Three-dimensional temperature and stress field simulation with all-hexahedral element mesh in a high efficiency cooling structure for the fabrication of amorphous ribbons

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Abstract
Planar flow casting is an advanced technology to produce amorphous and nanocrystalline ribbons in electrical application. At present, the utilization rate of roller surface is generally under 20% in industrial production. This work reports a new cooling structure that is designed to produce ribbons width 65 mm with the utilization rate of roller surface up to 65%. The cooling structure is simulated with the software ANSYS by employing three-dimensional temperature and stress field mathematic model incorporating actual production conditions with all-hexahedral element mesh. It is shown that the liquid velocity is uniform with consistent distribution of convective heat transfer coefficient in width direction between cooling water and rotating roller. The thermal expansion difference in the new cooling structure is only 0.0042 mm at the ribbon edge. In this case, it is unnecessary to make the slit with an upward bend. The new cooling structure is expected to be applied for industrial production at an ideal level.

1. Introduction
Planar flow casting (PFC) is an advanced technology to produce amorphous and nanocrystalline ribbons [1]. Figure 1 illustrates the schematic of the PFC process. In the process, molten metal is ejected onto a cooling roller surface from a crucible through a rectangle slit nozzle. A puddle constrained by the surface tension is formed between the nozzle and the cooling roller, adheres to substrate wheel. Then it is solidified into a ribbon with a high cooling rate \((10^6 \, ^\circ\text{C} \, \text{s}^{-1})\). The ribbon is dragged from the roller by air blower at near 1/4 of a rotation cycle [2].

Amorphous/nanocrystalline alloy ribbons are used in a wide range of industry fields such as pulse power devices, sensors and telecommunication devices for a new generation of soft magnetic materials due to their excellent magnetic properties compared to classical silicon or ferrite steel [3–5]. The key factors to produce ribbons with good properties are the cooling rate of the alloy, the thermal transfer and the thermal expansion of the cooling roller. It is important to analyze the thermal transfer and thermal expansion of the cooling roller used in the PFC process [6].

The variations in the thermal transfer and thermal expansion are determined by the puddle, cooling roller and the coolant. However, previous studies were mainly focused on puddle formation and temperature distribution between puddle and roller. Busmann et al [7] performed a numerical study of steady flow and temperature fields considering inertial, viscous, surface tension, and wetting effects by the momentum and energy equations. Byrne et al [8] developed a semi-empirical analysis of puddle pressure. They found an operating window of pressure bounds to predict the capillary stability limits in a liquid metal puddle during PFC process. Bichi et al [9] introduced a mathematical model to describe the solidification with z-component of vorticity of the fluid flow in the PFC process. Liu et al [10] used a 2D fluid flow model to analyze the initial development for amorphous ribbon formation in the PFC process.

In recent years, investigators began to pay attention to the effect of cooling roller on ribbon properties. Li et al [6] analyzed the three-dimensional thermal transfer and thermal expansion of the cooling roller using variable
heat flux that acted on the cooling roller as a boundary condition. They found that the thermal expansion has its maximum value in the middle width direction so that the nozzle should be adjusted upward with a slight bend of slit to keep gap constant and ribbon thickness consistent.

In the above studies, cooling water was assumed as uniform liquid with constant velocity and the effect on the temperature of cooling roller was neglected. Sowjanya used a constant temperature to replace cooling water in employing the numerical model \[11\]. Thus, the cooling structure that is called crystallizer was investigated scarce in PFC process. In fact, the water velocity distribution plays a significant role during the PFC process. The two kinds of crystallizers with water channel structure (WCS) and water gap structure (WGS), which are widely used in industrial production, can cause flow velocity of water to become irregular and distribution of convective heat transfer coefficients to become chaotic. Regular flow and convective heat transfer coefficient distribution are considered as important references for producing wider ribbon \[12\].

In this work, we design a new cooling structure and demonstrate flow distribution and convective heat transfer coefficient distribution through three-dimensional steady time-independent simulation by finite element analysis during the PFC process. We find that in the new crystallizer the uneven-rate of liquid velocity is only 1.47% in width direction, which is considerably lower than 355% in WCS structure and 105% in WGS structure. The uneven-rate of convective heat transfer coefficients in the new crystallizer is only 1.33%, considerably lower than in WCS structure 141% and in WGS structure 169%. The maximum ribbon width is designed as 65 mm, almost twice as large as the ribbon width of 35 mm in WGS structure. The utilization rate of roller surface reaches up to 65%, far greater than the utilization rate of 18% in WCS and WGS. Thermal expansion of the cooling roller is calculated with thermal structure coupling method. The simulation results show that the thermal expansion difference of the new crystallizer is 0.0042 mm at the ribbon edge. It is unnecessary to machine the slit nozzle upward bend because of such small difference.

### 2. Models and calculation methods

#### 2.1. Flow velocity simulation

In our mathematical model of the cooling structure, we assumed a three-dimensional and steady-state simulation. The water flow is incompressible and Newtonian turbulent liquid on the basis of the production practice.

The model is formulated based on the following assumptions:

1) Since the thermal expansion of the cooling roller is much smaller than the diameter, the dimensional change of the cooling roller is negligible. The geometry of the cooling roller is assumed as a circular ring.

2) When liquid flows in cooling structure, it comes to no-penetration and no-slip boundary condition.

3) Liquid inlet rate \(Q\) and temperature \(T\) are constant.

4) The viscosity of the cooling water is constant considering that the temperature difference is negligible from the inlet pipe to the outlet pipe.
5) Radiative heat transfer is neglected in this simulation. The above assumptions are not only applied for the construction of calculation models of two types of classical structures [12], but also for the new cooling structure. All the three models have steady-state, incompressible, turbulence flow problem with the cooling wall. The velocity is coupled accurately in solving the Navier–Stokes’s equation, which is called the momentum equation. The temperature field can be calculated with the energy equation. The results are checked by the continuity equation. The basic governing equations in the simulation are presented as following:

a) Continuity equation

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0
\]  \hspace{1cm} (1)

where \(\rho\) is the density of the cooling water and independent of time based on the assumption that the cooling water is incompressible and Newtonian liquid, hence

\[
\frac{\partial \rho}{\partial t} = 0
\]  \hspace{1cm} (2)

Combining equation (1) with equation (2)

\[
\frac{\partial (\rho u_i)}{\partial x_i} = 0
\]  \hspace{1cm} (3)

where \(\rho\) is constant, hence

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  \hspace{1cm} (4)

b) Momentum equation

\[
\frac{\rho D u_i}{D t} = \rho \dot{f}_i - \frac{\partial P}{\partial x_i} + \mu \frac{\partial^2 u_i}{\partial x_j \cdot \partial x_j}
\]  \hspace{1cm} (5)

c) Energy equation:

\[
\rho \left( \frac{\partial T}{\partial t} + u_i C_p \frac{\partial T}{\partial x_i} \right) = \frac{\partial}{\partial x_i} \left( \lambda \frac{\partial T}{\partial x_i} \right)
\]  \hspace{1cm} (6)

where \(\rho\) is the density of the cooling water, \(u_i\) the velocity, \(P\) the pressure, \(T\) the temperature, \(\mu\) the viscosity, \(C_p\) the specific heat, \(f_i\) the force per unit volume of fluid, \(\lambda\) the thermal conductivity. Subscripts \(i, j\) are the directions, \(i, j = 1, 2, 3\), represents \(x, y, z\) directions, respectively. \(z\) is width direction [13].

2.2. Convective heat transfer coefficient simulation

Based on Newton’s law of cooling, the theoretical interfacial heat transfer coefficient obtained by a coupled the water/roller surface.

\[
h = \frac{q}{T_i - T_w}
\]  \hspace{1cm} (7)

where \(q\) is the heat flux between cooling roller and water, and \(T_i\) and \(T_w\) are the temperatures of cooling roller inner wall and water, respectively.

According to Fourier’s law of heat conduction,

\[
q = -\lambda \frac{\partial T}{\partial x_i}
\]  \hspace{1cm} (8)

where \(\lambda\) is the coefficient of thermal conductivity [14].

2.3. Stress simulation

According to the elasticity theory, the relationship between thermal strain and temperature is as follows [15]:

\[
\varepsilon_T = \alpha (T - T_0)
\]  \hspace{1cm} (9)
strain. In other words, the total strain can be expressed as:

\[ \varepsilon = \varepsilon_M + \varepsilon_T \]  

where \( \varepsilon_M \) is thermal strain of the cooling roller, \( \alpha \) is the linear coefficient of thermal expansion of the cooling roller.

During the PFC process, the thermal strain of copper roller is produced by heat flux. At the same time, the mechanical strain will be produced due to interference when the copper roller is assembled. In line with Duhamel-Neumann theory, the thermal stress and strain can be superimposed with mechanical stress and strain. In other words, the total strain can be expressed as:

\[ \varepsilon = \varepsilon_M + \varepsilon_T \]  

where \( \varepsilon_M \) is mechanical strain of the cooling roller.

Therefore, the relationship between the total stress and the strain of the object is as follows:

\[ \sigma = D(\varepsilon_M + \varepsilon_T) \]  

where \( D \) is the elastic matrix of the cooling roller.

### 2.4. Computational domain, mesh quality and boundary conditions

As explained in our previous work, periodic distribution of flow and convective heat transfer coefficient are both very important. Based on our research, periodicity, consistency and symmetry are the key factors to design a good cooling model [12]. Thus, we divide the whole cooling roller into several parts with the same size. Each part has the same cooling structure to make sure that water flows periodically in the roller. We name the new structure as 'sector structure'. In sector structure, we change the conventional channels along the width direction into the longitudinal water channels along the circumference direction.

The size parameters of the cooling rollers are listed in Table 1. Compared with WCS and WGS structures, the outer diameter of sector structure is the same because the new structure is a renovation based on the classical cooling structures in order to meet the requirements of equipment upgrading. The width of new roller is much narrower, only 40% of the classical roller. But the maximum ribbon width of sector structure is almost twice as the classical one. The 3D mathematic model is established. The whole flow space in the spindle including cooling roller and pipes is selected as the computational domain for flow field and temperature field. The cooling roller is selected as the computational domain for stress field.

It is inaccurate to calculate the heat conduction with the one-dimensional or two-dimensional simplified methods. The reliability and accuracy of two-dimensional model have been questioned because the model is calculated without considering the distribution in the width direction, while heat transfer in PFC is a complicated process [16]. Three-dimensional model can utmost close to engineering practice when the model parameters are fixed.

When flows are calculated for regular geometries in CFD, block-structured grids can easily achieve the boundary fitting of the region and approach the actual model [17]. Hexahedral element meshes can increase the accuracy of simulation result [18]. Therefore, all hexahedral element meshes with block-structured grids are provided in three-dimensional model as the best algorithm to calculate the temperature and stress fields.

Spatial and temporal refinement of the solution are limited by hardware and calculation time restrictions, while the adequate description of the physical process depends on the current knowledge and the availability of mathematical models [19]. The finer meshes facilitate a better precision. Meanwhile, they could increase computation load and worsen numerical stability [20]. Dense uniform and well-distributed meshes with size of 0.5 mm–2 mm are chosen and the total number of meshes are 19,194,909, 12,047,960 and 7,484,211 in WCS, WGS and sector structure with software ANSYS, respectively.

The cooling areas consist of five parts: main spindle, primary inlet pipe, water ring, secondary inlet pipe and copper roller. Physical model of heat transfer in WGS and WCS is shown in figure 2(a). Different part is named with different color to distinguish the flow and heat exchange capacity, and to track the flow properties and cooling properties during post-processing. Part of the cooling structures discretized with meshes and grid refining magnifications for WGS and WCS structures are shown in figures 2(b) and (c) by software ANSYS.

Jacobian ratio is one of the ways to evaluate the meshes quality. In the range of 0–1.0, the larger the Jacobian value of an element, the better is its quality [21]. Jacobian ratios of the three models are shown in figure 3. In our

| Size                  | WCS structure | WGS structure | Sector structure |
|-----------------------|---------------|---------------|------------------|
| Roller outer diameter (mm) | 990           | 990           | 990              |
| Roller inner diameter (mm) | 940           | 940           | 940              |
| Roller width (mm)       | 250           | 250           | 100              |
| Roller thickness (mm)   | 25            | 25            | 25               |
| Maximum ribbon width (mm)| 35            | 35            | 65               |

Table 1. Size parameters of cooling roller.
simulation, minimum Jacobian ratio is larger than 0.4, and the elements with Jacobian values equal to 1.0 are 77.54%, 60.79% and 81.23%, respectively. Therefore, the mesh in three models satisfies the mesh quality permitted by finite element analysis.

Three-dimensional time-independent simulation of all-hexahedral element mesh in structured-grids with high Jacobian metric has been proved as an accurate and feasible method to calculate fluid flow field and heat transfer in cooling roller during the planar flow casting process [12]. Hence, the simulation of sector structure for temperature and stress field with the same algorithm can reflect actual conditions and are used in our work to judge structural rationality.

With the above set-up, it takes about 10h to run a typical case by using the current computer with the Intel (R) Core(TM) i7 CPU (four cores). With the use of proper mesh size, it is possible to obtain a convergent solution within an economical number of 700 iteration steps in ANSYS.

The boundary condition in equation (5) is that the velocity of the main inlet pipe in the simulation is 5.5 m s⁻¹. The velocity of water/wheel interface deducing with no-slip boundary condition is zero. The boundary condition on the pressure of the main outlet pipe reduces to P_{out} = 0, since pressure difference is the driving force to make water flow, thus the calculation can be simplified. The physical properties of air and copper wheel are taken from the in-built material database of ANSYS.

In order to eliminate the influence of puddle thermal properties on the cooling roller, the same melted material Fe₈₀Si₉B₁₁ alloy is used in our simulation for the three structures.
3. Numerical results and discussion

In order to verify the rationality of the sector structure, the flow, temperature and stress field of the cooling structure are simulated. The data of velocity distribution, convective heat transfer coefficient distribution and thermal expansion distribution are obtained.

3.1. Flow velocity distribution

The flow velocity distributions of six channels in sector structure on the sections \( y = 40 \text{ mm} \), \( y = 80 \text{ mm} \) and \( y = 120 \text{ mm} \) (S1, S2 and S3, respectively) in the width direction during the PFC process are demonstrated in figure 4(a). The velocity in the cooling zone under the cooling roller is higher than that in inlet. The velocity can reach up to 19.63 m s\(^{-1}\) at some position, nearly four times of inlet velocity due to its special cooling structure, compared with 3.37 m s\(^{-1}\) and 6.22 m s\(^{-1}\) in WCS and WGS. The cooling capacity of sector structure is higher than that in WCS and WGS because high flow velocity will cause high cooling capacity.

The velocity distributions of WCS and WGS are shown in figures 4(b) and (c), respectively. The velocity is changed from –3.60 m s\(^{-1}\) to 2.68 m s\(^{-1}\) in the water channels in WCS and from 0.56 m s\(^{-1}\) to 2.84 m s\(^{-1}\) in WGS in width direction. In WCS, there are negative values of velocities. Under this condition, the heat that originates from the cooling roller corresponding to the vortex cannot be taken away at the first time, thus increasing the vortex temperature. The cooling rate at this position will be lowered due to the reduced difference in temperature between cooling roller and water. Compared with flow velocities in WCS and WGS, the flow velocity in sector structure is well-distributed along width direction. According to our findings, it is clear that at the same section each channel displayed the same distribution. It means the flow velocity in the width direction is uniform. The uneven-rate of liquid velocity is only 1.47% in sector structure along width direction, considerably lower than those in WCS and WGS structure by 355% and 105%, respectively.

The distribution of flow velocity affects the distribution of convective heat transfer coefficient in the crystallizer. Irregular and uneven distribution of flow velocity will cause chaos of the cooling rate. Therefore, a good crystallizer should have uniform velocity distribution in width direction.

3.2. Convective heat transfer coefficient distribution

Convective heat transfer coefficient ‘\( h \)’ is the proportionality constant between the heat flux and the temperature difference. The convective heat transfer coefficient was usually assumed as an average value when heat transfer during PFC process was concerned \([1, 2, 9, 11, 22]\). However, it disagrees with the facts and is not reliable or accurate. Heat flux is dramatically variable in both width and circumferential direction in the cooling roller \([6]\).
Figure 5 demonstrates the distribution of convective heat transfer coefficient in three cooling structures, respectively. Figure 5(a) shows the convective heat transfer coefficient curve of the central line of the six channels from $x = 10$ mm to $x = 120$ mm in sector structure. The convective heat transfer coefficient is dramatically variable along the length direction of the channel. It rises rapidly, reaches the maximum value about $3.25 \times 10^4$ W m$^{-2}$K$^{-1}$ at the position $x = 25$ mm. Then it decreases sharply. Finally, it decreases slowly after $x = 75$ mm. Figure 5(b) shows the cloud picture of the distribution in sector structure corresponding the six channels in figure 4(a). Each channel almost has the same trend of convective heat transfer coefficient distribution in the length direction, which means that the sector structure has the completely consistent distribution of convective heat transfer coefficient in width direction.

Figures 5(c) and (d) show the distribution of convective heat transfer coefficient and contours at the roller/water interface of WCS roller during the PFC process. In figure 5(c), Line 1 to Line 7 correspond to the seven channels of the WCS roller, respectively. The x coordinate of the diagram shows the longitudinal direction of the channel, each line is in a prominent fluctuated status from $1 \times 10^5$ W m$^{-2}$K$^{-1}$ to $4 \times 10^4$ W m$^{-2}$K$^{-1}$. We
cannot find any clear trend in the diagram. Hence it is considered that the convective heat transfer coefficient is randomly distributed. Figures 5(e) and (f) show the convective heat transfer coefficient contours and distributions of Line 1 to Line 7 in WGS roller as at the same positions of WCS roller for the propose to compare with each other. It is found that the convective heat transfer coefficient decreases slightly at the channel intake, then rises rapidly to a maximum value at $z = -80$ mm with symmetrically distribution to $z = -67$ mm. It drops slowly down to $z = 50$ mm, and fluctuates again with modestly up and down near the outlet. Each line has a similar trend, meaning that the distribution in the entire inner wall of the wheel is same due to the periodic velocity distribution in the circumferential direction.

The uneven-rate of convective heat transfer coefficients in sector structure is just 1.33%, which is considerably lower than that in WCS and WGS structure by 141% and 169%, respectively. In summary, convective heat transfer coefficient of sector structure has the most uniform distribution in three models. Convective heat transfer coefficient represents the heat transfer capacity between fluid and solid surface. Therefore, the crystallizer with sector structure has the best performance of thermal conductivity.
3.3. Stress and thermal expansion distribution

The roller’s thermal expansion consists of two parts: expansion by interference and thermal expansion by heat flux from puddle. We analyze half of the roller width, from \( z = 0 \) mm to \( z = 50 \) mm, because the crystallizer has a symmetrical structure. Figure 6(a) is the roller’s expansion in sector structure caused by interference. Figure 6(b) shows the total expansion by interference and heat flux of the same position because the thermal expansion cloud picture cannot be displayed in ANSYS directly. The distribution of the thermal expansion is shown in figure 6(c). It can be observed the thermal expansion has a maximum value 0.0301 mm in position \( z = 0 \) mm and minimum value 0.021 mm in position \( z = 50 \) mm. It is also found the thermal expansion is 0.0259 mm at the position of ribbon edge so that the thermal expansion difference \( \Delta G \) is 0.0042 mm in the slit region.

The ribbon thickness changes sensitively with nozzle-wheel gap [23]. The ribbon thickness is proportional to the nozzle-wheel gap. When thermal expansion happens, the nozzle-wheel gap becomes smaller so that the ribbon thickness gets thinner. If thermal expansion has difference in the slit region, it will cause an uneven thickness for the ribbon. Therefore, the value of \( \Delta G \) is the lower the better.

Li et al [6] found that \( \Delta G \) is approximately 0.02 mm in classical structure which is used in industrial setting by simulation and experiments. It is about five times of \( \Delta G \) in sector structure that we calculated. Li reported the maximum thermal expansion is 0.217 mm in a classical crystallizer when producing ribbon width is 35 mm,

![Figure 6. Roller’s expansion distribution in sector structure. (a) by interference; (b) by interference and heat flux; (c) thermal expansion in width direction.](image-url)
about seven times of thermal expansion in sector structure that we calculated. It is unnecessary to make the slit upward bend to keep the gap constant in width direction since $\Delta G$ is almost negligible in sector structure. In contrast, Li suggested to make slit upward bend to keep the gap between nozzle and cooling roller constant. It is a great improvement compared with classical structure because it saves one step during PFC process so that the manufacturing is simplified.

3.4. The utilization rate of roller surface

The uniform distribution or symmetric distribution of convective heat transfer coefficient $'h'$ are the most suitable locations to produce ribbon [12]. There are only limited uniform area and symmetric area in roller width direction of convective heat transfer coefficient distribution in conventional structures. In this situation, wide ribbon cannot be produced in conventional systems. However, convective heat transfer coefficient is uniform distributed in roller width direction with sector structure. Although the roller width in sector structure is lower than that of the other two, the ribbon width in sector structure is larger than for the conventional systems.

As found in the experiments we carried out, the max ribbon width that can be achieved is 45 mm in the central zone in the cooling roller with the width of 250 mm [12]. The utilization rate of roller surface is only 18%. It is very low in industrial production. In contrast, the utilization rate of roller surface in the new crystallizer we design reaches up to 65%, three times more than that of the classical one.

The high utilization rate of roller surface can increase productivity and cut the production cost. Hence the new sector structure shows a great prospect for industrial application. The application of wireless power transfer requests wider and thinner ribbon. Sector structure with high utilization is a reasonable choice to produce this kind of ribbon in the future.

4. Conclusion

We designed a new crystallizer structure based on the periodicity, consistency and symmetry in distributions of flow velocity and convective heat transfer coefficients. Three-dimensional, steady, time-independent simulations of temperature and stress fields in the cooling roller were realized. Results show a low uneven-rate of liquid velocity of 1.47%, as well as a low uneven-rate of convective heat transfer coefficients of 1.33% in the sector structure along the width direction. The thermal expansion difference of the new crystallizer structure is 0.0042 mm. The utilization rate of the roller surface in the new design is 65%, far greater than the utilization rate of 18% in previous designs WCS and WGS. The simulation result shows that the crystallizer with the new sector structure design is good for mass production.

Acknowledgments

This work was financially supported by National Natural Science Foundation of China (Grant Nos. 51771020 and 51801230). Key Project of the Equipment Pre-Research Field Fund of China (6140922010302).

Data availability statement

All data that support the findings of this study are included within the article (and any supplementary files).

Declaration of interests

On behalf of all authors, the corresponding author states that there is no conflict of interest.

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References

[1] Mattson J, Theisen E and Steen P 2018 Rapid solidification forming of glassy and crystalline ribbons by planar flow casting Chem. Eng. Sci. 192 1198–208
[2] Altieri A L and Steen P H 2014 Adhesion upon solidification and detachment in the melt spinning of metals Metallurgical and Materials Transactions B 45B 2262–8
[3] Enomoto Y, Tokoi H, Imagawa T, Suzuki T, Obata T and Souma K 2015 Development of IE5-class efficiency standard amorphous motor Journal of the Japan Society of Applied Electromagnetics and Mechanic 24 258–63
[4] Hasegawa B 2004 Applications of amorphous magnetic alloys Mater. Sci. Eng. A 375–377 90–7
[5] Li F C, Liu T, Zhang J Y, Shuang S, Wang Q, Wang A D, Wang J G and Yang Y 2019 Amorphous–nanocrystalline alloys: fabrication, properties, and applications Materials Today Advance 41–20
[6] Li Y K, Yang Y and He C Y 2018 Temperature and thermal expansion analysis of the cooling roller based on the variable heat flux boundary condition JOM 70 855–60
[7] Bussemann M, Mostaghimi J, Kirk D W and Graydon J W 2002 A numerical study of steady flow and temperature field within a melting spinning puddle Int. J. Heat Mass Transfer 45 3997–4010
[8] Byrne C J, Weinstein S J and Steen P H 2006 Capillary stability limits for liquid metal in melt spinning Chem. Eng. Sci. 61 8004–9
[9] Bichi A B, Smith W R and Wissink J G 2008 Solidification and downstream meniscus prediction in the planar–flow spin casting process Chem. Eng. Sci. 63 685–95
[10] Liu H P, Chen W Z, Qiu S T and Liu G D 2009 Numerical simulation of initial development of fluid flow and heat transfer in planar flow casting process Metallurgical and Materials Transactions B 40 411–29
[11] Sowjanya M and Kishen Kumar Reddy T 2014 Cooling wheel features and amorphous ribbon formation during planar flow melt spinning process J. Mater. Process. Technol. 214 1361–70
[12] Chen L, Li Y L, Shen N N and Hui X D 3D steady time–independent simulation on the fluid flow field and heat transfer in cooling roller during the planar flow casting process Rare Met. Mater. Eng. In press
[13] Su Y G, Chen F L, Wu C Y and Chang M H 2017 Simulation for the effect of wetting conditions of melt puddle on the Fe–Si–B ribbon alloy in the planar–flow melt–spinning process JSME Int. 57 100–6
[14] Wang G X and Mattheis E F 2002 Mathematical simulation of melt flow, heat transfer and non-equilibrium solidification in planar flow casting Model. Simul. Mater. Sci. Eng. 10 35–55
[15] Zhang S F, Yin J, Liu Y, Liu N, Sha Z H, Wang Y N and Rolfe B 2018 Thermal–structural coupling analysis of brake friction pair based on the displacement gradient circulation method Advances in Mech. Eng. 10 1–13
[16] Li Y K, Yang Y and He C Y 2018 Three-dimensional transient temperature analysis of cooling roller for preparing amorphous ribbon J. Non-Cryst. Solids 481 276–81
[17] Ahusborde E and Glockner S 2010 An implicit method for the Navier–Stokes equations on overlapping block-structured grids Int. J. Numer. Methods Fluids 62 784–801
[18] Lu S, Zhao G Q and Ma X W 2012 Quality improvement methods for hexahedral element meshes adaptively generated using grid-based algorithm Int. J. Numer. Methods Eng. 89 726–61
[19] Grahn A, Pescador E D, Klimes S, Schäfer F and Höhne T 2021 Modelling of complex boron dilution transients in PWRs—Validation of CFD simulation with ANSYS CFX against the ROCOM E2.3 experiment Nucl. Eng. Des. 372 110938
[20] You Y H, Wang S, Lu W, Chen Y Y, Gross U and CFD A 2021 Model of frost formation based on dynamic meshes technique via secondary development of ANSYS fluent Int. J. Heat Fluid Flow 89 108807
[21] Knupp P M 2003 A method for hexahedral mesh shape optimization Int. J. Numer. Methods Eng. 58 319–32
[22] Su Y G, Chen F L, Chang C M, Wu C Y, Chang M H and Chung CA 2014 Tuning the planar–flow melt-spinning process subject to operability condition JOM 66 1277–86
[23] Li D R, Zhuang J H, Liu T C, Lu Z C and Zhou S X 2011 The pressure loss and ribbon thickness prediction in gap controlled planar–flow casting process J. Mater. Process. Technol. 211 1764–7