is done. Therefore the transferred heat has to be referred to the time the cooling is applied (t_{cool}). The heating power of the glass then equals:

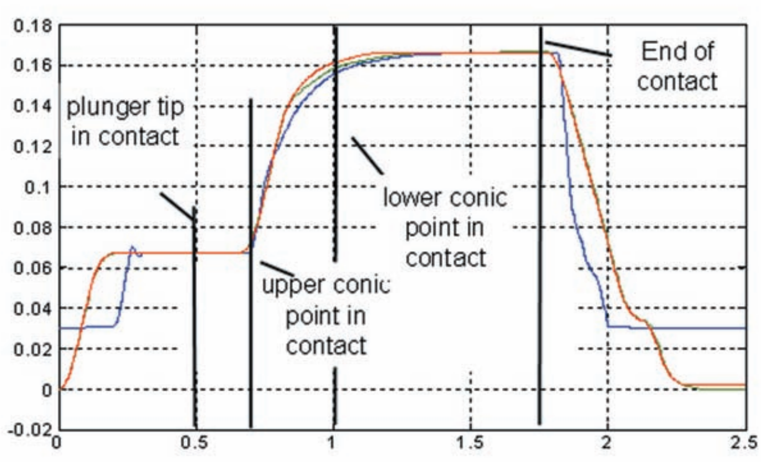


Figure 2: Measured motion profile of the plunger for a 33cl bottle

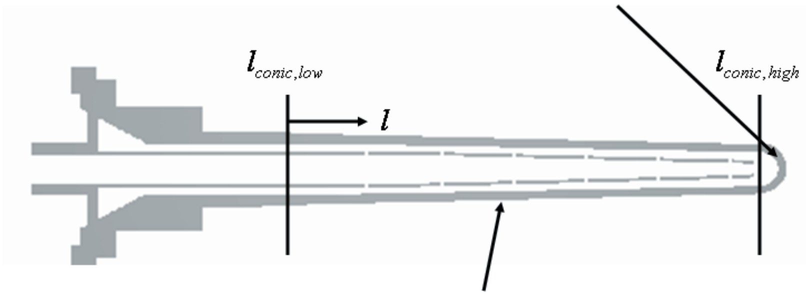
$$\dot{Q}_{ref} = \frac{Q}{t_{cool}} \qquad \text{Equation 2}$$

With equation 1 and 2 the heat load on the plunger can be determined if the contact time is known. The contact time depends on the location. Figure 2 shows a typical measured motion profile for a plunger used for 33cl beer bottle. The plunger tip has the longest contact time. Later the plunger begins to move upwards and the upper point of the conic part comes into contact. After that the lower point of the conic part and small parts of the cylindrical part come into contact until the plunger is retracted.
For a correct determination of the heat load on the plunger it would be necessary to measure this motion profile for every container. As this is costly (both money and time wise) the motion profile shown in Figure 2 is used for all container examined in this study.

In order to transfer the motion profile to another container the contact times are referred to the overall contact time $t_{contact,\ total}$ = 1.25 s. The fraction obtained for the three characteristic points plunger tip in contact, upper conic point in contact and lower conic point in contact is shown in Figure 3.

The contact times for the characteristic points of a different container can now be calculated by multiplying the

$$Q_{tip,local} = \frac{T_{gob} - T_{plunger,tip,local}}{C_1} \cdot 10^{C_2} \cdot \sqrt{t_{contact,tip}}$$



$$t_{contact,local} = t_{contact,conic-low} + \frac{t_{contact,conic-high} - t_{contact,conic-low}}{l_{conic-high} - l_{conic-low}} \cdot \left(l - l_{conic-low}\right)$$

$$Q_{tip,local} = \frac{T_{gob} - T_{plunger,tip,local}}{C_1} \cdot 10^{C_2} \cdot \sqrt{t_{contact,tip}}$$

Figure 4: Summarised boundary conditions on the plunger surface

above obtained fraction with the overall contact time. The overall plunger contact time in return can be calculated from the machine settings depending on the individual job.

$$t_{contact,tip} = 1 \cdot t_{contact,total} \ ,$$
$$t_{contact,conic-high} = 0.84 \cdot t_{contact,total} \ ,$$
$$t_{contact,conic-low} = 0.6 \cdot t_{contact,total}$$

With this information the localised heat input on the plunger can be calculated. Equation 1 and 2 become:

$$Q_{local} = \frac{T_{gob} - T_{plunger,local}}{C_1} \cdot 10^{C_2} \cdot \sqrt{t_{contact,local}} \qquad \text{Equation 3}$$

$$\dot{Q}_{local} = \frac{Q_{local}}{t_{cool}} \qquad \text{Equation 4}$$

In order to further simplify the calculation it is assumed that the motion profile is linear. Under this assumption it is possible to interpolate between the high and low conic contact points. The local contact time becomes a function of the contact time of the high and the low contact point and the length of the plunger. No profile is assumed on the plunger tip. This correlation is illustrated in Figure 5. All surfaces not in contact with the glass are assumed to have a zero heat flux.

That means that no heat is transported into the plunger, nor heat is lost.

3. Model validation

For the validation of the above-described models verification trails were performed. The mass flow through the investigated designs is determined to verify the flow part of the model. The thermal model was validated by monitoring the cooling of a preheated plunger.

$$t^{measure}_{contact,total} = 1.25$$

$$t^{measure}_{contact,tip} = 1.25 \quad \Rightarrow \quad \frac{t^{measure}_{contact,tip}}{t^{measure}_{contact,total}} = 1$$

$$t^{measure}_{contact,conic-high} = 1.05 \quad \Rightarrow \quad \frac{t^{measure}_{contact,conic-high}}{t^{measure}_{contact,total}} = 0.84$$

$$t^{measure}_{contact,conic-low} = 0.75 \quad \Rightarrow \quad \frac{t^{measure}_{contact,conic-low}}{t^{measure}_{contact,total}} = 0.6$$

Figure 3: Contact time fractions for characteristic points using measured motion profile

$$t_{contact,total} = 1{,}25s$$

3.1. Flow model validation

A laboratory test rig was used in which the mass flow through the plunger/ cooler set-up was determined with a flow meter. Pressurized air is supplied via a 0.1m long flexible tube (see Figure 5).
The flow model is verified by comparing the measured overall mass flow through the different plunger/cooler assemblies with calculated values. The following assumptions are made for the calculations:

- steady state
- compressible air flow
- temperature dependent air properties

Lab experiments at 20 psi are used for validation purposes. Figure compares the measured with the calculated values. For the five measurements the agreement between measured and calculated values is very good.

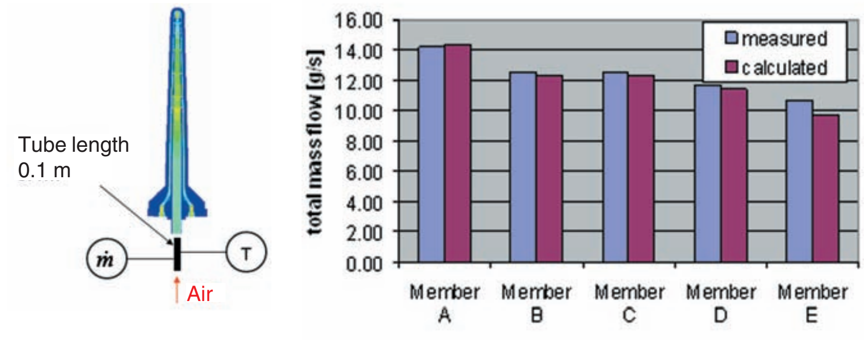


Figure 5: Measurement principle in lab and mass flow results

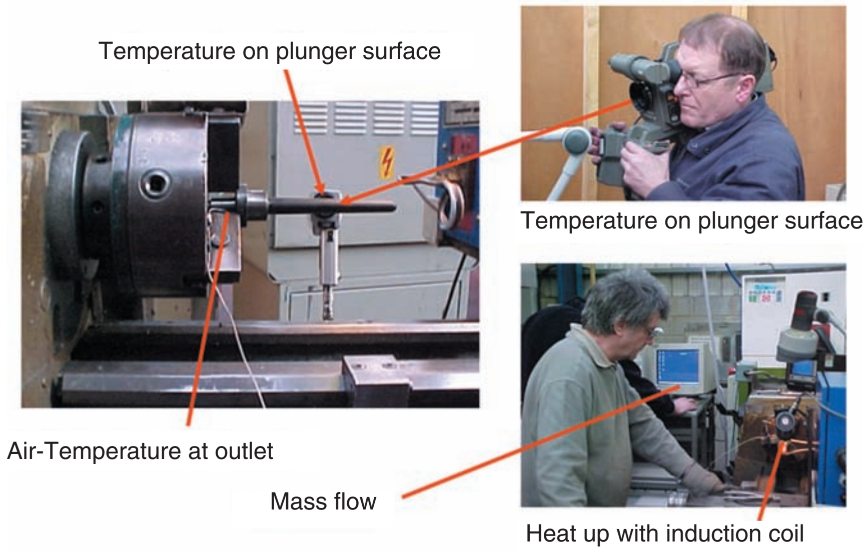


Figure 6: Experimental set-up for thermal model verification trails

3.2. Thermal model validation

The thermal model cannot be validated with the above-described set-up. The idea of the validation measurements is to heat up the plunger with an induction coil on a lathe. After the plunger is heated up uniformly the temperature change on the plunger surface is continuously measured with a thermal area camera. In order to minimise the measurement uncertainties, the plunger surface is sprayed with a high temperature resistant finish in order to get a defined emissivity. Figure 6 shows the experimental set-up.

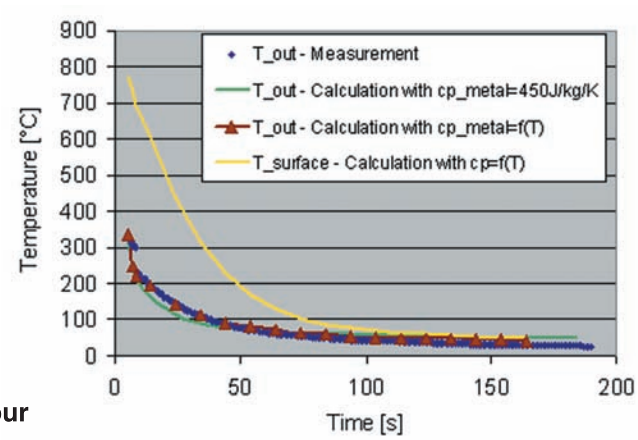


Figure 7: Comparison of transient behaviour

Unfortunately the camera could not be successfully connected to a computer, so that no monitoring of the thermal process could be obtained. The mass flow measurement device could not monitor the time dependence either. Nevertheless the air outlet temperature could be monitored over time using a type K thermocouple placed in one of the outlet bores. This time dependent measurement gives the chance to validate the model against the measurement.
For the modelling a transient calculation is performed. As for the flow model calculations a compressible air flow and temperature dependent properties are used for the air. As start condition a uniform temperature of the heated part of the plunger is assumed. The start temperature is set to the value measured with the area camera.

Figure 7shows the comparison of measurement and calculation

The calculations show that the heat capacity of the material is very important. Assuming a constant heat capacity for the plunger the experimental data could not be reproduced by the model. Using a temperature dependent heat capacity the model is in very good agreement with the measurements.

4. Modelling results

In the plunger cooling project in a first step all the plunger/cooler set-ups supplied by the IPGR members were compared against each other. In a second step one of the designs was chosen to perform a parametric study. Several influencing parameters were varied, while keeping the others constant. In order to identify interactions and quantify the effect of the parameters a statistical method called Design of Experiment (DOE) is used in a third step. In this report only the results of the DOE are discussed.

4.1.1 DOE Set-up

The Design of Experiment method bases on factorial designs. This means that two or more factors are varied simultaneously. Each factor is assigned a plus (+1) and a minus (-1) level. The evaluation of these factorial designs is basically the determination of the so-called effects. It is distinguished between main effects and interactions.
The main effect of a factor A (E_A) shows, which change a response variable Z will take, if the factor A is changed from its minus to its plus level. An interaction is the impact on a response variable, which is caused by the simultaneous change of several factors. A response variable is a measure that is characterising the process. In this case the following response variables are used:

- Overall mass flow [g/s]
- Overall heat extraction [J]
- Average Temperature on the conic part of the plunger surface [°C]
- Average temperature on plunger tip surface [°C]
- Minimum temperature on the conic part of the plunger surface [°C]
- Maximum temperature on the conic part of the plunger surface [°C]
- Temperature difference on the conic part of the plunger surface [°C]

The overall mass flow and the overall heat extraction are a measure how high the cooling capacity is. The average temperatures on the conic part and the tip specify how effective the cooling in these areas is. The minimum, maximum and difference of the temperatures on the conic part indicate the quality of the thermal distribution on the conic part of the plunger. In order to compare the influence from main effects and interactions on the response variables Pareto charts are used. These summarise the main effects and the interactions for the response variables.
A factorial 2^{5-1} design is used and analysed using the commercial software MiniTab. This means that the effects and second order interactions for 5 factors can be determined. The parameters that showed the biggest influence on the temperature in the parametric study are chosen as follows

- total inlet gauge pressure
- cooling time
- thermal conductivity of plunger material
- diameter of tip hole
- area of cooling holes

A further, very important factor in determining the temperature distribution is the heat input on the plunger. The heat input is influenced by the motion profile. As described above one motion profile was assumed for the study. In order to estimate what difference a change in the motion profile makes, two other profiles are discussed for the base case design. It is assumed that the motion time between the characteristic points of the profile is 25% shorter (fast motion), respectively longer (slow motion).

4.1.2. Results of the DOE

The pareto charts represent the results for the standard motion profile. Coloured stars in the charts indicate the results for the slow and fast motion profile. The "+" and "-" shown in the bars of the Pareto chart indicate if the value of the response variable increases (+) or decreases (-) if the factor is changed from its minus to plus level. The red lines indicated in the Pareto chart mark a significance level. This means that main effects and interactions with lower effect values as indicated by this line show within a 95% probability no influence on the response variable.
Comparing the results for the different motion profiles it is obvious that for all motion profiles and response variables the same effects show a significant influence. The identification of the governing effects seems to be independent of the motion profile.
Figure 8 shows that the overall removed heat is significantly influenced by inlet pressure, cooling time, hole area and the thermal conductivity of the plunger material. All other main effects and interactions are not significant regarding the heat removal.

With higher inlet pressures and hole areas the mass flow through the system increases. This results in a higher cooling capacity. Increasing the cooling time means applying the same mass flow for a longer time and therefore increasing the heat removed. With a higher thermal conductivity more heat is removed as it is distributed better from the outer surface to the inner surface of the plunger.
Figure 9 clarifies that the average conic temperature is significantly influenced by the same effects as the removed heat. The average conic temperature and the removed heat are close-linked as most heat is removed

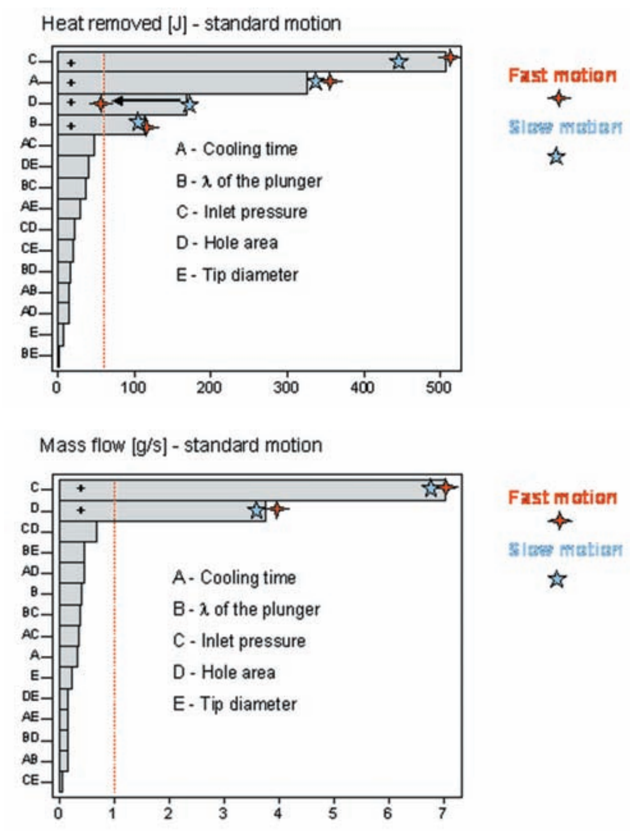


Figure 8: Pareto charts of response variables – cooling capacity

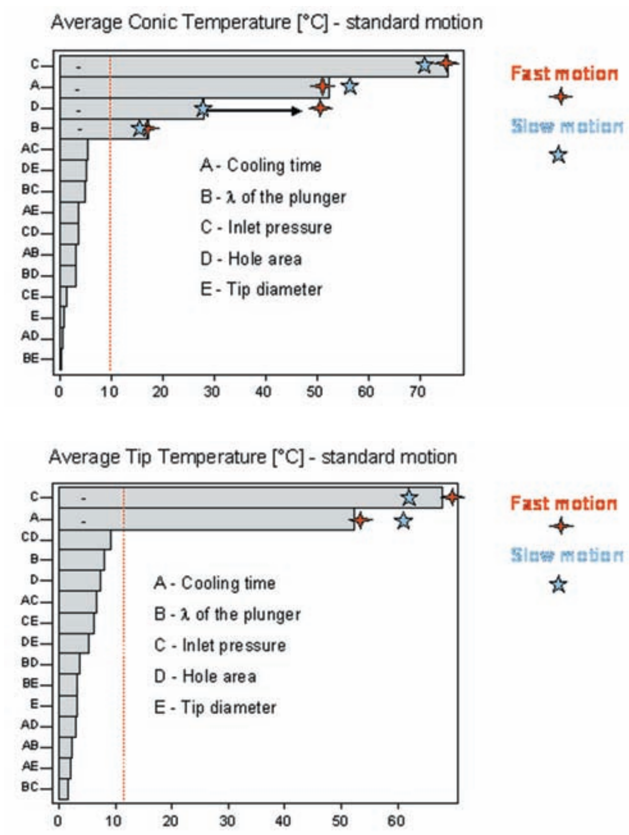


Figure 9: Pareto charts of response variables – Average temperatures

where the plunger is hottest and has the highest surface area. The region this applies to is the conic part.

The Pareto chart for the avera2ge tip temperature in Figure illustrates that only the inlet pressure and the cooling time influence the tip temperature. As the increase of the bore area has a minor influence on the mass flow through the tip bore, the heat extraction and thus the temperature is not influenced by a change in hole area. Furthermore the tip region is too small and the contact time longer that the better heat distribution due to a higher thermal conductivity has a significant influence on the tip temperature either.

The influencing effects for the temperature distribution on the conic part represented by the minimum and maximum temperature and the temperature difference are the same as for the heat removal. The hole area has a more significant influence as the mass flow distribution along the cooler is influenced, when the hole area is changed.

5. In factory measurements to prove model applicability

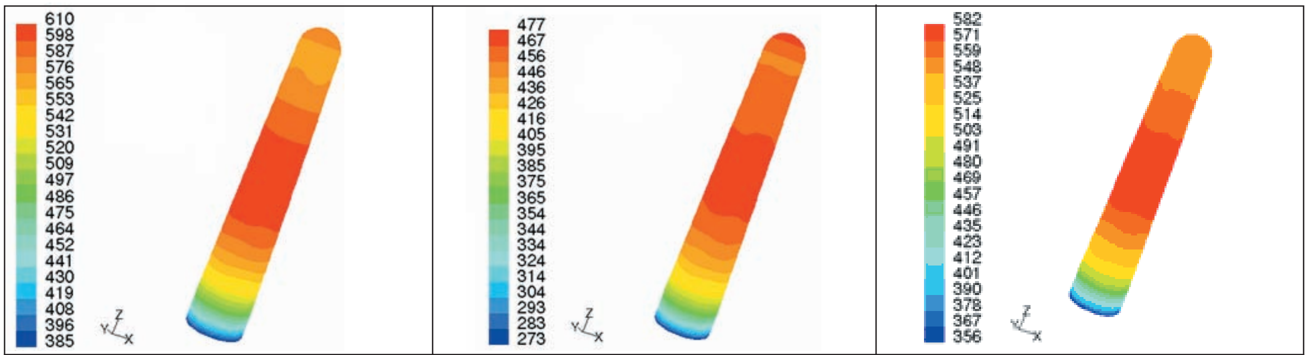
Using the above developed and verified model one plunger/cooler design was further analysed. Based on the experience of the parametric study and the DOE results an “optimised set-up” was determined. The predictions were compared to measurement data from field trials.

5.1. Model prediction

As a first step the temperature distribution of the current (= standard) set-up was calculated. Applying more bores in the cooling tube and changing the inner contour of the plunger, the mass flow throughput could be increased from 10.6g/s for the standard to 13g/s for the optimised design. This results in a temperature reduction of 130°C in the hottest region as can be seen from Figure 10. This reduction in return can be used for decreasing the pressure, meaning saving energy. Figure 10 shows that at a pressure of 1.5 bar still 20°C lower temperatures are observed. Changing the pressure from 2.8 bar to 1.5 bar results in a temperature increase of roughly 105°C.

5.2. Experimental set-up

In order to validate the model findings, the mass flow through the set-up and the plunger temperature have to be measured. The temperature is measured with a pyrometer as shown in Figure 11. This means that the temperature measurement is restricted to



Standard at p= 2.8bar	Optimized at p= 2.8bar	Optimized at p=1.5bar

Figure 10: Results of the optimisation

Figure 11: Pyrometer measurements in the field

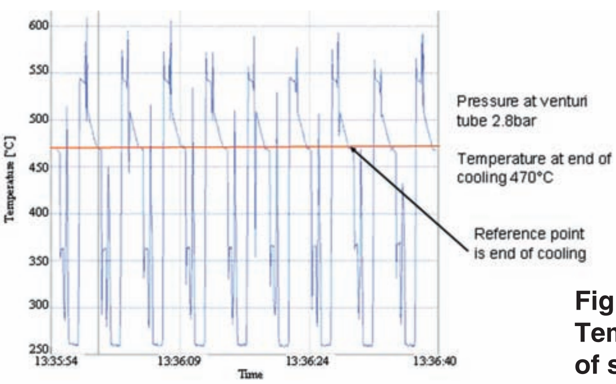


Figure 12: Temperature reading of standard set-up

the plunger area that is visible during the time the mould is open. An integrated camera helps to adjusts the pyrometer measurement point. The right picture in Figure 11 illustrates that the temperature is measured about 2mm below the tip. The emission coefficient is set to a value of 0.9. This value influences the absolute temperature value, but as the analysis of the measured data is relative the value of the emission coefficient has no meaning for the validation of the model. Unfortunately it is not possible with this experimental set-up to determine the temperature distribution on the plunger surface. This means that the modelling results concerning the equalisation of the temperature profile cannot be verified.

For the mass flow measurement a Venturi-tube is mounted in the piping between valve and machine.

5.3. Temperature measurements

Figure 12 shows the temperature measurement for the standard set-up. It shows the temperature over time assuming an emission coefficient of 0.9. For the analysis of the temperature measurements it is necessary to pick one reference point of each cycle. The reference point is chosen to be the point at the end of the cooling. It is marked in Figure 12. For the steady state of the plunger operation the temperature at this reference is constant over the cycles. This temperature is therefore called the "steady state reference temperature" in the following.

According to Figure 12 the steady state reference temperature for the standard set-up is about 470°C. The pressure measured at the Venturi tube was 2.8bar. The cooling was turned on from 65-290° [1]. Figure 13 shows the temperature response of the optimised set-up (left part) with the

[1] In the hollow glass forming process the event sequence is defined in degrees due to the historical design of the machines using timing drums. One forming cycle corresponds to 360°. The cooling time of 65-290° can be related to a time in seconds by the known length of the forming cycle in s.

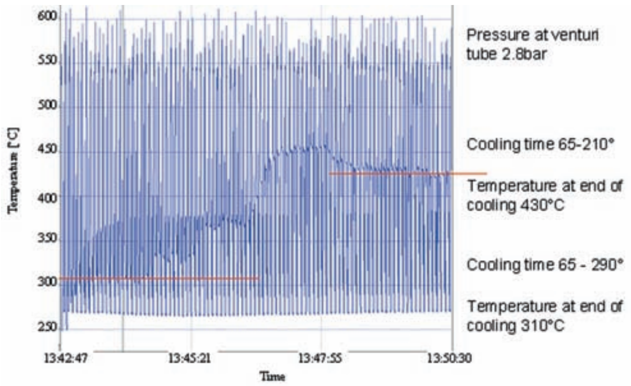


Figure 13: Temperature readings with optimised set-up changing the cooling time

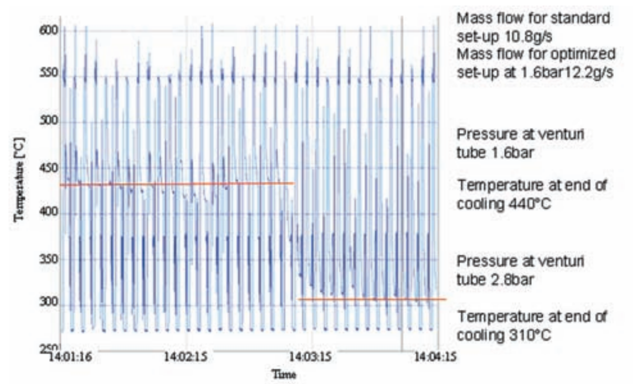


Figure 14: Temperature readings with optimised set-up changing the inlet pressure

same cooling time as for the standard set-up. The resulting steady state reference temperature for the optimised set-up is approximately 310°C. This corresponds to a temperature reduction of 160°C. This is well in accordance with the model prediction of 140°C.
Every increase in cooling capacity due to increased mass flow can be used for a reduction in either cooling time or pressure decrease. The pressure and the cooling time are reduced till the same steady state reference temperature is reached as with the original set-up.
If the cooling time is decreased to 65-210° (i.e. by 35%), the steady state reference temperature rises to 430°C as indicated in Figure 13. This is in the range of the steady state reference temperature for the standard set-up. If the pressure is decreased to 1.6 bar while keeping the same cooling time the steady state reference temperature rises to 440°C as shown in Figure 14. The temperature increase of 130°C when changing the pressure from 2.8 to 1.6 bar, corresponds reasonable with the model prediction of 105°C.
The measured mass flow for the optimised set-up at 1.6 bar is 12.2g/s, for the standard set-up 10.8g/s at 2.8 bar. In order to estimate the energy savings the following linear dependence of the energy consumption is assumed:

$$Q = \Delta p \cdot \dot{m} \cdot t_{cool}$$

Under this assumption about 35% less energy is needed with the optimised set-up to provide the same cooling capacity (to reach the same the steady state reference temperature respectively).

6. Summary

A steady state model to describe the flow and temperature distribution in plunger and cooler set-ups for the hollow glass industry was developed. The model is based on the commercially available CFD (Computational Fluid Dynamics) Code Fluent. The necessary boundary conditions to describe the coupled conservation equations for momentum, mass and energy are formulated and validated in laboratory experiments.
Applying the model parametric studies and design comparisons of existing designs were done. Furthermore the model was used to determine the governing factors for the plunger surface temperatures and identify interactions using the Design of Experiment method. The mass flow and the cooling capacity are close-coupled. The mass flow and the cooling capacity are mostly influenced by the cooling time, the pressure and the summed cooler bore area.
The model applicability was proven in field trials with a 33cl beer container. The model predictions regarding mass flow and temperature met the experimental results well. The plunger temperature could be reduced by 160°C when using the proposed optimised design, which in return can be used to save 35% energy for compressed air ot to speed up the process if the plunger cooling is limiting.

Further information/ author:
Dr.-Ing. Thomas Bewer,
Emhart Glass SA,
Cham, Switzerland

The work was done as project funded by the International Partners in Glass Research (www.ipgr.com). The author thanks the IPGR members for the collaboration and help, a special thank you to David Braithwaite for the discussions and experimental work done together.