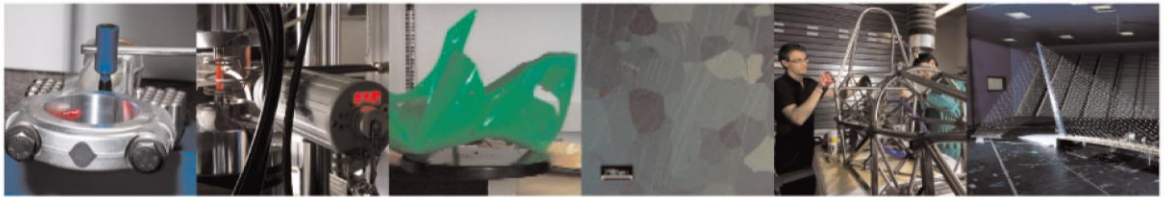




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Experimental and Inverse Numerical Analysis of the Thermal Behavior of Pantograph Collector Strips under Natural and Forced Convection

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Abstract: The thermal behavior of pantograph collector strips significantly influences their wear performance. However, most existing numerical studies rely on generic thermal parameters and empirical convection conditions, which limit their applicability to actual operating conditions. This paper aims to investigate the thermal behavior of the collector strip and to identify its unknown critical parameters experimentally and numerically. A transient two-dimensional thermal model based on the finite difference method is developed for the collector strip, and full-scale temperature rise tests are conducted under natural and forced convection conditions. By employing the Levenberg-Marquardt inverse method with an explicit sensitivity matrix on experimental data obtained under natural convection, the unknown thermal parameters of the collector strip are identified. The identified parameters and the thermal model are further validated by reproducing temperature profiles under distinct experimental conditions. Furthermore, forced convective heat transfer coefficients are identified at various airflow speeds. Those coefficients exhibit a power-law relationship with airflow speed, consistent with classical correlations reported in the literature. Comparisons with other thermocouple data demonstrate acceptable relative errors, validating the applicability of the identified coefficients.

Keywords: Pantograph-overhead catenary system; Collector strip; Thermal behavior; Inverse parameter identification; Forced convection.

1. Introduction

As one of the critical technologies and an integral part of the railway system, the pantograph-overhead catenary system (PCS), plays a vital role in ensuring stable electrical energy transmission to electric locomotives in the high-speed lines and the metro lines [1,2]. The key to achieving this functionality lies in maintaining good interaction performance of the PCS, which ensures continuous contact and high-quality current collection [3,4]. With the continuous increase in train speeds and

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the growing complexity of operating environments, PCS components face new maintenance challenges, such as the recently reported abnormal wear of the collector strips [5–7]. Abnormal wear deteriorates the current collection quality, thereby affecting the operating stability and safety of electric locomotives [8,9]. Therefore, a comprehensive investigation into the wear behavior of the PCS is needed to address those emerging challenges.

As a byproduct of electrical energy transmission, the contact temperature is regarded as a critical parameter influencing the wear behavior of the collector strip (CS) [10]. Experimental studies have revealed a positive correlation between the contact temperature and the wear rate of the strip, with an accelerating effect observed at higher temperatures [11,12]. A temperature gradient in the CS causes delamination wear and uneven wear [13]. Several heuristic wear models for the strip were developed from the perspective of energy dissipation [14–16], whose formulations are largely consistent with the heat generation. These studies indicate that the wear behavior of the CS is closely coupled with its thermal behaviors. Therefore, an investigation into the thermal behavior of the collector strip is crucial for advancing the understanding of its wear behavior.

Currently, numerical simulations serve as the primary approach for investigating the thermal behavior of the collector strip. A three-dimensional thermal model considering the rough contact surface of the contact pair was established using commercial software in Ref. [17]. Ref. [18] established a two-dimensional thermal model of the specimen collector strip solved by the finite difference method (FDM), which accounted for variations in the convective heat transfer coefficient. In Ref. [19], an actual-scale thermal model of the pantograph collector strip was developed by finite element method and compared with test results that considered additional airflow effects and high current levels. Moreover, the coupled thermal behavior of the CS under multi-physics coupling has been investigated in Refs. [20,21].

Although considerable progress has been achieved in thermal behavior analyses of the PCS, most of the existing studies are based on the solutions of forward problems, in which generic thermal parameters of the materials are prescribed as known inputs for simulations. However, accurate thermal parameters are critical to determining the reliability of numerical predictions. Due to confidentiality and material differences, these critical parameters are often unavailable or not explicitly provided, which reduces the applicability of numerical models to real conditions, especially in the thermal behavior study of newly developed collector strip materials. In addition, the treatment of forced convection boundary is often simplified. Many studies adopt constant assumed values or empirical correlations without direct experimental validation, which further limits the accuracy and credibility of the simulated thermal behavior. Therefore, identifying thermal parameters and boundary conditions from experimental data is of great significance for improving the physical reliability of numerical models. This process, referred to as the inverse problem in thermal analysis [22], has been widely applied to parameter identification in various engineering heat transfer problems, and has recently been applied to the thermal behavior analysis of the PCS. In particular, Ref. [23] reports an inverse problem solution for the unknown parameters in a contact

wire temperature rise model. Furthermore, the electrical resistance of the sliding contact in the PCS (referred to hereafter as electrical contact resistance, ECR) is another critical parameter in the analysis of thermal behavior, particularly for evaluating the heat flux dominated by Joule heat [24]. In this paper, the electrical contact resistance is also considered, and its mathematical relationship with operating parameters is established experimentally.

This study aims to identify the unknown thermal parameters and the forced convective heat transfer coefficients of the collector strip by combining temperature-rise experiments, a forward thermal numerical model, and inverse analysis. The following sections of the paper are organized as follows. In Section 2, the forward thermal model of the actual-scale collector strip and the calculation procedures employed for solving the inverse problem based on an explicit sensitivity matrix are presented. Section 3 details the experimental setup under both natural and forced convection, as well as the measurement of ECR. In Section 4, a mathematical expression relating the ECR to the operational parameters of the PCS is developed, and the experimental thermal behaviors of the CS under both natural and forced convection are further analysis. The unknown thermal parameters of the CS, as well as the average convective heat transfer coefficients under forced convection, are identified and validated based on the experimental data in Section 5. The conclusions are presented in Section 6.

2. Numerical methodology

2.1 Forward problem

Considering the actual PCS interaction, the analysis of the CS thermal behavior requires the determination of the temperature profile of the body under the effect of heat supply from a moving source. The CS is simplified to a two-dimensional planar case, with domains Φ_{cbs} and Φ_{alc} , defined in the zox plane as shown in Fig. 1, assigned to the carbon strip and to the aluminum carrier, respectively. In order to establish suitable boundary conditions for the problem, the external surface of the CS is partitioned into 5 regions: Γ_1 representing the region over which the contact with the CW, Γ_2 representing the remainder of the upper surface of the CS, the interface Γ_3 between the carbon strip and aluminum carrier, the side surface Γ_4 , which is divided in turn into parts Γ_4^{cbs} and Γ_4^{alc} for the two domains Φ_{cbs} and Φ_{alc} , and on the lower surface Γ_5 .

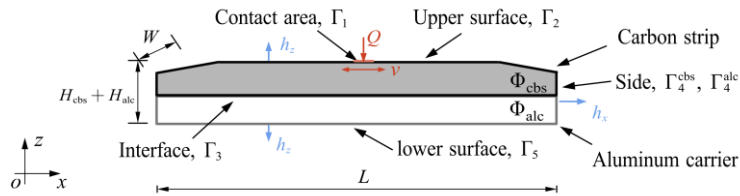


Fig. 1 Thermal model scheme of the collector strip

The transient thermal model of the CS is defined:

$$\begin{cases} \rho_{\text{cbs}} c_{\text{cbs}} \frac{\partial T_{\text{cbs}}}{\partial t} = \frac{\partial}{\partial x} \cdot \left(k_{\text{cbs}}^x \frac{\partial T_{\text{cbs}}}{\partial x} \right) + \frac{\partial}{\partial z} \cdot \left(k_{\text{cbs}}^z \frac{\partial T_{\text{cbs}}}{\partial z} \right) - q_{\text{cbs}}^{\text{loss}}, & \text{in } \Phi_{\text{cbs}} \\ \rho_{\text{alc}} c_{\text{alc}} \frac{\partial T_{\text{alc}}}{\partial t} = \nabla \cdot (k_{\text{alc}} \nabla T_{\text{alc}}) - q_{\text{alc}}^{\text{loss}}, & \text{in } \Phi_{\text{alc}} \end{cases} \quad (1)$$

where ρ , c and k are respectively the density, specific heat capacity, and thermal conductivity of the body, and the use of the subscripts *cbs* and *alc* denote the carbon strip and the aluminum carrier, respectively. The carbon strip material is considered as anisotropic, then the thermal conductivity is assumed to take different values along the x and z directions, so the corresponding coefficients are distinguished by the relevant superscript. The term $q_{\text{cbs}}^{\text{loss}}$ and $q_{\text{alc}}^{\text{loss}}$ represent the equivalent volumetric heat loss rate due to convective heat loss from the front and back surfaces of the carbon strip and the aluminum carrier, respectively, written as:

$$q_{\text{cbs}}^{\text{loss}} = \frac{2h_y}{W} (T_{\text{cbs}} - T_{\infty}), \quad q_{\text{alc}}^{\text{loss}} = \frac{2h_y}{W} (T_{\text{alc}} - T_{\infty}) \quad (2)$$

where h_y is the convective heat transfer coefficient on the front and back surface, and the coefficients for both surfaces are assumed to be identical according to the results reported in Ref. [25], W is the width of the CS, and T_{∞} is the ambient temperature.

The boundary conditions can be written as:

$$\begin{cases} k_{\text{cbs}}^z \frac{\partial T_{\text{cbs}}}{\partial z} = Q_c / A_c, & \text{on } \Gamma_1 \\ k_{\text{cbs}}^z \frac{\partial T_{\text{cbs}}}{\partial z} = -h_z (T_{\text{cbs}} - T_{\infty}), & \text{on } \Gamma_2 \setminus \Gamma_1 \\ k_{\text{alc}} \frac{\partial T_{\text{alc}}}{\partial z} = -h_z (T_{\text{alc}} - T_{\infty}), & \text{on } \Gamma_5 \end{cases} \quad (3)$$

where Q_c and A_c are the total heat input to the carbon strip and the actual cross-sectional area between carbon strip and CW, while h_z is the shared average convective heat transfer coefficient on the upper/lower surfaces. In addition, since both side surface of the CS are partially shielded by the panhead horns in real operation conditions, heat transfer in these regions is dominated by conduction rather than convection, the coefficient h_x then can be neglected.

The total heat input to the carbon strip through the contact area is defined considering Joule and frictional heating, with the arc heating ignored under the assumption that the quality of current collection is sufficiently good to make this term negligible, thus:

$$Q_c = \eta (\mu F V + R_c I^2) \quad (4)$$

where F , μ and R_c represent the contact force, the friction coefficient and the ECR, respectively, V is the trains' operation speed, and I is the electric current flow. η is the heat partition coefficient of heat transfer to the carbon strip.

For the thermal boundary condition at Γ_3 , the carbon strip is assumed to be in perfect contact with the aluminum carrier, without any additional contact resistance. This assumption implies both heat flux and temperature continuity across Γ_3 :

$$\begin{cases} k_{\text{cbs}}^z \frac{\partial T_{\text{cbs}}}{\partial z} = k_{\text{alc}}^z \frac{\partial T_{\text{alc}}}{\partial z}, \text{ on } \Gamma_3 \\ T_{\text{cbs}} = T_{\text{alc}} \end{cases} \quad (5)$$

Then, the CS thermal behavior could be computed with the transient thermal model Eq. (1) and the boundary conditions described by Eqs. (3)~(5). The problem is solved using the explicit FDM. A 2D spatial grid is defined over the CS and based on this discretization, the temperature gradients are defined using central difference approximations, resulting in the following set of discrete-time equations:

$$\mathbf{T}^{n+1} = \mathbf{A}\mathbf{T}^n + \mathbf{B}^n \quad (6)$$

where $\mathbf{T}$ is the vector defining the temperature at the points of a spatial grid matrix, $\mathbf{B}$ is the vector of the thermal boundary conditions, $\mathbf{A}$ is the coefficient matrix representing the thermal diffusivity, and superscripts $n+1$ and n denote two successive iterative times. According to the discretization scheme mentioned before, the $\mathbf{A}$ matrix exhibits a banded sparse structure. Taking the carbon strip as an example, the diagonal and off-diagonal terms read as:

$$\begin{aligned} A_{p,p} &= 1 - \frac{2k_{\text{cbs}}^x \cdot \Delta t}{\rho_{\text{cbs}} c_{\text{cbs}} (\Delta x)^2} - \frac{2k_{\text{cbs}}^z \cdot \Delta t}{\rho_{\text{cbs}} c_{\text{cbs}} (\Delta z)^2} - \frac{2h_y \cdot \Delta t}{W \cdot \rho_{\text{cbs}} c_{\text{cbs}}} \\ A_{p\pm 1,p} &= \frac{k_{\text{cbs}}^x \cdot \Delta t}{\rho_{\text{cbs}} c_{\text{cbs}} (\Delta x)^2}, \quad A_{p,p\pm 1} = \frac{k_{\text{cbs}}^z \cdot \Delta t}{\rho_{\text{cbs}} c_{\text{cbs}} (\Delta z)^2} \end{aligned} \quad (7)$$

where p represents the discrete nodal index, Δx and Δz denote the discrete size in the x - and z -directions, respectively, and Δt is the time interval. Accordingly, the vector $\mathbf{B}$ can be written as:

$$B_p = \frac{2h_y \cdot \Delta t}{W \cdot \rho_{\text{cbs}} c_{\text{cbs}}} T_{\infty} \quad (8)$$

Based on the above thermal behavior analysis of the collector strip, the accuracy of the temperature prediction depends on several key parameters, including the thermo-physical properties of the collector strip, the heat input and the convective heat transfer coefficients. In this study, an actual-scale pantograph collector strip is considered, and the unknown thermal parameters include the thermal conductivities k_{cbs}^x and k_{cbs}^z , as well as the volumetric heat capacity $\rho_{\text{cbs}} c_{\text{cbs}}$ of the collector strip. In addition, the boundary condition parameters related to the average convective heat transfer coefficients also need to be determined.

2.2 Inverse problem

An effective method for parameter estimation involves the application of multiple sets of thermal experimental data to the numerical thermal model, and the minimization of an objective function represented by the 2-norm of the difference between the measurements T^{exp} at all test points and discrete time instants and the corresponding simulation results T^{sim} .

The problem can be formulated as follows:

$$\min_{\mathbf{X}} \Xi(\mathbf{X}), \quad \Xi(\mathbf{X}) = \|\mathbf{F}(\mathbf{X})\|^2 = \sum_{nc=1}^{N_c} \sum_{i=1}^{N_c} \sum_{j=1}^{N_t} w_{nc,i,j}^2 \left(T_{nc,i,j}^{\text{sim}}(\mathbf{X}) - T_{nc,i,j}^{\text{exp}} \right)^2 \quad (9)$$

where $\mathbf{X}$ is the vector of unknown parameters to be identified, w is the weight factor, N_c , N_c

and N_t are the number of operating conditions, measurement points and time instants, and superscripts *sim* and *exp* denote simulation results and measurements, respectively.

This problem can be efficiently solved using the Levenberg-Marquardt method (LMM) [22], iteratively updating the unknown parameters vector $\mathbf{X}$ according to:

$$\mathbf{X}^{(m+1)} = \mathbf{X}^{(m)} + \Delta\mathbf{X} \quad (10)$$

with m the index of the iteration. The parameters increment is obtained by:

$$\left(\mathbf{J}^{(m)T} \mathbf{J}^{(m)} + \lambda^{(m)} \mathbf{D}^{(m)} \right) \cdot \Delta\mathbf{X} = -\mathbf{J}^{(m)T} \cdot \mathbf{F}^{(m)} \quad (11)$$

where $\lambda^{(m)}$ is the positive parameter set to ensure convergence and update at each iteration step [26], $\mathbf{D}^{(m)} = \text{diag}(\mathbf{J}^{(m)T} \mathbf{J}^{(m)})$ is the scaling matrix, $\mathbf{J}$ is the sensitivity matrix determined by the function $\mathbf{F}(\mathbf{X})$ with respect to $\mathbf{X}$, and superscript T denotes the transpose of a matrix. The sensitivity matrix can be written as:

$$\mathbf{J}^{(m)} = \left[J_{(nc,i,j),s}^{(m)} \right], J_{(nc,i,j),s}^{(m)} = \frac{\partial \mathbf{F}_{nc,i,j}^{(m)}}{\partial \mathbf{X}_s^{(m)}} \quad (12)$$

where s represents the s th parameter to be identified.

Combining the Eqs. (6)~(8) and (12), the explicit sensitivity matrix can be written as:

$$J_{(nc,i,j),s}^{(m)} = \frac{\partial \mathbf{F}_{nc,i,j}^{(m)}}{\partial \mathbf{X}_s^{(m)}} = \frac{\partial \mathbf{T}_{nc,i,j}^{(m)}}{\partial \mathbf{X}_s^{(m)}} = \mathbf{A}^{(m)} J_{(nc,i,j),s}^{(m-1)} + \frac{\partial \mathbf{A}^{(m)}}{\partial \mathbf{X}_s^{(m)}} \mathbf{T}_{nc,i,j}^{(m)} + \frac{\partial \mathbf{B}^{(m)}}{\partial \mathbf{X}_s^{(m)}} \quad (13)$$

Compared with the conventional finite difference method and the complex-variable method [27], Eq. (13) allows the sensitivity matrix to be obtained directly during the forward solution, without introducing additional perturbations to the parameters. This significantly simplifies the computation procedure and improves the efficiency.

The iteration procedure is terminated when either of the criteria in Eq. (14) is satisfied:

$$\|\nabla \Xi(\mathbf{X})\| \leq \delta_1 \text{ or } \|\Delta\mathbf{X}\| \leq \delta_2 \quad (14)$$

where $\nabla \Xi(\mathbf{X}) = 2 \cdot \mathbf{J}(\mathbf{X})^T \cdot \mathbf{F}(\mathbf{X})$ is the gradient of the objective function, δ_1 and δ_2 are predefined tolerances. Once convergence is achieved, the updated parameter vector $\mathbf{X}^{(m+1)}$ is regarded as the identified and acceptable solution.

3. Experimental setup

3.1 Experimental test rig

A simple static contact temperature rise test rig built in Politecnico di Milano was used to analyze the thermal behavior of collector strips, to gather the experimental data required for the solution of the inverse problem and to verify the accuracy of the thermal forward model. The test rig is shown in Fig. 2. It includes a sample of the contact wire, the entire collector strip, an industrial fan, five Chromel/Alumel K-type thermocouples with an operating range from -220°C to 1260°C , a power supply, and the data acquisition system (DAQ).

A short section of CW, clamped by wire clips, is set in contact with the CS, which is fixed to

an insulated support base. The industrial fan provides forced airflow cooling of the test area, reproducing the cooling effect caused in the real service by the relative motion between the CS and the air. The speed of the airflow generated by the fan is regulated by means of the fan's own control system. To ensure good contact conditions between the two bodies, the CW was ground, creating an 8 mm wide flat contact region, as shown in Fig. 2 (c). The CW specimen and the CS are pressed against each other by placing weights on the wire clips, so that the contact force between the two bodies is well controlled and precisely known.

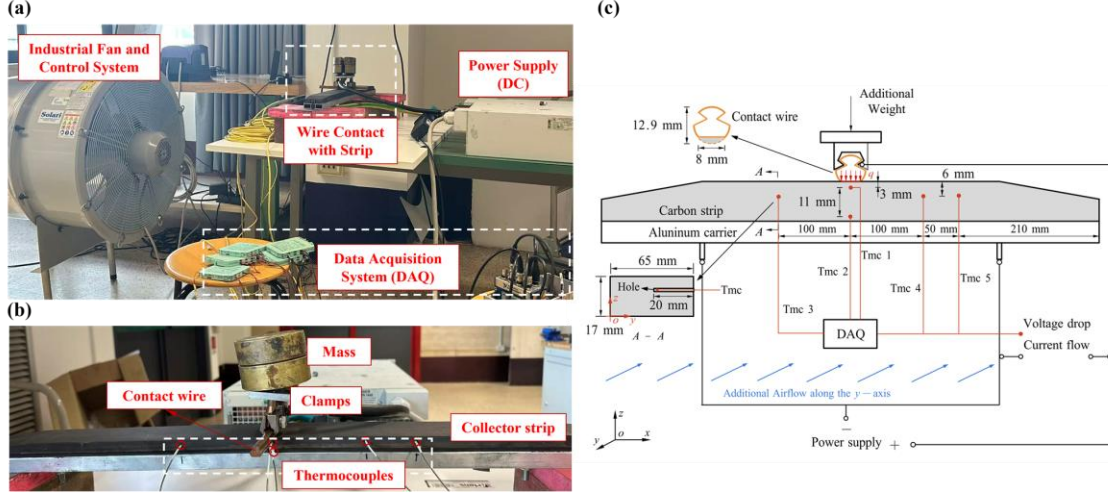


Fig. 2 The experimental test rig: (a) Overall view; (b) Front view of the wire contact with strip; (c) Detailed schematic of the test rig components.

Thermocouples are placed at the center and on both sides of the CS (the details as shown in Fig. 2 (c)). Two measurement points are set at the centerline of the CS, with one (Tmc 1) located near the upper surface and the other (Tmc 2) near the interface of the CS and the aluminum carrier. The remaining three measurement points are placed on both sides of the CS. Tmc 5 is located close to the farthest position at which contact with the CW can occur in real service, while Tmc 3 is positioned 100 mm from the CS centerline and Tmc 4 is symmetrically arranged with respect to Tmc 3 about the centerline. The voltage drops and the electrical current flow between the CS and the CW are measured concurrently. The data are recorded using the DAQ described in [28], consisting of a NI cDAQ-9178 chassis and two NI 9239 modules. The experimental data are recorded with a sampling frequency of 100 Hz.

3.2 Experimental scheme and data processing

The experiments were performed under different levels of contact force, electrical current flow and airflow speed, as shown in Table 1. The table also reports the own weight of the CW with clips.

Table 1 Summary of the testing conditions

Scope of test	Parameters	Values
Contact resistance	Own weight of CW and clips (kg)	0.5
	Additional Weights (kg)	0, 1, 2, 4, 6, 8
	Current flow (A)	100, 150, 200
Temperatures under natural convection	Additional Weights (kg)	2, 4, 8
	Current flow (A)	50, 100, 200

Temperatures under forced convection	Basic	Additional weight (kg)	2
	settings	Current flow (A)	200
		Additional air flow (m/s)	4, 8, 12, 16

The experiments were performed to address three main scopes outlined below.

Scope 1: Electrical contact resistance measurement. The ECR was measured under different contact forces and current flow values. For each contact force, the current flow value was adjusted and the voltage drop between the CS and CW was recorded under steady-state conditions. Each test condition was repeated five times and the measurements were averaged over the five nominally identical tests to reduce the effect of exogenous disturbances. In this way, the contact voltage drop ΔU was obtained and its ratio to the measured current flow I provides the estimated ECR R_c :

$$R_c = \frac{\Delta U}{I} \quad (15)$$

Scope 2: Temperature measurements under natural convection conditions. This part of the experiments was conducted without additional airflow and was performed in a closed room, so it can be regarded as being under natural convective condition. Each test consisted of a 10-minute power-on phase. The test conditions were designated using a nomenclature that combines the current flow and the additional weight. For instance, C200M8N denotes the test performed under 200 A current flow, with an additional weight created by an 8 kg mass and under natural convection. After completion of a test, the next one was not started until the CS had completely cooled down.

Scope 3: Temperature measurements under forced convection. In these tests, the industrial fan was used to generate an additional airflow. For safety reasons, the maximum airflow speed reached only 16 m/s, which corresponds to a train speed of 57.6 km/h. Considering that the forced convective condition primarily governs the cooling phase, each test condition was maintained for 20 minutes, with 7 minutes power-on and 13 minutes power-off. The additional airflow was applied throughout the entire test period.

4. Experimental Results

4.1 Electric contact resistance

4.1.1. Experimental results

The ECR measurements performed using the setup described in Sec. 3 are presented in Fig. 3. Fig. 3 (a) shows the results in the form of mean ECR values and standard deviations, while (b) shows only the mean values for each condition to avoid overlapping of the curves. Although a certain degree of dispersion is observed among individual experiments, the error bars are relatively small for most test conditions, indicating good stability of the ECR measurements. A clear trend can be discerned with the ECR decreasing for increasing contact force for the same current flow. Furthermore, the ECR shows an approximately linear decreasing trend with the increase of current flow for the same contact force.

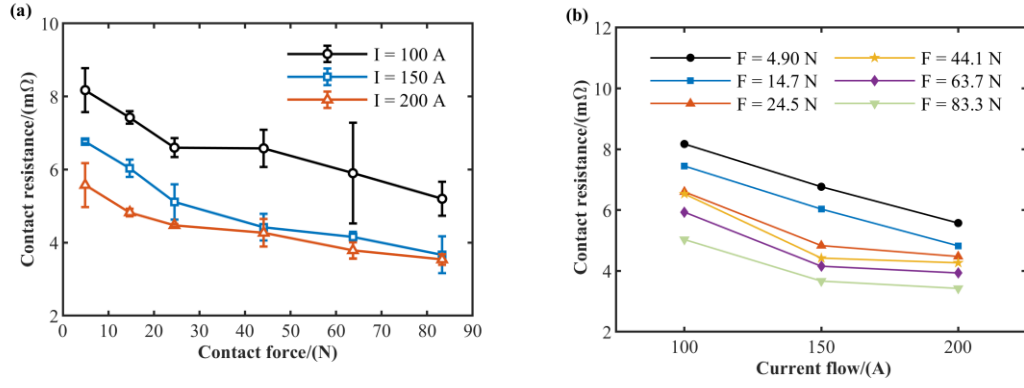


Fig. 3 Relationships between ECR with contact force (a) and with current flow (b)

According to the electrical contact theory, the transfer of electric current between two contacting bodies takes place through small regions corresponding to the contact between the asperities on the two surfaces. The transfer of electric current through these conductive contact spots gives rise to the constriction resistance [29]. Based on the correlation between contact force and the contact spots, it can be concluded that the ECR is inversely proportional to some power m_c of the contact force: $R_c \propto F^{-m_c}$. The exponent m_c is approximately 1/2 for plastic deformation and 1/3 for elastic deformation, while in real engineering applications it falls within the 0.5-1 range [30].

Therefore, the interpretation of Fig. 3 is as follows: the PCS contact area and the number of contact spots increase with the contact force, resulting in an increase in the average contact radius and a decrease in the ECR. However, as the contact force further increases, the contact area continues to grow but at a decreasing rate, leading to a reduction in ECR which is less than inversely proportional to the force. On the other hand, the flow of electric current raises the temperature of the contact surface, thereby softening the contact asperities and causing the radius of the contact spots to increase and ultimately leading to a decrease of the ECR.

4.1.2. Mathematical model of the ECR

The ECR is directly related to the heat input to the CS and serves as a key parameter in the thermal behavior analysis of the CS. However, it is difficult to directly use the ECR as a prescribed input parameter in the analysis. Therefore, it is extremely useful to synthesise the results of ECR measurements in a mathematical expression describing how the ECR varies as function of the contact force and current flow.

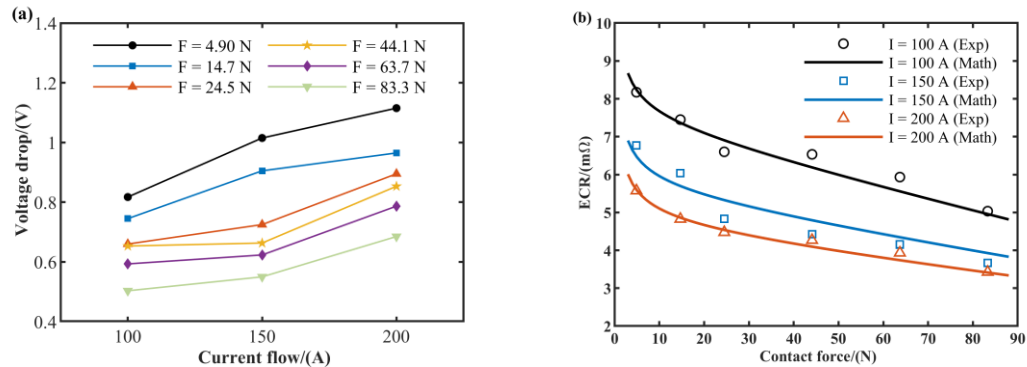


Fig. 4 The experimental results and mathematical model validation. (a) experimental results of voltage drop (b) comparison between mathematical and experimental results of ECR.

By applying Eq. (15), an ECR mathematical model can be obtained from the measured voltage drop and current flow. From Fig. 4 (a) it is noted that, under a same value of the contact force F , the voltage drop ΔU is approximately linearly related to the current flow I . Then a linear relationship between the voltage drops and current flow is assumed and a mathematical expression for the ECR is established as follows:

$$R_c(F, I) = \frac{\Delta U}{I} = \frac{a(F)I + b(F)}{I} = a(F) + b(F)I^{-1} \quad (16)$$

where the $a(F)$ and $b(F)$ are force-dependent coefficients whose values are found by fitting the voltage drop and current data, leading to the following expression of the ECR mathematical model:

$$R_c(F, I) = (3.1236 \cdot F^{-0.4738} - (2.8F - 542) \cdot I^{-1} + 1.4843) \times 10^{-3} \text{ } (\Omega) \quad (17)$$

This expression includes the power-law decay relationship between contact force and ECR with a decay exponent close to 1/2, and the combined contribution of contact force and current flow to the ECR value. The comparison between the established mathematical model of the ECR, Eq. (17), and the experimental results is shown in Fig. 4 (b). A good agreement is observed, although slight deviations occur for large values of the current flow. The root mean square error (RMSE, see Eq. (18)) is 0.25 m Ω , with ECR values falling in the 3.5-9 m Ω range, thus confirming the good representativity of the model established.

$$RMSE = \sqrt{\frac{1}{N_c} \sum (R_c^{\text{Math}} - R_c^{\text{Exp}})^2} \quad (18)$$

4.2 Thermal behavior of the PCS

4.2.1. Thermal behavior under natural convection conditions

The results of the thermal tests performed under natural convection are shown in Fig. 5. A common temperature trend is observed, despite the diverse conditions applied in the tests. Generally, increasing the contact force or decreasing the current flow leads to a reduction in the magnitude of the temperature rise of the CS. This difference can be attributed to variations in the heat input, which are dictated by the ECR and the current flow.

The details of the temperature rise in the CS can be described as follows: after power-on, the temperatures measured by Tmc 1 and Tmc 2 increase rapidly and then gradually exhibit a decelerating rate of rise, with both points showing similar temperature evolutions. The temperatures recorded by the other thermocouples, set at a larger distance from the point at which the current is fed to the CS, rise after a short delay, and their rates of increase become smaller with increasing distance from the contact area. The symmetric temperature responses recorded by Tmc 3 and Tmc 4 confirm the reliability of the experimental setup.

In addition, the CS temperature under specific conditions did not exhibit a strictly steady increase. For instance, under condition C200M4N, the temperature responses recorded by Tmc 1 showed slight fluctuations within its overall increase. This phenomenon may be attributed to the

random occurrence of minor electric arcs at the contact interface during high-current flow.

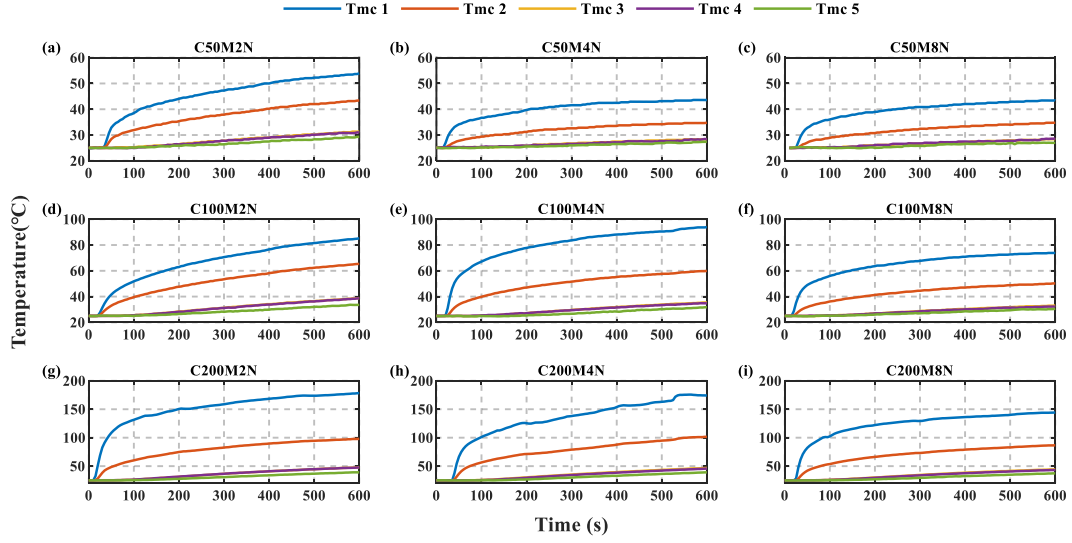


Fig. 5 Temperature measurement results under natural convection

4.2.2. Thermal behavior under forced convection conditions

Similar to the experimental results under natural convection conditions, the temperature trends under forced convection at different airflow speeds are generally consistent. The experimental results under a current flow of 200 A, a mass of 2 kg and an airflow speed of 4 m/s are presented in Fig. 6 (a). Due to the additional airflow covering the entire test process, the temperature rise behavior differs from that under natural convection conditions.

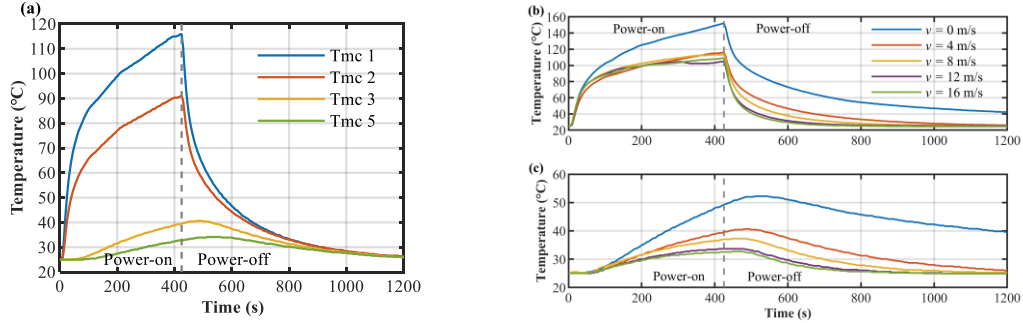


Fig. 6 Temperature measurement results under forced convection at 200 A current flow and 2 kg mass. (a) experimental results with the additional airflow at 4 m/s; (b) comparison results of all conditions at Tmc 1; (c) comparison results of all conditions at Tmc 3.

To compare more details of the experimental results, the temperature results at Tmc 1 and Tmc 3 were extracted under various airflow speeds, as shown in Fig. 6 (b) and (c). Compared with natural convection conditions, the temperature at Tmc 1 under forced convection rises more gradually, and the final temperature increase is significantly lower. However, during the power-on state, the temperature evolution under different airflow speed shows similar trend, with only slight variations. During the power-off state, the cooling rate is strongly influenced by the airflow speed. As for Tmc 3, the temperature behaviors more clearly reflect the differences induced by varying airflow speeds, the maximum temperature gradually decreases with the additional airflow speed increases, and the time to reach the maximum temperature shows a decreasing trend. Under higher airflow speeds, such as 12 m/s and 16 m/s, the temperature results show basically the same trend, with small

numerical differences.

Under natural convection conditions, instead, the temperature decrease in the cool-down phase is significantly slower and at the end of the test the temperature is still approximately 40 °C, significantly higher than the ambient temperature. With increasing additional airflow speed, the cooling temperature gradually approaches the ambient temperature. Under higher airflow speed conditions, the time required for the CS to cool to ambient temperature is significantly reduced.

5. Parameter identification and validation

In the previous section, the thermal behavior of the CS under both natural and forced convection conditions was characterized through experiments. In this section, the inverse method described in Sec. 2.2 is employed to identify the unknown thermal parameters of the carbon strip and the convective heat transfer coefficients based on the experimental results and the forward model. The convective heat transfer coefficients are selected as 8 W/(K·m²) when identifying the unknown thermal parameters under natural convection conditions [31].

5.1 Thermal parameters identification

The experimental results under natural convection conditions are divided into two groups: Group A (C50M8N, C100M4N, C200M2N and C200M8N) for the identification, and Group B (C50M2N, C50M4N, C100M2N, C100M8N and C200M4N) for validation. Group A comprises four experimental conditions, in which different combinations of current flow and mass levels are considered.

The experimental results from all operating conditions in Group A are assembled into a single matrix for thermal parameter identification. To ensure a balanced inversion process and mitigate the bias introduced by variations in temperature rise magnitudes across different conditions, a weighting factor is incorporated. The factor is defined as $w = 1/\Delta T_{nc}^{\exp}$, where $\Delta T_{nc}^{\exp}$ denotes the maximum temperature rise for each experimental condition. In addition, the values of volumetric heat capacity and thermal conductivity vary by orders of magnitude. Therefore, a logarithmic mathematical treatment to ensure the numerical stability in the identification process is used. The unknown vector of parameters object of the identification process is $\mathbf{X} = [k_{\text{cs}}^x \quad k_{\text{cs}}^z \quad \log(\rho_{\text{cs}} c_{\text{cs}})]$.

$$RE = \left(|T^{\text{sim}} - T^{\text{exp}}| / T^{\text{exp}} \right) \times 100\% \quad (19)$$

The identified thermal parameters, after rounding, are $k_{\text{cs}}^x = 5.97 \text{ W}/(\text{K} \cdot \text{m})$, $k_{\text{cs}}^z = 13.66 \text{ W}/(\text{K} \cdot \text{m})$ and $\rho_{\text{cs}} c_{\text{cs}} = 1.893 \times 10^6 \text{ J}/(\text{K} \cdot \text{m}^3)$. The relative errors (REs), between the measured and reconstructed temperature trend, calculated according to Eq. (19), are presented in the form of error bars in Fig. 7 (a) for all tests in Group A. The relative error bars are characterized by the mean relative error and its standard deviation. It can be observed that the mean REs are all below 5%, and the corresponding standard deviations demonstrate that the relative errors exhibit limited dispersion for most conditions, indicating a high degree of consistency between the measured and reconstructed temperature distributions, both in magnitude and in temporal evolution.

To further evaluate the applicability of the identified thermal parameters, the thermal model

with the parameters set is applied to the tests in Group B. The comparison results are also shown in Fig. 7 (a). In most cases, the mean REs remain are low, about 5% with relatively small standard deviations, while only for test C100M2N the relative error and standard deviation are significantly larger. The larger deviations observed for this test are likely due to the random occurrence of minor electric arcs between the CS and the CW, which induce sudden local temperature fluctuations. In addition, the comparison between numerical and experimental temperature distributions for test C200M8N is shown in Fig. 7 (b), the numerical results exhibit good agreement with the measurements.

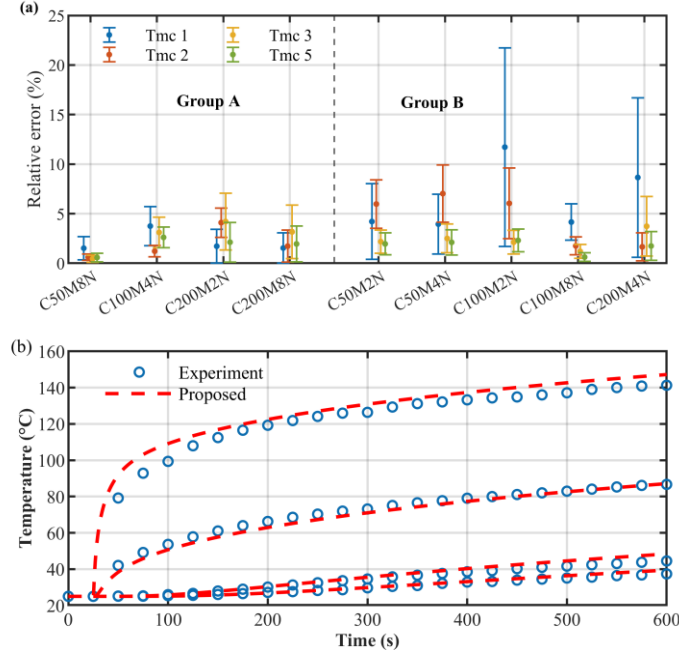


Fig. 7 Application of the identified thermal parameters and comparison. (a) Relative error bars; (b) Comparison among experimental and proposed numerical results for test C200M8N.

Generally, the identified thermal parameters enable the forward numerical model to accurately reproduce the temperature distribution of the CS at different experimental conditions under natural convection, validating the process of parameters identification and confirming the reliability of the established forward numerical model.

5.2 Forced convective heat transfer coefficient

Based on the identified thermal parameters of the carbon strip, the average convective heat transfer coefficients are identified in this section. The continuous contact between the CW and CS locally obstructs the airflow, leading to non-uniform forced convection on the strip surface. Consequently, the local heat transfer coefficient significantly influences the temperature gradients near the contact region, such as at Tmc 1. To eliminate the uncertainty introduced by this difference, the experimental temperature results at Tmc 3, which is relatively distant from the contact region, are used for the identification of the convective heat transfer coefficients under forced convective.

The identification results of the forced convective heat transfer coefficient are presented in Fig. 8 (a), where the values exhibit an upward trend with increasing additional airflow speed. The data fitting is based on the power-law relationship between the forced convective heat transfer

coefficients and the additional airflow speed mentioned in Refs. [18,32]. It should be noted that, the natural convection coefficients at zero additional airflow speed are shown as a reference and not used in the power-law relationship fitting. The relations are:

$$h_y = 16.02v^{0.5590}, h_z = 19.79v^{0.6627}, v \geq 4 \text{ m/s} \quad (20)$$

The exponents in Eq. (20) are in line with the values of 1/2 for the front surface and 2/3 for the top and bottom surfaces reported in Ref. [32] for a rectangular cylinder under cross-flow conditions. The temperature trend and relative error curves of comparison between experimental and numerical results a current flow of 200 A, a mass of 2 kg at Tmc 3 are shown in Fig. 8 (b). It can be observed that, the numerical results exhibit good agreement with the experimental measurements, with the relative errors generally below 4%.

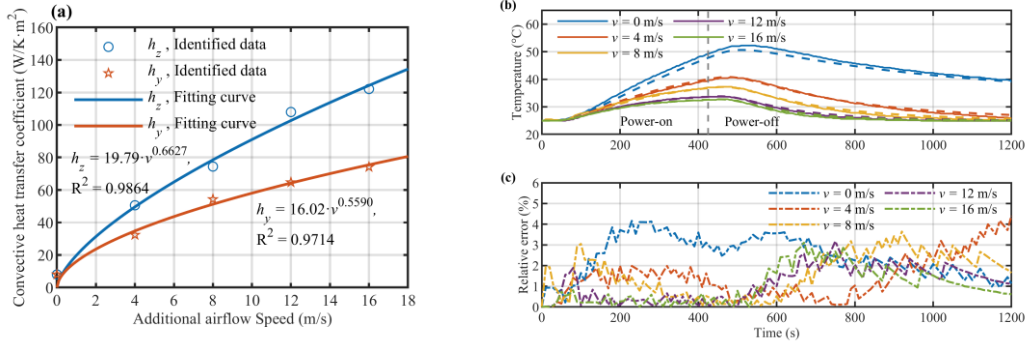


Fig. 8 Identification results of the convective heat transfer coefficient and temperature comparison: (a) Parameter identified and fitted results; (b) Comparison between experimental (solid lines) and numerical results (dotted lines) at Tmc 3; (c) Relative error distribution at Tmc 3.

In addition, the measured points located at different positions, Tmc 2 and Tmc 5, are selected for validation and comparison. The comparison of temperature evolution between numerical and experimental results at airflow speed of 4 m/s is presented in Fig. 9 (a). A good agreement is observed at Tmc 5 in the temperature evolution, indicating that the identified coefficients can effectively characterize the overall heat dissipation behavior of the CS surface. However, at Tmc 2, a temperature difference of approximately 8 °C is observed. This discrepancy can be attributed to the fact that Tmc 2 is located closer to the contact region, where localized convective influences the temperature behavior.

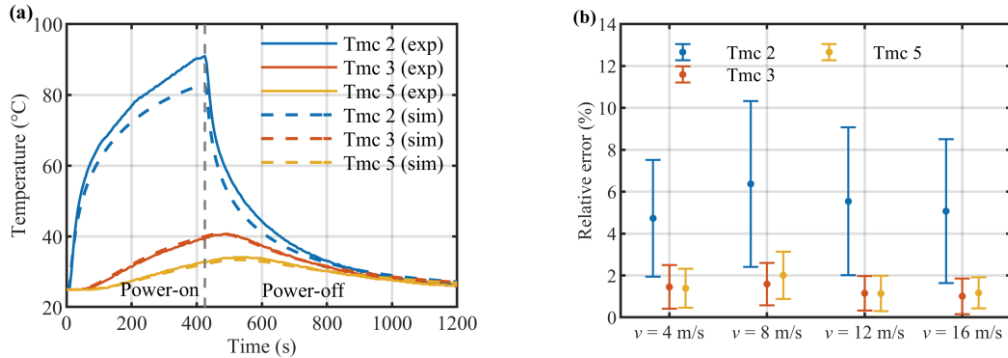


Fig. 9 The comparison between numerical and experimental results at 200 A current flow and 2 kg mass. (a) Under additional airflow speed of 4 m/s; (b) Relative error bars.

The relative error bars between the experimental and numerical results at Tmc 2, Tmc 3 and Tmc 5 are statistically analyzed and presented in Fig. 9 (b). The mean relative errors are all below 8%, with most of them remaining at or below 2%, especially at Tmc 3 and Tmc 5, and the corresponding standard deviations indicate a low dispersion of the relative errors.

In general, the numerical simulations accurately reproduce the temperature distributions at other measured positions, this indicates that the identified convective heat transfer coefficients are applicable for the study of the CS thermal behavior.

6. Conclusions

This paper delves into the thermal behavior of the pantograph collector strip under static contact with catenary wire under natural and forced convective conditions. For the first time, the unknown thermal parameters and the forced convective heat transfer coefficients are identified through the numerical and experimental combined inverse analysis, significantly improving the reliability of thermal analysis for the PCS. The main conclusions are summarized as follows:

1. The proposed numerical thermal model for the actual-scale pantograph collector strip accurately reproduces its temperature distribution, and good agreement is achieved with experimental measurements.
2. The unknown thermal parameters of the collector strip are identified using temperature measurement under natural convection. When the identified parameters are applied to additional experimental conditions, the numerical results reproduce the temperature evolution trends with good accuracy, most of mean relative errors are in most cases within 5%.
3. The forced convective heat transfer coefficients of the collector strip are identified using temperature measurements obtained at different airflow speeds. The identified correlations between the coefficient and the airflow speed exhibit a power-law relationship, agree well with published literature. Acceptable relative errors are achieved at most locations where temperature measurements were carried in the tests, indicating the applicability of the identified coefficients for the detailed analysis of the CS thermal behavior under forced convection conditions.

Furthermore, the validated forward thermal model and the identified parameters established in this paper provide a reliable thermal foundation for further investigations into the pantograph collector strip wear behavior.

Disclosure statement

The authors report there are no competing interests to declare.

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