Development of Swirling Flow Generator in Immersion Nozzle

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(Received on June 3, 1999; accepted in final form on January 13, 2000)

1. Introduction

Recently, it has been acknowledged that swirling motion in the immersion nozzle is remarkably effective to control the fluid flow pattern in both the slab and billet type molds; a very uniform velocity distribution is established within a very short distance downstream of the outlet of the immersion nozzle, i.e., the penetrating distance of inclusions is decreased,1) in case of slab-mold meniscus-turbulence, generation of vortex and generation of self-excitation-vibrations in the bulk mold flow are depressed, and quite stable outlet-flow pattern of the immersion nozzle is realized.2) In the refining process, it has been cleared recently that it is very useful to make a fine bubble using centrifugal force acting on the bubble through the swirling motion in the pouring tube.3) Also, making a bubble curtain in the pouring tube with swirl flow is considered to be an effective method to trap the inclusions contained in the molten steel.4) Therefore, it is very important to develop an as simple and stable swirling generator in its structure as possible. Here, a twist-tape-swirling blade shown in Fig. 1 is proposed to fill the above-mentioned conditions. The twist-tape-swirling generator has been formerly used in heat transfer field and so forth.5,6) However, applying it in the metallurgical process, particularly in the flow control of the immersion nozzle, its characteristics have not been well clarified. Therefore, using numerical and experimental methods, effectiveness in applying it in the immersion nozzle was investigated.

2. Governing Equations

A specific diagram of the calculation model is shown in Fig. 2. The elliptic partial differential equations governing this problem are of the general form

\[ \frac{\partial}{\partial x} \left( \rho u \phi - \Gamma_x \frac{\partial \phi}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho w \phi - \Gamma_y \frac{\partial \phi}{\partial y} \right) + \frac{\partial}{\partial z} \left( \rho w \phi - \Gamma_z \frac{\partial \phi}{\partial z} \right) = S_{\phi} \]

Fig. 1. Swirling blade of twisted tape.
where \( u, v, \) and \( w \) represent the time-averaged velocities in the three directions of the Cartesian coordinate system, respectively.

The variable \( f \) represents various time-averaged quantities, i.e., mean velocities, turbulence kinetic energy, and turbulence energy dissipation rate. The quantity \( G_f \) represents the diffusivity of the transport variable \( f \). Pressure forces, body force and generation are contained in the source term, \( S_f \).

Various transport equations and the standard \( k \)-\( e \) turbulence model\(^7\) are listed in Table 1.

### 3. Experimental Apparatus

Figure 3 shows the experimental apparatus used in the water model test. A constant fluid velocity is obtained through the nozzle using an overflow tank. Water is used as test liquid. Its flow rate is 1 m/s or 1.5 m/s.

A straight nozzle of inner diameter 40 mm \( \varnothing \) and water jacket \( \varnothing \) are set to observe the magnitude of swirling. The twist-tape-swirling blade \( \varnothing \) having thickness of 7 mm are inserted into the nozzle tube of an inner diameter of 40 mm and length of 200 mm.

### 4. Results and Discussion

Figures 4 to 6 show the experimental and calculated radial profiles of the tangential velocity for various twist ratios (180° twist pitch length/inner diameter \( D \)). The \( X \) and \( Y \) direction illustrated in Fig. 3 show the measurement and calculation directions across the inner diameter at the position of 25 mm downstream of the outlet of the swirling.
Fair coincidence can be observed between the calculated and measured results just after the outlet of the swirling blade. The Rankine-vortex having both the forced vortex (namely, tangential velocity \( w = \omega r \)) in the inner region \((r < 15 \text{ mm})\) and the free vortex (namely, tangential velocity \( w = K/r \)) in the outer region \((r > 15 \text{ mm})\) can be seen in Figs. 4 to 6. Here, \( K \) is a constant and \( r \) is the radial distance. It can be considered that the anisotropy of the separation of the outlet flow because of the swirl blade thickness in the outlet of the blade contributes to the discrepancies between the experimental and calculated results shown in these figures.

Figure 7 shows a calculated development of velocity vectors between an upstream position before the entrance of the swirl blade and a downstream position of 115 mm after the outlet of the swirl blade. A slightly strained tangential velocity profiles just below the outlet of the swirl blade (e) are observed through the effect of the swirl blade thickness. However, considerable axi-symmetric tangential velocity profiles can be seen at a downstream position of 25 mm after the outlet of swirl blade (f).

Setting a swirling blade in the nozzle, calculated developments of tangential velocity from the front of the swirl blade to the downstream positions of the swirl blade are shown in Fig. 8 for the various radial positions, 7, 13, 17 mm from the nozzle axis for a twist ratio of 1.

The tangential velocity begins to be accelerated from the front of the swirl blade, increases significantly in the order of decreasing radial distance passing through the first part of the swirl blade, whereas it begins to be decelerated in the order of decreasing radial distance passing through the latter half of the swirl blade and then gradually declines a little with flow going down after passing through the swirl blade. Such a tendency can be seen for every twist ratio.

In particular, both the developments of the tangential velocity along the swirl blade for the cases of twist ratios 1/2 and 3/8 have quite the same tendency. Therefore, it is cleared that swirling flow is generated by making use of the twist-tape-blade while the liquid-flow streaming through along the twist-tape of the swirl blade.

Figure 9 shows a relationship between the twist ratio and swirl number. The used swirl number \( Sw \) (shown in
Appendix) is defined as follows;

$$Sw = \frac{2W}{3U}$$

where $W$ and $U$ are the tangential and axial mean velocities across the nozzle tube at the downstream position of 25 mm after the outlet of the swirl blade.

The swirl number increases with decreasing the twist ratio.

Figure 10 shows a relationship between the dimensionless radial position and tangential velocity deduced from the experimental and calculated results at the distance of 25 mm from the outlet of the swirling blade. The calculated results fairly coincide with each other for the various twist ratios, respectively. However, there are some systematic differences in the calculated and experimental results from the ratio from 1/2 to 1/1. Namely, the magnitudes of dimensionless tangential velocities result in becoming larger in the order 1/2, 3/4, 1/1. These tendencies seem to be explained by the phenomenon that the energy-losses (friction losses) due to flow through in the swirling blade from the entrance to the outlet of the swirl-blade, become larger in order 1/1, 3/4, 1/2 because the energy-losses (friction losses) being proportional to the square of each velocity. But, the dimensionless radial positions vs. tangential velocities for various twist ratios can be approximately represented by the solid line for the twist ratio of 3/4. Accordingly, tangential velocity required can be deduced through knowing the axial velocity of flow through the nozzle and twist ratio of the swirl blade inserted into the nozzle.

Figure 11 shows the developments of calculated static pressure from the entrance of the swirl blade to the downstream positions through the outlet of the swirl blade (z-position 0) for various swirl twist-ratios. The static pressures rapidly decrease between the entrance and outlet of the swirl-blade, become larger in order 1/1, 3/4, 1/2 because the energy-losses (friction losses) being proportional to the square of each velocity. But, the dimensionless radial positions vs. tangential velocities for various twist ratios can be approximately represented by the solid line for the twist ratio of 3/4. Accordingly, tangential velocity required can be deduced through knowing the axial velocity of flow through the nozzle and twist ratio of the swirl blade inserted into the nozzle.

Fig. 7. Flow pattern at cross section of various points. Twist ratio, 3/4.

Fig. 8. Development of tangential velocity with increasing z position. Twist ratio, 1/1.

Fig. 9. Swirl number vs. twist ratio.

Fig. 10. Dimensionless tangential velocity $w'$ vs. radial position $r/R$ for various twist ratios.
the created magnitudes of tangential velocities become larger in order, 1/1, 3/4, 1/2 in the range from the entrance to the outlet of the swirl blade. Accordingly, the energies consumed in the creating tangential velocities and friction-losses due to flow through the swirling blade from the inlet to the end, become larger in the order, 1/1, 3/4, 1/2. After leaving the outlet of the swirling blade, the consumed energy is attributable only to friction loss. It is also observed that it consumes much energy to create the tangential velocity between the entrance and the outlet of the swirl blade compared with the consuming friction loss in the ranges from the outlet of the swirl blade to downstream positions.

Figure 12 shows a calculated relationship between the peripheral average of the dimensionless static pressure difference and the twist ratio. The pressure difference was evaluated between the inlet and outlet of the swirl blade. These relationships for the cases of the axial-velocities, 1 m/s and 1.5 m/s, are quite the same. The necessary pressure difference can be obtained if the axial velocity and the twist ratio of the swirl blade are given.

5. Conclusions

(1) The swirl number and pressure difference increase with decreasing the twist ratio of the swirling blade.

(2) Swirl flow begins to be generated at the front of the swirl blade, and then the tangential components of velocity are accelerated in the order of decreasing radial distance in the first part of the swirl blade, while those are decelerated in the order of decreasing radial distance in the second part of the swirl blade.

(3) A slightly strained tangential velocity can be seen just below the swirling blade, but that becomes axi-symmetrical at a further downstream position.

(4) Numerical results obtained by the $k$–$\varepsilon$ turbulence model and experimental results coincide fairly well with each other for the tangential velocity distributions at the outlet of the swirl blade.

(5) Knowing the axial velocity and the twist ratio of the swirl blade, the corresponding tangential velocity and pressure difference can be deduced through the relationship between the dimensionless radial position and tangential velocity.

Acknowledgement

This work was partially supported by the research fund on iron and Steel from Iron and Steel Institute of Japan.

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Appendix

As discussed by Dilawari and Szekely, the swirl number, $Sw$, which is the ratio of the axial flux of the azimuthal momentum to the axial momentum, normalized by an appropriate radius $Ro$ (taken here as the nozzle radius) is given by

$$Sw = \frac{G_\theta}{G_R} \tag{A-1}$$

$$G_\theta = \int_0^R \rho \omega w^2 r dr \tag{A-2}$$

And the axial flux of axial momentum (neglecting the pressure term) is

$$G_z = \int_0^R \rho u^2 r dr \tag{A-3}$$

If characteristic values for $u$ and $w$ are to be constant, denoting them as $U$ and $W$, the swirl number can be obtained as follows;

$$Sw = \frac{1}{2} \rho UWR^3 = \frac{2W}{U} = \frac{2W}{U} \tag{A-4}$$