Mathematical Comparison of Two VOD Nozzle Jets

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Studies of physical phenomena in a jet caused by VOD (Vacuum Oxygen Decarburization) nozzles have been carried out. The VOD process is a metallurgical process where the steel-making route is controlled under vacuum environment with oxygen top blowing. In this work, two VOD nozzle models have been employed for an investigation based on two real De Laval geometries used in industry. Numerical modeling was used to study oxygen blowing states of the nozzles at different temperatures and ambient pressures. The nozzle models were numerically computed with two dimensional domains, where vacuum conditions and temperatures were specifically defined. The modeling results showed that one of the nozzles was more applicably proper for lower pressures, displaying a more stable flow pattern. Furthermore, it was found that a change in ambient pressure has a stronger effect on the jet force than a change in ambient temperature. In addition, it was proved that the profiles of the dynamic pressure at a certain blowing distance fit well to Multi-Gaussian curves.

KEY WORDS: VOD; modeling; jet; blowing; pressure; temperature.

1. Introduction

The De Laval nozzle is widely used in the steel industry, especially in top blowing steel-making processes. This type of nozzle is characterized by a converging part before the nozzle throat and a diverging part after the throat. This, in turn, results in a compressible flow which gives rise to a high jet speed. One common case of employing the De Laval nozzle exists in the VOD steel-making process. In reality, a De Laval nozzle is supposed to be operated around designed pressures to avoid an incorrect expansion, which incurs an energy loss.

Many pieces of research have been carried out focusing on the nozzle top blowing processes. Ersson et al. employed both water-model experiments and mathematical modeling to study the effects of the top-blown jet on the steel bath in a LD converter. Also, Hirata et al. showed the stirring effect induced by top and side blown oxygen and bottom blown nitrogen converter. It showed that the appropriate combination of top, side and bottom blowing would produce a high HTE (heat transfer efficiency) and PCR (post combustion ratio). Nordquist et al. focused on the research in the top-blown water model. It was showed that the nozzle diameter, lance height, aspect ratio and the gas flow rate influence the penetration depth and the swirl phenomena. Naito et al. investigated jet incorrect expansion and multiple jets behaviors by cold and hot model experiments. Yuan et al. utilized a small-scale measurement with a double-parameter lance equipped to study the jet velocity and its deviation from the jet centerline. It was shown that by using the double-parameter lance the metallurgical performance could be improved. Also, Tago et al. used cold model experiments and numerical modeling methods to study single nozzles and multi-nozzles. They found that jet dynamic pressures and nozzle inclination angles could influence the jet properties. Wang et al. developed a three-dimensional mathematical model of multiple jets to comprehend the multi-jet coalescence and its dynamic power. Sumi et al. utilized both physical modeling and numerical modeling to investigate the effects of changeable ambient pressures on the top-blown jet. Besides, Sumi et al. studied the temperature effects on the jet behavior by the use of physical modeling. Also, in a preceding paper by Song et al., a VOD (vacuum oxygen decarburization) nozzle model was developed to investigate the flow properties by the effects of both ambient temperatures and pressures. However, little research about the mathematical comparison of temperature-pressure combined effects on real nozzles from the industry has been carried out. The jet force from a top-blown nozzle has not been fully understood. It would, for example, be of great interest to explore the oxygen flows of different oxygen nozzles used for varying ambient pressures at different ambient temperatures.

This work has focused on the comparison of two De Laval nozzles of different lengths which are being used in a production VOD vessel. It is aimed to uncover the different blowing behaviors affected by ambient pressures and temperatures by means of numerical modeling, and furthermore, to try to find the better nozzle for a specific ambient pressure. The flow patterns and dynamic pressures of two nozzles were compared for different ambient pressures. The temperature effects on the jets for the two nozzles were studied. Also, suggestions as how to choose the better nozzle for a certain ambient pressure were proposed. Besides, a Multi-Gaussian function was employed to fit profiles of the jet dynamic pressure.

2. Nozzle and Jet Theory

The typical characteristic of a De Laval nozzle is a con-
vergent and a divergent part. Fluids could be accelerated from the inlet through the nozzle to meet a velocity demand at the outlet. When it comes to the calculation of the fluid flow from a De Laval nozzle, the gas flowing in the nozzle is assumed to behave as a compressible ideal gas. In addition, the whole system is assumed to be isentropic, indicating that the process proceeds without the entropy change being frictionless and adiabatic.

The relations among different properties in a De Laval Nozzle, which can be used for the nozzle design, were adopted in the present work. To obtain the design point of a De Laval nozzle, the mass flow rate, inlet pressure and ambient pressure should be specified.

Three basic phenomena (Fig. 1) can occur in a jet when using a De Laval nozzle. When the ambient pressure is larger than the exit pressure, an over-expansion can happen. Also, when the ambient pressure is smaller than the exit pressure, an under-expansion can occur. Moreover, shock waves and expansion waves will be released if an incorrect expansion takes place. However, neither of the two phenomena is expected by people designing the nozzle. The optimum expansion of the nozzle would only be achieved if the ambient pressure is equal to the exit pressure. In the present work, calculations based on the nozzle theory have been carried out.

The jet momentum is a significant parameter when the jet impinging ability is examined on the steel surface. The jet momentum flux (jet force) can be expressed as:

\[ M = \rho v^2 \]  

Where \( \rho \) and \( v \) are the jet density and velocity, respectively. In this paper, we also employ the jet dynamic pressure to investigate the jet impinging force:

\[ P_{\text{jet}} = \frac{1}{2} \rho_{\text{gas}} v_x^2 \]

where \( \rho_{\text{gas}} \) is the blowing gas density and \( v_x \) is the gas velocity along the blowing axis.

3. Mathematical Modeling

3.1. Model Prototype

Two De Laval nozzles which are employed in the same VOD vessel were taken into consideration for the modeling. The geometries of them are schematically illustrated in Fig. 2. In this work, the left nozzle is named Nozzle A and the right one is named Nozzle B for the convenience of discussion. The inlet and outlet of Nozzle B are larger than those of Nozzle A, while its throat is narrower. Besides, the open angle for Nozzle B is nearly twice as big as for Nozzle A.

3.2. Numerical Assumptions

The two dimensional model domains of the two nozzles are shown in Fig. 3, along with the assigned boundary conditions.

For the mathematical modeling, a few assumptions were made:

- An axisymmetry
- A compressible ideal fluid
- A constant molecular viscosity
- A fully developed turbulence

3.3. Transport Equations

i) Equation of continuity:

\[ \rho v = \text{constant} \]

\[ \rho = \text{constant} \]

\[ v = \text{constant} \]

Fig. 1. Phenomena occurring in a jet. (From up to down: over-expansion, optimum-expansion and under-expansion).

Fig. 2. Geometries of Nozzle A and Nozzle B.

Fig. 3. Boundary conditions of Nozzle A and Nozzle B.
\[ \frac{\partial \rho}{\partial t} + \nabla \cdot \rho \mathbf{v} = 0 \] 

(3)

\[ \frac{\partial (\rho \mathbf{v})}{\partial t} - \nabla \cdot \rho \mathbf{v} \mathbf{v} = -\nabla p + \nabla \cdot (\rho \mathbf{v}) + \rho g \] 

(4)

where \( \nabla p \) is the pressure scalar gradient and \( \rho g \) represents the body force on the fluid. The notation \( \mathbf{v} \) is the momentum flux tensor, which can be expressed as:

\[ \mathbf{v} = \frac{\mathbf{V}}{\rho} \] 

(5)

where \( \mathbf{V} \) are the transpose of the velocity gradient tensor \( \nabla \mathbf{V} \). The quantities \( \mu \) and \( \mu_s \) are the viscosity and dilatational viscosity, respectively.

\[ \frac{\partial}{\partial t}(\rho E) + \nabla (\rho E + p) = \nabla (k_{eff} \nabla T) + \sum_i \mathbf{h}_i \mathbf{j}_i + \left( \mathbf{F}_{eff} \mathbf{v} \right) + S_i \] 

(6)

where \( k_{eff} \) is the effective conductivity and \( k_{eff} \nabla T \), \( \sum_i \mathbf{h}_i \mathbf{j}_i \), and \( \left( \mathbf{F}_{eff} \mathbf{v} \right) \) stand for energy transfer induced by conduction, species diffusion and viscous dissipation.\(^{16}\)

3.4. Turbulence Model

Previously, Ersson et al.\(^{25}\) has successfully utilized the model to look into jets for top blown converters. The simulation results were correlated with water models. Therefore, the standard \( k-\varepsilon \) model was employed in this work to describe the turbulence flow. This two-equation model is shown below with the turbulence kinetic energy \( k \) and turbulence dissipation rate \( \varepsilon \):\(^{17}\)

\[ \frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \frac{\sigma_k}{C_{\mu}} \rho \varepsilon C_k \varepsilon \] 

(7)

\[ \frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_k \frac{\varepsilon^2}{k} \] 

(8)

where \( k \) is the turbulence kinetic energy and \( \varepsilon \) is the turbulence dissipation rate. The parameters \( C_k, C_\varepsilon, \sigma_k \) and \( \sigma_\varepsilon \) are constants. Also, for the standard \( k-\varepsilon \) model, the turbulent viscosity \( \mu_t \) can be defined as:

\[ \mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \] 

(9)

where \( C_\mu \) is a constant.

3.5. Computational Domain and Boundaries

The two mathematical models including boundaries are already illustrated in Fig. 3. The meshing grids were created with quadrilateral cells of uneven sizes. The domain size is 0.4\( \times \)2 m with 171 609 and 135 367 number of cells for Nozzle A and Nozzle B, respectively. The mesh of the domain decreases its density gradually down from the nozzle outlet.

- **Inlet**
  A mass flow inlet is used as an inlet boundary condition for both nozzles. A mass flow rate of 0.34 kg/s was employed as the inlet mass flow rate and the temperature at the inlet would be the room temperature, 293.15 K.

- **Outlet**
  A pressure outlet boundary condition was employed as the outlet for both nozzles. Mass flow rate of 0.34 kg/s was employed as the outlet mass flow rate and the temperature at the outlet would be the room temperature, 293.15 K.

4. Results and Discussion

4.1. Calculation Comparison

In order to know the optimum outlet diameter under different ambient pressures, values of calculated nozzle outlet diameters are shown in Table 1. The throat dimensions of Nozzle A and Nozzle B are 35 mm and 25 mm, respectively. All ambient pressures employed in the present study are shown with a pressure unit of millibar (mbar). These data are based on the industrial parameters of the mass flow rate, the ambient pressure and the nozzle throat diameter.

Figure 4 shows the comparison of the real nozzle outlet and the calculated outlet for Nozzle A and Nozzle B. For the calculation results of Nozzle A, it is seen that the calculated outlet for an ambient pressure 400 mbar is the closest to the actual nozzle diameter. Nevertheless, the deviation between the real and the calculated outlet diameter becomes wider when the ambient pressure decreases from 200 mbar to 10 mbar. When it comes to the comparison of the results for Nozzle B, it is seen that for the ambient pressure 400 mbar, 200 mbar and 100 mbar, the calculation results are less than the outlet diameter of Nozzle B. However, the calculated diameter is larger than that for the ambient pressure.

Table 1. Calculation results from nozzle theory.

| Results from the nozzle theory | Nozzle A | Nozzle B |
|--------------------------------|----------|----------|
| Oxygen flow rate \( V_0 \) (Nm\(^3\)/h) | 850 | |
| Mass flow rate \( m \) (kg/s) | 0.34 | |
| \( T_0 \) (°C) | 20 | |
| \( p_0 \) (mbar) | 1414 | 2772 |
| \( d_{\text{喉}} \) (mm) | 1.41 D\(_a\) | 2.08 D\(_b\) |
| \( d_{\text{喉}} \) (mm) | 1.41 D\(_a\) | 2.08 D\(_b\) |
| \( d_{\text{喉}} \) (mm) | 1.41 D\(_a\) | 2.08 D\(_b\) |
| Nozzle Length (mm) | 1.29 D\(_a\) | 5.76 D\(_b\) |
| Open Angle (°) | 4.289 | 9.772 |
| Calculated outlet \( d_{\text{喉}} \) (mm) | 1.08 D\(_a\) | 1.26 D\(_a\) |
| Ambient pressure \( p_0 \) (mbar) | 400 | 200 |
| 100 | 100 | 10 |
In order to understand the discrepancy between the real and the calculated outlet diameter for both nozzles, it is necessary to study the flow patterns in detail.

4.2. Flow Patterns

Mach number contours for these two nozzles are showed to find out the effects of the ambient pressure on the flow pattern. Four ambient pressures (400 mbar, 200 bar, 100 mbar, 10 mbar) that are present in a real VOD vessel, were exerted on the domain outlet boundaries for a temperature of 1873 K. This is assumed because the steel temperature in the VOD vessel could be around 1600°C.

Figure 5 displays the Mach number contours of both nozzles from the overall view. The distinct difference between Figs. 5(a) and 5(b) would be the flow pattern around the outlet, where the flow is choked in the case of Nozzle B. Figure 6(b) clearly shows the phenomenon of choking, with a sudden increase of Mach number to around 3 followed by a sharp drop. In addition, it can be noticed that the separations between the nozzle wall and the flow occur at the fringe of the outlet. On the contrary, Nozzle A (see Fig. 6(a)) doesn’t show a rapid velocity drop, but a velocity attenuation with oscillations. From the results it can be concluded that the ambient pressure of 400 mbar for Nozzle B is so high that it causes an over-expansion behavior. Furthermore, Nozzle A works normally for this pressure. These observations correspond to the nozzle theory calculations for a 400 mbar ambient pressure, where the calculated outlet diameter for Nozzle A is closer to its real outlet diameter. Hence, this results in a smoother blowing, than for Nozzle B, where the calculated diameter is quite lower than its real diameter resulting in a choked flow.

Figure 7 shows the Mach number contours for Nozzle A and B for the ambient pressure 200 mbar. It can be seen that this time the flow can extend more from Nozzle B (Fig. 7(b)) instead of choking at the outlet as in the previous case (Fig. 6(b)). A clear under-expansion can be seen for Nozzle A, while an obvious over-expansion can be observed for Nozzle B. This agrees with the calculations where the calculated outlet diameter for Nozzle A is higher than its real diameter. However, for Nozzle B it is lower.

Heavier under-expansion takes place for Nozzle A when the ambient pressure is decreased to 100 mbar (Fig. 8(a)). Moreover, the flow is still over-expanded for Nozzle B for this specific ambient pressure (Fig. 8(b)). However, the expansion situation is better compared to the former case (Fig. 7(b)). Furthermore, the flow from Nozzle A becomes much more unstable than that for Nozzle B. This could indicate that for this certain mass flow rate, the ambient pressure...
100 mbar is a little low to be used for Nozzle A. However, nonetheless it is still too high to be used for Nozzle B. Also, the calculation results show two different calculation deviations for both nozzles.

Figure 9 illustrates the Mach number contours of both nozzles for an ambient pressure of 10 mbar. In Fig. 4 we can notice that the calculated outlet diameters for a 10 mbar pressure for both nozzles are way higher than their real outlet diameters. This is characterized by a heavy under-expansion flow in Fig. 9. For Nozzle A, a big Mach disk forms near the nozzle outlet and a velocity vacuum appears. Despite a heavy under-expansion flow, the situation for Nozzle B is better and the flow remains continuous, without a drastic velocity drop behind the Mach disk. It can be concluded from the results that an extremely low ambient pressure might induce a severe turbulence, which results in a failed blowing.

4.3. Blowing Velocity

The modeling results of both nozzles can show the Mach number change along the axis as well as reflect its attenuation. Comparisons have been made to understand the blowing behaviors of the two nozzles for an outlet temperature of 1873 K and for different ambient pressures (400 mbar, 200 mbar, 100 mbar, 10 mbar). These pressure data were taken from an existing VOD plant.

Figure 10 shows the Mach number changes of both nozzles along the blowing axis for the following ambient pressures, 400 mbar, 200 mbar, 100 mbar and 10 mbar. For Nozzle A (Fig. 10(a)), it shows that the Mach number increases to a peak value. Thereafter, it decreases gradually with slight oscillations for the ambient pressure of 400 mbar.
The oscillation becomes heavier when the ambient pressure drops to a value of 200 mbar. Furthermore, it is seen that the Mach number in the whole jet area in this case is larger than that for a 400 mbar pressure value. The same tendency as for the previous two ambient pressures is seen for a 100 mbar pressure. However, the flow fluctuates with even larger oscillations at the start. In addition, a higher Mach number value can be obtained throughout the whole domain. In the Mach number profile for an ambient pressure of 10 mbar, the Mach number sharply decreases to a subsonic value after a steep increase, which is represented by a Mach disk (Fig. 9(a)). Due to the stiff oscillation of the jet for this ambient pressure, the whole domain is not able to reflect all flow fluctuation in this case.

Different observations are shown for Nozzle B in Fig. 10(b). The Mach number straightly drops after the initial acceleration for the ambient pressure of 400 mbar (also in Fig. 6(b)). When the ambient pressure is decreased to 200 mbar, where the over-expansion still exists, obvious oscillations occur. These are gradually weakened along the axis. In addition, a higher Mach number value can be obtained throughout the whole domain. In the Mach number profile for an ambient pressure of 10 mbar, the Mach number sharply decreases to a subsonic value after a steep increase, which is represented by a Mach disk (Fig. 9(a)). Due to the stiff oscillation of the jet for this ambient pressure, the whole domain is not able to reflect all flow fluctuation in this case.

4.4. Jet Dynamic Pressure

A high jet force can yield a strong impinging effect on the steel surface which leads to good reaction and stirring conditions. To investigate the jet force, the dynamic pressures of jets from both nozzles were compared. It was assumed that the steel bath was located 1.3 m straight down from the nozzle. Therefore, the jet dynamic pressure at a blowing distance of 1.3 m was investigated. The temperatures were still 1 873 K at the outlet boundaries. Furthermore, the jet dynamic pressure is displayed with a pressure unit of Pascal (Pa) in order to avoid the confusion to the ambient pressure (mbar).

Figure 11 shows the dynamic pressures for both nozzles for four ambient pressures. In the case of an ambient pressure of 400 mbar, the dynamic pressure of Nozzle A at 1.3 m is around 2 500 Pa higher than that of Nozzle B (2 000 Pa). This can be explained by the calculations and flow patterns. More specifically, the flow from Nozzle B for 400 mbar is choked at the outlet, bringing about a limitation of the blowing velocity. As the ambient pressure is lowered to 200 mbar, the dynamic pressures for both nozzles increase and become close, which are more than 3 500 Pa. Furthermore, for the flow patterns, the flow for Nozzle A shows an under-expansion while the flow for Nozzle B shows over-expansion. When the ambient pressure is decreased to 100 mbar, the dynamic pressures for both nozzles increase and become close, which are more than 3 500 Pa. Furthermore, for the flow patterns, the flow for Nozzle A shows an under-expansion while the flow for Nozzle B shows over-expansion. When the ambient pressure is decreased to 100 mbar, the opposite situation arises. More specifically, the dynamic pressure of Nozzle B is higher than that of Nozzle A. In this case, a little heavier under-expansion is shown for Nozzle A and a minor over-expansion is seen for Nozzle B. However, both nozzles give rise to a lower dynamic pressure for the ambient pressure of 10 mbar, which is due to the failed blowing for Nozzle A and the ambient pressure of 10 mbar makes the flow extremely unstable, which results in heavy fluctuations. A supersonic region (Ma>1) can also be compared here to investigate the supersonic flowing distance. In the case of Nozzle A, the supersonic region increases from 0.57 m to 1.21 m when the ambient pressure decreases from 400 mbar to 100 mbar. The supersonic region of Nozzle B increases from 0.55 m to 1.44 m. It can hence be concluded that a lower ambient pressure improves the blowing velocity. On the other hand, it is of interest to note that the supersonic region of Nozzle B for the ambient pressures of 200 mbar and 100 mbar (0.92 m, 1.44 m) is larger than that for Nozzle A (0.85 m, 1.21 m). However, for the ambient pressure of 400 mbar (0.55 m), it is smaller than for Nozzle A (0.57 m). Therefore, it can be concluded that Nozzle B is more suitable for a lower ambient pressure, than Nozzle A in producing higher velocities.
unfinished oscillation for Nozzle B. Still, Nozzle B shows a higher dynamic pressure than Nozzle A around a blowing distance of 1.3 m. To summarize, it can be concluded that Nozzle B would produce a stronger impinging force for lower ambient pressures than Nozzle A would.

4.5. Jet Force

The jet total force over the blowing area at the same blowing distance of 1.3 m was captured in the modeling by integrating the jet dynamic pressure over the cross-sectional area of the domain at 1 873 K. Table 2 shows the jet force for different ambient pressures for both nozzles at a blowing distance of 1.3 m and at 1 873 K. From 400 mbar to 10 mbar, it can be noticed that for both nozzles, the jet force continues to increase, from 135.33 N to 181.38 N for Nozzle A and 122.83 N to 216.96 N for Nozzle B. This would be attributed to the ambient pressure decrease, resulting in a jet force increase to meet the energy conservation. Moreover, as the same trend found in the dynamic pressure, Nozzle B is more competent for a lower ambient pressure with respect to giving a higher jet force.

4.6. Temperature Impact

Three different temperatures, 1 873 K, 1 000 K and 293.15 K at the outlet boundaries of both nozzles were simulated to investigate the influence of the thermal effects on the oxygen jet characteristics. In addition, the influence of three ambient pressures (400 mbar, 200 mbar and 100 mbar) on the jet characteristics was simulated.

The temperature effects on the jet velocities for nozzle A for different ambient pressures are shown in Fig. 12. It displays that a typical curve with oscillations at the start becomes smooth as the blowing distance increases. It is seen that higher temperatures yield higher blowing velocities after oscillations for all ambient pressures. In addition, no big difference can be found within the oscillation period for different temperatures. Also, the oscillations become larger with a decreasing ambient pressure, because the phenomenon of under-expansion is more serious. For Nozzle B, Fig. 13 shows a similar trend with respect to the velocity change as observed for Nozzle A. The velocities at higher temperatures for these three ambient pressures are always larger than those at lower temperatures. However, the oscillation extent becomes weaker as the ambient pressure decreases and the blowing tends to be smoother. This is due to that the over-expansion becomes less obvious. The lower velocities at lower temperatures result from the higher density of the surrounding gas at lower temperatures. A higher density of the surrounding gas results in a higher resistance to the flow of the jet.

Figures 14 and 15 show the temperature effects on the dynamic pressures for different ambient pressures. A positive effect on the jet is found with a temperature increase. In the case of Nozzle A (Fig. 14), the dynamic pressure for a higher temperature is only slightly larger than that for a lower temperature after the oscillations before a distance around 1 m from the nozzle. However, the dynamic pressure values for different temperatures converge at the end of the domain. The difference in dynamic pressures for Nozzle B (Fig. 15) becomes even smaller, showing that the three curves of dynamic pressures are very close to each other. Nevertheless, for the initial part (before 1 m) from the nozzle exit of the curves after the oscillation region, it can be

| Temperature: 1 873 K | Jet Force (N) |
|---------------------|--------------|
| Ambient Pressure     | 400 mbar     | 200 mbar     | 100 mbar     | 10 mbar     |
| Nozzle A             | 135.33       | 159.95       | 171.87       | 181.38      |
| Nozzle B             | 122.83       | 166.39       | 190.81       | 216.96      |

Fig. 12. Centerline velocities along the blowing axis for Nozzle A.

Fig. 13. Centerline velocities along the blowing axis for Nozzle B.
still noticed that the dynamic pressures for these three ambien
t pressures are higher at higher temperatures. This also
means that the jet can move with a high force in a low-
resistant ambiance even though undergoing a density
decrease. From the results so far shown, it can be concluded
that an increased temperature favors an increase of the
dynamic pressure in a certain region of the domain. How-
ever, the effects are quite small compared to a variation in
ambient pressure. In addition, the temperature effects for
Nozzle A seem to be larger than for Nozzle B.

Overall, the temperature change shows no contribution to
the alteration of the jet force. This can be noticed in Tables
3 and 4, which reveal the jet force for different temperatures
for three ambient pressures. Horizontally considering the jet
force value in the tables, it shows that temperature change
is not favorable with respect to the increase of the total jet
force. The comparison indicates that increasing the temper-

ature is not as satisfactory as decreasing the ambient pres-
sure when a high jet force is desired.

5. Metallurgical Discussion

In an industrial blowing process, a higher jet impinging
force is wanted, resulting in a deep penetration depth and an
improved reaction efficiency. Flinn et al.\textsuperscript{[18]} and Koria et al.\textsuperscript{[19]} have reported that a higher oxygen total pressure
increases the penetration dept h. In the present work, from
the results and discussion above, it would be concluded that
the jet dynamic pressure can be improved through another
two ways, more specifically, by decreasing the ambient
pressure or increasing the ambient temperature.

From the dynamic pressure analysis, however, it can be
noticed that after a certain blowing distance, the dynamic
pressures of both nozzles become close to each other. In
order to show the dynamic pressure for a further blowing
distance away from the nozzle outlet, the blowing distance
of 1.3 m is still taken into account for the study. Figures
16(a) and 16(b) reflect both temperature and ambient pres-
sure effects on Nozzle A and B respectively. Different
results can be noticed at a distance of 1.3 m from the nozzle
exit for Nozzle B. More specifically, higher temperatures
cannot all the time give rise to high dynamic pressures. This
is especially true for the cases of the temperatures of 293 K
and 1000 K. Here, the dynamic pressures for a 293 K tem-
perature are larger than that for a 1000 K temperature for
the cases of Nozzle A (400 mbar), Nozzle B (200 mbar and
100 mbar). This can somehow be explained by that the jet
energy attenuates at the end of the domain. Thus, it will not
be able to support a higher dynamic pressure at a higher
temperature. On the other hand, it is clearly seen that on
average the jet dynamic pressure is always higher at a lower
ambient pressure for both nozzles. In addition, the temper-

ature in a real VOD vessel is not possible to alter within rea-
sonable values. However, the ambient pressure changes from 1 bar at the start of the blowing to approximately 10 mbar for the last five minutes of the blowing under production conditions. This means that for a normal VOD operation, the changes in ambient pressure would have a larger impact on the jet dynamic pressure than the temperature changes would.

Furthermore, Nozzle A shows a large dynamic pressure difference between 1 873 K and 293.15 K. However, the case of Nozzle B indicates a smaller dynamic pressure difference. Hence, this implies that the temperature change has a greater effect on Nozzle A than on Nozzle B for an environment of low ambient pressures.

Figures 17(a) and 17(b) show the dynamic pressure profiles at the blowing distance of 1.3 m from the nozzle exit at 1 873 K for Nozzle A and Nozzle B respectively. For ambient pressures of 400 mbar, 200 mbar and 100 mbar, it is found that the dynamic pressure curve fits well to the Multi-Gaussian distribution for both nozzles. However, the curve representing a 10 mbar ambient pressure does not show a good correlation with a Gaussian distribution for the two nozzles. This is probably due to the rigorous blowing environment.

The Gaussian fitting equations for an ambient temperature of 1 873 K are shown in Table 5 below. From the fitting results above, it is seen that the dynamic pressure value abides well by the Multi-Gaussian distribution. In addition, by integrating the jet momentum flux over the cross-sectional area of the domain at the blowing distance of 1.3 m, the jet impinging force can be obtained from Eq. (10):

\[ F = \int_{0}^{2\pi} \int_{0}^{r_{\text{max}}} \rho \theta v_{z} r dr d\theta \] \hspace{1cm} (10)

where \( F \) is the jet force and \( r_{\text{max}} \) is the blowing cross-sectional radius, which is 0.4 m in this work. In connection with \( P_{\text{jet}} \), the jet force can be expressed as:

\[ F = 2 \int_{0}^{2\pi} \int_{0}^{r_{\text{max}}} P_{\text{jet}}(y) dy d\theta \] \hspace{1cm} (11)

By employing the fitting functions, the jet force can be calculated from Eq. (11) and compared to that obtained from the mathematical modeling. Table 6 shows calculated jet force values from Multi-Gaussian fitting and mathematical modeling. The comparison of the error calculations shows that the jet force values from the fitting and from the mod-
Table 6. Comparison of jet force calculations from fitting and modeling results.

| Ambient Pressure (mbar) | From Multi-Gaussian fitting \((F_f)\) | From mathematical modeling \((F_m)\) | Error \(\%\) | \(\frac{F_f - F_m}{F_m}\) |
|-------------------------|---------------------------------|---------------------------------|----------|------------------|
| Nozzle A                |                                 |                                 |          |                  |
| 400                     | 135.07                          | 135.33                          | 0.19     | 0.19             |
| 200                     | 158.89                          | 159.95                          | 0.66     | 0.66             |
| 100                     | 168.53                          | 171.87                          | 1.94     | 1.94             |
| Nozzle B                |                                 |                                 |          |                  |
| 400                     | 122.42                          | 122.83                          | 0.33     | 0.33             |
| 200                     | 165.33                          | 166.39                          | 0.63     | 0.63             |
| 100                     | 187.15                          | 190.81                          | 1.91     | 1.91             |

6. Conclusions

Two production nozzles with different geometries (named by Nozzle A and Nozzle B in this paper) from industry were mathematically compared with respect to different ambient pressures as well as temperatures. A correlation was found between the calculated results and the nozzle geometries. In addition, Mach number contours and dynamic pressures were investigated to compare the two nozzles’ properties. The specific findings within this work can be summarized as follows:

- Mathematical results of both nozzles showed good correlations between hand-calculations and simulations. In addition, the flow pattern of Nozzle B seems better than the one using Nozzle A at a very low ambient pressure.
- Lower ambient pressures increase the blowing velocity. Nozzle A can produce a higher velocity region for higher ambient pressures, while Nozzle B is more stable to provide higher velocities for lower ambient pressures. An extremely low ambient pressure makes the flow fluctuate heavily.
- The analysis of the jet dynamic pressure is in conformity with the Mach number analysis. The blowing distance 1.3 m from the nozzle was taken as the comparison standard (i.e., an approximate distance between the nozzle and bath surface). It showed that Nozzle B gives rise to a higher jet dynamic pressure than Nozzle A for low ambient pressures. Moreover, slightly lower ambient pressures improve the dynamic pressures of the jet for both nozzles.
- The modeling predictions showed that the temperature change has a high impact on the velocity distribution. This indicates that higher ambient temperatures improve the jet velocity. However, the temperature’s effect on dynamic pressures is not as noticeable as on velocities.
- It could be found that a reasonably decreased ambient pressure can to a larger extent increase the dynamic pressures of the jet than an increased ambient temperature. Furthermore, a change in ambient temperature has a stronger effect on the jet dynamic pressure for Nozzle A than for Nozzle B. It should be noted that the knowledge of temperature and pressure effects should be used a priori when designing the nozzle. Also, in the absence of a variable nozzle geometry, the effects of altering the temperature and pressure outside the nozzle design parameters can be known.
- There is no indication that higher ambient temperatures support higher total jet forces. The jet force profile at a blowing distance of 1.3 m conforms favorably to the Multi-Gaussian fitting.
- It is concluded from this work that Nozzle B should be employed for a lower ambient pressure. Furthermore, Nozzle A is more suitable to use at a higher ambient pressure. At the same time this proposes a possibility of the usage of geometry-changing nozzles in a real VOD vessel.

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Nomenclature

\(\rho_{bl} \) Blowing gas density \((\text{kg m}^{-3})\)
\(m \) Mass flow rate \((\text{kg s}^{-1})\)
\(M_0 \) Mach number
\(M \) Jet momentum flux \((\text{kg m}^{-1} \text{s}^{-2})\)
\(P_{jet} \) Jet dynamic pressure \((\text{Pa})\)
\(V_c \) Gas velocity along the blowing axis \((\text{m s}^{-1})\)
\(D_A \) Throat diameter of Nozzle A \((\text{m})\)
\(D_B \) Throat diameter of Nozzle B \((\text{m})\)
\(k \) Turbulence kinetic energy \((\text{m}^2 \text{s}^{-1})\)
\(\varepsilon \) Turbulence dissipation rate \((\text{m}^2 \text{s}^{-3})\)
\(k_{eff} \) Effective conductivity \((\text{W m}^{-1} \text{K}^{-1})\)

\(F \) Impinging jet force \((\text{N})\)

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