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ARTICLE

Numerical Simulation of Gear Heat Distribution in Meshing Process Based on Thermal-structural Coupling

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ABSTRACT

The thermal balance state of high-speed and heavy-load gear transmission system has an important influence on the performance and failure of gear transmission and the design of gear lubrication system. Excessive surface temperature of gear teeth is the main cause of gluing failure of gear contact surface. To investigate the gear heat distribution in meshing process and discuss the effect of thermal conduction on heat distribution,a finite element model of spur gear is presented in the paper which can represent general involute spur gears. And a simulation approach is use to calculate gear heat distribution in meshing process. By comparing with theoretical calculation, the correctness of the simulation method is verified, and the heat distribution of spur gear under the condition of heat conduction is further analyzed. The difference between the calculation results with heat conduction and without heat conduction is compared. The research has certain reference significance for dry gear hobbing and the same type of thermal-structural coupling analysis.

1. Introduction

With the development of machinery industry, the requirement of gear transmission is increasingly raised, and gear transmission is developing towards high speed and heavy load [1]. The friction of gear engagement will produce a large amount of heat in high power transmission with high speed or low speed and heavy load. Thus thermal deformation and thermal stress will dominate the gear stress distribution and further cause failure or pitting gear scuffing [2].Gear surface temperature is an important factor affecting the gluing of gear surface and has an important influence on the performance and failure of gear transmission [3]. Therefore, it is crucial to study the temperature field of gear surface and establish an accurate gear analysis model for the thermal design and verification of high-speed and heavy-duty gears.

Many scholars at home and abroad use different analytical methods and means to study the temperature of the contact surface of gear teeth. At present, the most influential international standards for calculating the transmission capacity of gears, such as AGMA American Gear Standard [4], have established the calculation criteria for the anti-gluing ability of the tooth surface according to the flash temperature theory of tooth surface proposed by H. Blok in 1973 [5] and the theory of partial temperature on the tooth surface proposed by H. Winter in 1975 [6]. The theoretical analysis method gives the estimated

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value of gear tooth temperature under hypothetical conditions [7], but it is not suitable for the analysis of complex practical systems. The experimental measurement technology provides the temperature value of the discrete points of the gear contact surface under the actual operating conditions[8], but the measurement technology is complex and requires special testing equipment and measuring equipment. The numerical simulation, which combines finite element analysis with theoretical calculation, can accurately solve the temperature field and heat flux distribution by establishing accurate meshing model and thermal boundary conditions, and can provide more effective means for gear meshing [9].

In order to accurately obtain the instantaneous heat of gears in meshing process, a Thermal-Solid coupling transient finite element model is established in this paper. The transient heat and heat flux of gears in meshing process are calculated by using finite element software *ABAQUS*. By comparing with theoretical calculation, the correctness of the simulation method is verified.

And the analysis theoretical method generally does not consider the heat conduction of gears when calculating the meshing heat flux density of gears,which is inconsistent with the actual situation. To discuss the effect of thermal conduction on heat distribution,subsequently the thermal conduction is added to numerical simulation,and their different behaviour will be explained in more detail later.

2. Determination of Thermal Boundary Condition

2.1 Thermal Conductivity and Heat Transfer Coefficient

The thermal conductivity is expressed as the modulus of the heat flux density passing through a unit temperature gradient, which reflects the thermal conductivity of an object. It is an important thermal physical parameter of a substance. In engineering calculation, the values of thermal conductivity of various substances applied to practical analysis are obtained by experimental measurements. Referring to the relevant literature, the thermal conductivity of carbon steel can be calculated by the following empirical formulas:

$$\lambda = 70 - 10.1C - 16.7Mn - 33.7Si \tag{1}$$

Note: In above formula, C is carbon content in steel; Mn denotes manganese content in steel; Si represents silicon content in steel.

Convective heat transfer refers to the heat transfer in the condition where fluid and solid contact directly. In calculating the temperature field of gears, the heat transfer coefficient of gears mainly depends on the operating conditions and lubrication modes. The convective heat transfer coefficients of different surfaces of gears are different, which are mainly divided into two parts: the heat transfer coefficient of tooth surface and the heat transfer coefficient of end surface. Under the condition of no lubrication, the temperature of the end surface is consistent with that of the environment, mainly considering the heat exchange between the tooth surface and the environment. For the determination of heat transfer coefficient of gear tooth surface, many scholars at home and abroad have made theoretical analysis and research on it, and summarized some empirical formulas. The computational formula of forced convection heat transfer coefficient between tooth surface and fluid is as follows[10]:

$$\alpha_i = 0.228 R_e^{0.731} p_r^{0.333} \lambda / L \tag{2}$$

Note: In the formula, R_e is the Reynolds number; P_r devotes the Prandtl number of fluid; λ presents the Thermal conductivity of fluid; L refers to the setting size of gear.

2.2 Friction Coefficient

The friction coefficient of the gear tooth surface varies with the change of speed and contact load, and is affected by the meshing position of the gear, the roughness of the tooth surface, the dynamic viscosity of lubricating oil and the average temperature of the gear [11]. For any meshing position C, the friction coefficient can be expressed as [12]:

$$\mu = 0.002\left[F_t / \left(b \times 0.001\right)\right]^{0.2} \times \left[\frac{2}{\cos\alpha\left(V_1 + V_2\right)\rho_e \times 0.001}\right]^{0.2} \eta^{-0.05} X_r \tag{3}$$

Note: In the formula, F_t is the tangential load of gear meshing point; b denotes the tooth width of gear meshing point; X_r represents the tooth surface roughness factor; η refers to the dynamic viscosity of lubrication oil.

2.3 Material Property

The material used in this paper is $12CrNi4A$. Gear parameters are shown in Tables 1 and 2. The speed of driving gear is 4200 r/min, and the resistance moment of driven gear is 500 $N{\cdot}m$.

Table 1. Geometric parameters of gear

Tooth Number Z_1	Tooth Number Z_2	modulus $m/$m	Center Distance $a/$mm	Pressure Angle $\alpha/$m(◦)
33	34	4	96	20

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Table 2. Material parameters of gear

Poisson Ratio μ	Density / (t/mm^3)	Thermal Conductivity λ_1,λ_2/(mW/(mm·K))	Specific Heat Capacity c/ (mJ/(t·K))	Thermal Expand Coefficient/ a_T/(1/K)
0.3	7.85×10^{-9}	30.98	0.5×10^{-9}	1.3×10^{-5}

3. Theoretical Calculation of Gear Friction

This paper uses the finite element software *ABAQUS* to simulate the gear friction heat. In order to compare the correctness of the results, the theoretical formulas for calculating the friction heat generation in gear meshing are given below.

3.1 Calculation of Gear Sliding Speed

Tangential velocity of gear contact point is affected by angular velocity and meshing point position. As shown in Figure 1, the relative sliding speed of each point on the meshing line is different. Tangential velocity of driving and driven gears can be expressed as:

$$v_{t1} = v_{k1}\alpha_{k1} = \omega_1 r_{k1}\sin\alpha_{k1} = \omega_1\overline{N_1K}$$
$$v_{t2} = v_{k2}\alpha_{k2} = \omega_2 r_{k2}\sin\alpha_{k2} = \omega_2\overline{N_2K}$$

Then the relative motion velocity of gears:

$$v_{21} = v_{t2} - v_{t1} = \omega_2\overline{N_2K} - \omega_1\overline{N_1K} = \left(\omega_2+\omega_1\right)\overline{CK}$$

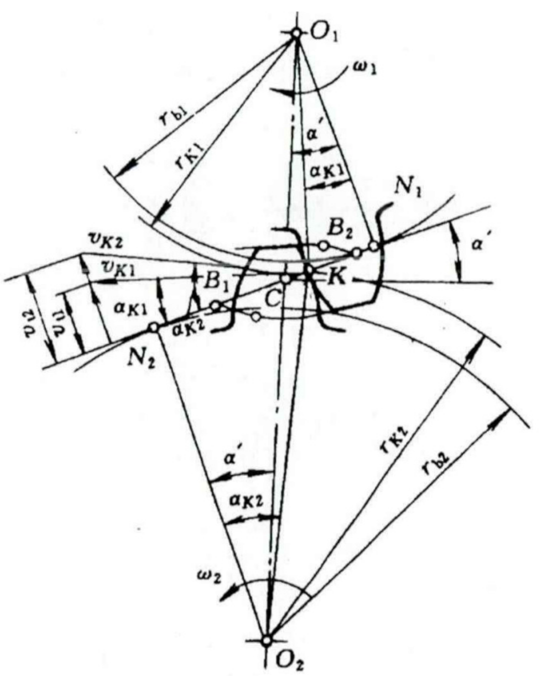

Figure 1. Gear meshing relative motion model

According to the calculation, the tangential velocity of the meshing point are obtained shown as Figure 2

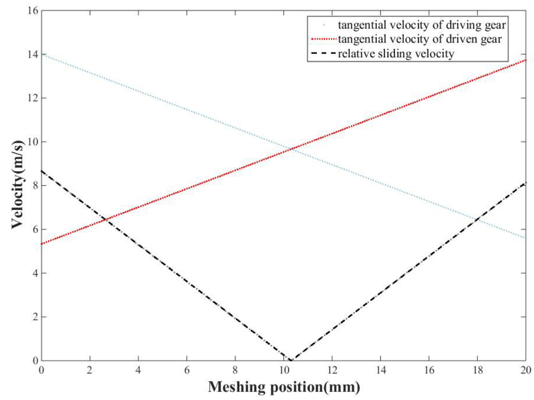

Figure 2. Relative motion velocity of gears

3.2 Heat Flux Distribution

The friction heat per unit time per unit area produced by the gear meshing (i.e. density of heat flow rate) is as follows:

$$q = \frac{1}{J}p_e\mu v \tag{4}$$

In the formula, q *is the total friction heat flux density;* J *denotes thermal work equivalent;* p_e *presents average contact pressure of the tooth surface;* μ *refers to surface friction coefficient;* v *is relative sliding speed of the tooth surface.*

Assuming that the heat distribution coefficient is ψ ,the friction heat of big gear and small gear is respectively:

$$\begin{cases} q_1 = \psi q \\ q_2 = \left(1-\psi\right)q \end{cases} \tag{5}$$

Thermal distribution coefficient ψ is expressed as [13]:

$$\psi = \frac{\sqrt{\lambda_1\rho_1c_1v_{t1}}}{\sqrt{\lambda_1\rho_1c_1v_{t1}}+\sqrt{\lambda_2\rho_2c_2v_{t2}}} \tag{6}$$

In the formula, λ_1、λ_2 *are the thermal conductivity of the material for two transmission gears,* ρ_1、ρ_2 *denotes the density of the material for two transmission gears,* c_1、c_2 *present the specific heat capacity for two transmission gears,* v_{t1}、v_{t2} *refer to the tangential velocity at the meshing point for two transmission gears.*

Thermal distribution coefficient ψ shows the heat flux distribution rules of master-slave gears at each meshing point shown as Figure 3. The gear with large tangential velocity at meshing points have larger distribution coefficient and can be allocated more heat. And at the picth line where their tangential velocity are equal,they have the same distribution coefficient. To compare with numerical

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simulation conclusion,the friction heat flux of driving gear by theoretical calculation method is displayed on Figure 4.

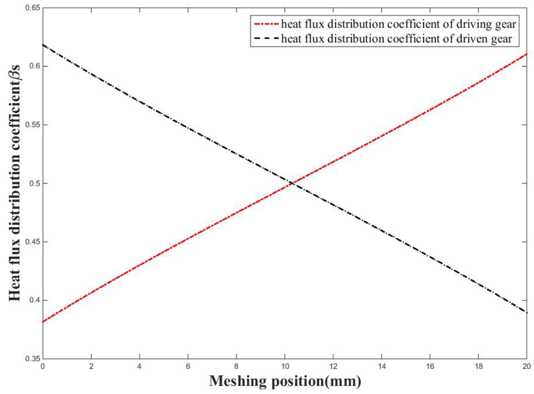

Figure 3. Heat flux distribution coefficient(ψ) of gears

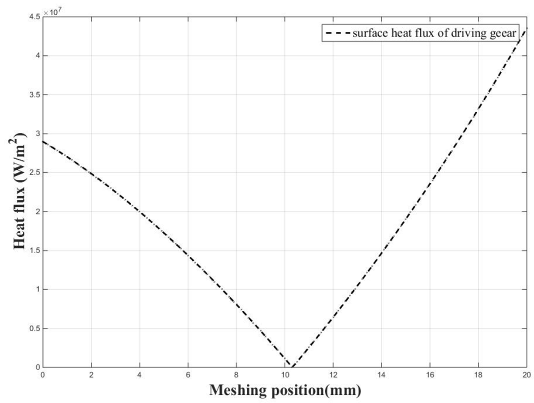

Figure 4. Friction heat flux of driving gear

The above analysis shows the following results: (1) The heat flux density at the pitch line is zero, because the relative sliding velocity at the pitch line is zero. (2) The heat flux of the driving gear is higher than that of the driven gear. The main reason is that the driving gear has fewer teeth and higher speed, and produces more heat per unit time. (3) The heat flux density at the root of the driving gear and driven gear exceeds that at the top of gears. (4) Heat production and heat distribution formulas do not involve heat conduction within gears and heat exchange between gears and environment.

4. Finite Element Simulation

4.1 Basic Equation

Because of frictional heating and plastic deformation,the gear temperature increases during in rotating. The thermal conduction equation with internal heating will be employed as follows:

$$\frac{\partial}{\partial x}\left(k_x\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_y\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_z\frac{\partial T}{\partial z}\right)+\dot{q}=\rho c\frac{\partial T}{\partial t}$$

where $\dot{q}$ is the internal heat intensity, ρ is the density, c the specific heat,and k_x, k_y, k_z are the thermal conductivity in the x , y and z directions,respectively.

If $k_x = k_y = k_z = k$,and the k is constant,then

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}+\frac{\dot{q}}{k}=\frac{1}{\alpha}\frac{\partial T}{\partial t}$$

Where $\alpha = k/\rho c$ is the heat conduction coefficient.

4.2 Pretreatment Stage

First of all, the standard spur gear geometrical model with 33 and 34 teeth numbers in UG (Unigraphics NX) is established and subsequently discretized in Hypermesh. The minimum size of the grid is $1mm$, and the number is 500 thousand. The grid element adapts "temperature-displacement coupling" thermal element C3D8RT: "An8-node thermally coupled brick,trilinear displacement and temperature,reduced integration".

Subsequently,material properties are assigned to model as shown in Table.2. Release only Y-direction rotation constraint on gears. The boundary conditions choose convection and heat transfer with air and setting room temperature $20^\circ C$. Meanwhile,setting "power, temperature-displacement, display" analysis steps,driving wheel speed $4200r/\min$, driven wheel resistance moment $500N\cdot m$. The finite element mesh model of gear meshing.

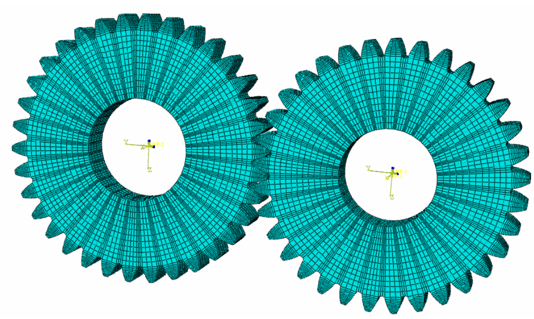

Figure 5. Finite element mesh model of gear meshing

Because the theoretical calculation method does not consider the influence of heat conduction and thermal expansion, in order to facilitate the comparison and ensure the correctness of simulation results, the gear meshing model simulation is divided into two parts: not considering heat conduction and considering heat conduction.

4.3 Simulation Results without Heat Conduction

The calculation results of meshing temperature field of spur gears without heat conduction are shown in Figure 6

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and Figure 7. Figure 6 and Figure 7 are the simulated temperature nephogram of the whole model when the calculation is completed. In order to study the temperature time history of the gear tooth surface, the temperature-time history curves of three points near the root, top and boundary are respectively extracted, as shown in Figure 8.

From Figure 6 Figure 7 and Figure 8, it can been seen that: (1) No heat is generated at the pitch line and the temperature remains constant at room temperature. (2) The temperature increases gradually along the two sides of the pitch line away from the pitch line and reaches the maximum at the root and the top of the tooth. (3) The temperature of the root of the tooth is higher than that of the top of the tooth, which is consistent with the trend of theoretical calculation. (4) Without heat conduction, the temperature could not be transmitted after the gear meshing is completed, and remains unchanged until entering the next meshing. Therefore, the temperature-time curve of the gear surface presents a "ladder" growth trend.

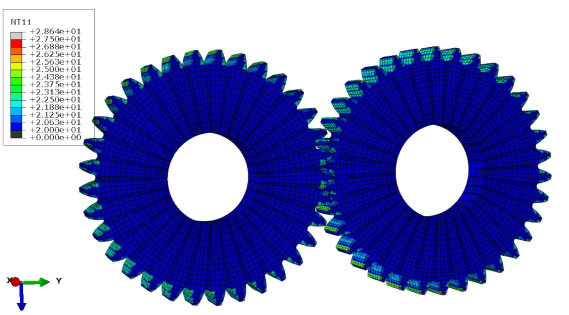

Figure 6. Temperature nephogram of friction heating

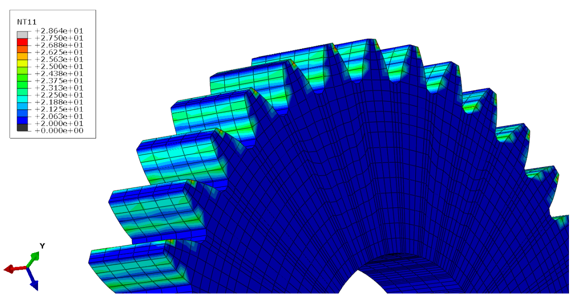

Figure 7. Temperature nephogram of drivinggear

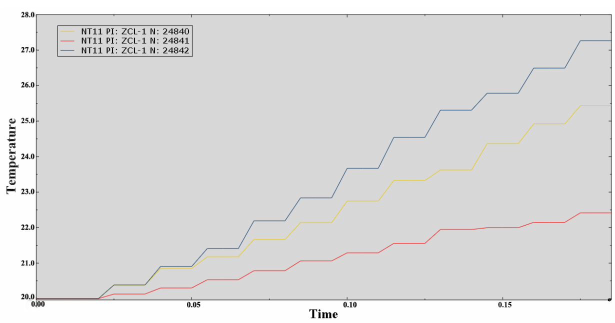

Figure 8. Temperature time history curve of meshing area

By further extracting the surface temperature-time diagram of a certain gear tooth of the driving wheel, and obtaining the surface heat flux of the driving gear through conversion, and then the finite element simulation results are compared with the theoretical calculation results, as shown in Figure 9.

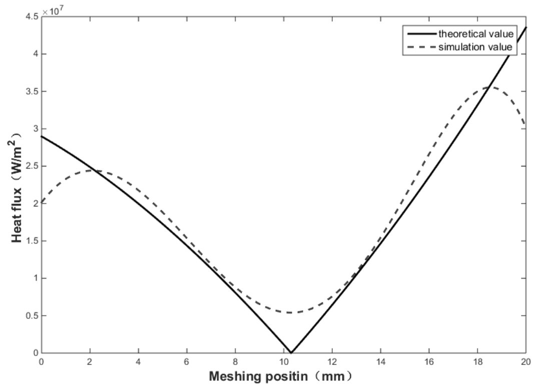

Figure 9. Heat flux diagram of driving gear tooth surface

The comparison of the curves shows that the simulation results are in agreement with the theoretical analysis. The heat flux reaches the maximum value at the root and top of the teeth, and the minimum one at the pitch line. The biggest difference lies in the root and top of the teeth. It is mainly because of the deviation while contacting, which makes the simulation value deviate from the calculation value slightly.

4.4 Simulation Results with Heat Conduction

In actual working conditions, heat conduction exists in the material, which has a certain influence on the temperature distribution of gears. Therefore based on the above simulation, heat conductivity is added to observe the change of tooth surface temperature and its trend with time.

Figure 10 and Figure 11 are transient temperature simulation nephogram of gears with heat conduction. Friction heat flux mainly distributes in meshing area. Similarly, selecting the elements near the top, root and pitch line as the example of analysis, the temperature time history curves of three points are shown in Figure 12.

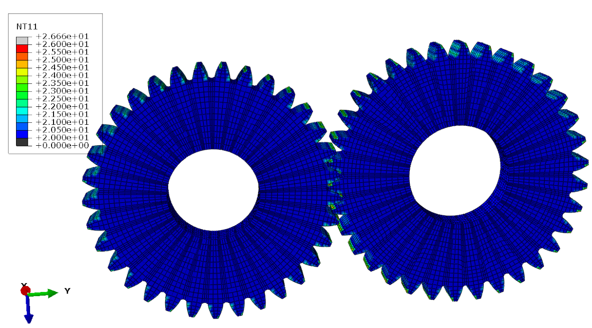

Figure10. Temperature nephogram of friction heating

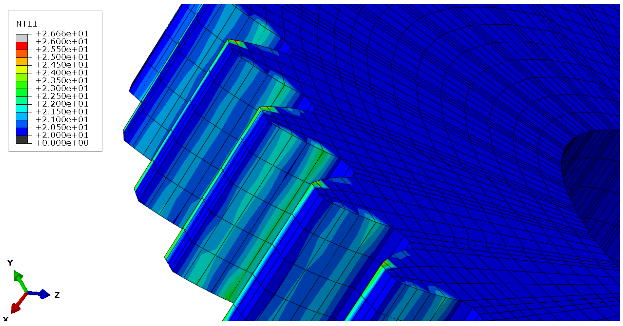

Figure 11. Temperature nephogram of driving gear

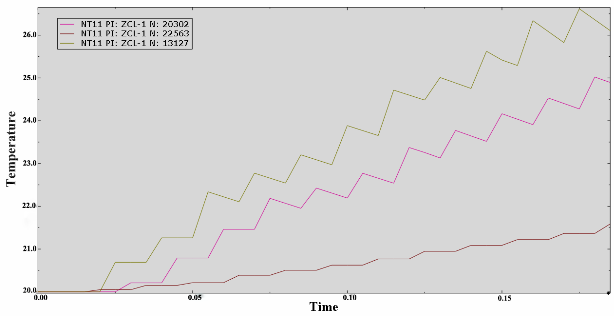

Figure 12. Heat flux diagram of driving gear tooth surface

Comparison of temperature distributions with and without heat conduction shows that: (1) During the same meshing period, the temperature variation of the model with thermal conductivity is smaller than that without thermal conductivity; (2) Because of the existence of heat conduction, the temperature of the model with thermal conductivity increases in the meshing area, and it decreases gradually in the non-meshing area, showing a "zigzag" growth trend as shown in Figure12, while the model without thermal conductivity presents a "ladder" trend in Figure 8; (3) Because of the existence of thermal conductivity, the heat transfer from the tooth surface to the inner and tooth width, and the temperature in the middle of tooth width is slightly higher than that at the edge of tooth.

5. Conclusion

While calculating the heat flux on the meshing surface of gears by analytic method, the influence of heat conduction and expansion is not taken into account, and the results are idealized. Numerical simulation can more truly simulate the situation and take all kinds of factors into consideration. Thus numerical simulation on gear heat flux distribution in meshing process based on thermal-structural coupling analysis technique is apparently expected and it is increasingly paid to attention.

Finite element analysis of gear meshing based on thermal-structural coupling in this paper shows that the temperature distribution of meshing tooth surface is basically symmetrical along gear pitch line. When considering heat conduction, the temperature conducts towards the tooth edge and internal structure, and the intermediate temperature of tooth structure is slightly higher than the temperature on the side surface of tooth.

Because the heat generated by each meshing is equal, but when the thermal conductivity is added, the heat generated by the meshing friction of the gear will be transferred to the internal direction, the direction of tooth width and the direction of tooth height. Therefore, the temperature in the meshing area will decrease. So the trend of the temperature-time curve with the thermal conductivity model increases first and then decreases, showing a "zigzag" shape, while the non-thermal conductivity model shows a "ladder" shape.

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