

Two-phase Flow Characteristics of Evaporative Cooling System for Large-capacity High-speed Hydrogenerators' Rotor

Jiaqi Rao, Bin Xiong, and Ruiwei Li

Abstract—Research of evaporative cooling system for a rotor is of great significance for large-capacity, high-speed hydrogenerators. The two-phase flow characteristics are important parameters of the evaporative cooling system, and it is urgent to research the two-phase flow characteristics under rotating conditions. In this paper, based on the self-developed experimental platform of pipe-cooled cooling evaporative cooling system for a rotor, a real-time image acquisition system is established. Experiments are conducted under different thermal loads and centrifugal accelerations, and two-phase flow images of the gas outlet section of the system are collected to analyze the characteristics and patterns of two-phase flow. The results show that under the action of a strong centrifugal force field, the two-phase flow in the pipe-cooled cooling evaporative cooling system for a rotor is more turbulent and complex. Bubbles are more likely to break up and form a dispersed flow. Increasing the thermal load can reduce the instability of the two-phase flow in the system, while increasing the centrifugal acceleration can raise the system pressure and exacerbate the instability of the two-phase flow in the system.

Index Terms—Hydrogenerator, Large capacity, High speed, Evaporative cooling system, Rotating, Analysis of images, Characteristics of two-phase flow.

I. INTRODUCTION

WITH the development of the hydropower industry, the capacity of low-speed hydrogenerators is continuously increasing. For instance, the largest hydrogenerator in the Baihetan Hydropower Station has reached a capacity of 1000 MW. However, the capacity of high-speed hydrogenerators used in pumped storage power stations or high-head hydropower stations is only half that of low-speed hydrogenerators. One of the main reasons for this outcome is the difficulty in rotor cooling.

Traditional low-speed hydrogenerators rely on the self-ventilation design of the large-diameter rotor to achieve

effective cooling. In contrast, high-speed hydrogenerators, which have a slender structure and smaller diameter, cannot meet the cooling requirements of large-capacity, high-speed units through self-ventilation alone [1]-[2]. Although water cooling technology has the advantages of high utilization coefficient of the motor, uniform coil temperature, and good thermal stability of the motor core [3], dual water-cooled generators with both stator and rotor cooling often suffer from leakage at the joints or rupture of composite pipes due to the combined effects of centrifugal force and vibration [4]-[5], resulting in a much higher failure rate than stator water-cooled generators.

In addition to the water-inner-cooling technology for generator rotors, researchers have conducted extensive work on cooling issues in other rotating machinery (e.g., machine tool spindles), aiming to suppress thermal deformation and improve machining accuracy by optimizing cooling strategies. For example, Hsieh *et al.* in [6] applied an adaptive neuro-fuzzy inference system (ANFIS) to predictively control the coolant demand of a spindle, effectively reducing thermal deformation. Li *et al.* in [7] optimized the cooling channel design of an internally cooled spindle, lowering thermal deformation from 53.8 to 30.9 μm and shortening the time to reach thermal steady state. [8] proposed a variable oil volume (VOV) control method that adjusted the coolant flow rate according to spindle speed and load, achieving a reduction in thermal deformation of more than 70%. These methods mainly enhance heat dissipation by strengthening convective heat transfer or intelligently regulating the flow rate of single-phase coolants (oil or water).

Different from the above, the self-circulating evaporative cooling technology is based on the principles of phase change heat transfer and vapor power cycles. It utilizes the principle of heat absorption through the boiling and vaporization of the cooling medium to remove heat. While retaining the advantages of water cooling technology, it also offers the benefits of a simple and easy-to-maintain operating system, low operating pressure to prevent leakage, and non-corrosive properties. Existing studies have shown that the evaporative cooling system for a rotor has significant superiority in terms of cooling performance [9]-[10]. However, the mechanical structure of the evaporative cooling system for a rotor is in a rotating state. The establishment of its self-circulating head is not only affected by gravity but also by centrifugal force.

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Under the combined influence of gravitational and centrifugal acceleration, the two-phase flow characteristics, heat transfer characteristics, and circulation characteristics of the evaporative cooling system for a rotor become much more complex. Conducting research on the two-phase flow characteristics of the evaporative cooling system for a rotor will help promote the engineering application of evaporative cooling technology for large-capacity, high-speed hydrogenerators' rotor. Li *et al.* in [11] verified that the rotor evaporative cooling architecture had excellent heat dissipation performance, but they did not observe the specific flow patterns, and the analysis of two-phase flow characteristics was relatively limited.

At present, research on two-phase flow characteristics under rotating conditions is relatively scarce. Yao *et al.* in [12]–[14] used a centrifuge to simulate an overload environment of 0–2*g* and studied the two-phase flow characteristics of water in a 20 mm glass tube, recording the changes in flow patterns with a synchronous rotating camera. Xu *et al.* of [15]–[17] simulated a hypergravity environment of 0–3*g* using a centrifuge to investigate the flow boiling characteristics of R134a in horizontal copper tubes with inner diameters of 1.002, 2.168, and 4.065 mm. In the above studies, the centrifugal acceleration range did not exceed 3 times the gravitational acceleration. However, for typical large-capacity, high-speed hydrogenerator rotors, the centrifugal acceleration can reach hundreds of times the gravitational acceleration. Therefore, the existing research results are difficult to apply directly to such scenarios. Consequently, further investigation into two-phase flow characteristics under high centrifugal acceleration conditions is still needed.

As a preliminary exploration in this direction, this paper intends to select a centrifugal acceleration range of 30*g*–60*g* for experimental study (considering the safety of the rotating experimental platform, this range is feasible for safe equipment operation). The research objective is to clarify the two-phase flow characteristics and behavior in the outlet pipe section of the rotor evaporative cooling system under these conditions. To this end, based on a self-built experimental platform of a pipe-cooled rotor evaporative cooling system, a real-time image acquisition system is constructed. Experiments are carried out under heat loads of 100–500 W and within the above centrifugal acceleration range, and two-phase flow images are collected and analyzed.

II. ROTATING EXPERIMENTAL PLATFORM

A. Rotor Structure of a High-speed Vertical Hydrogenerator

The three-dimensional structural model of the rotor of a high-speed vertical hydrogenerator is shown in Fig. 1. The model mainly consists of a thrust collar, an upper guide bearing collar, an upper shaft, a rotor spider, a brake ring, a lower shaft, magnetic poles, and inter-pole supports. The magnetic pole part includes the core, coil, cooling fins, and insulation.

Among these components, the rotor magnetic pole coil (hereinafter referred to as the coil) is the core heating component of the hydrogenerator rotor and is also the cooling object of the rotor evaporative cooling system.

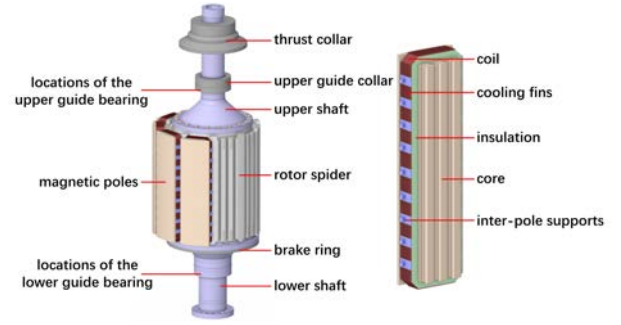


Fig. 1. Three-dimensional structural model of the rotor of a high-speed vertical hydrogenerator.

B. Principle and Characteristics of the Pipe-cooled Evaporative Cooling System for a Rotor

The evaporative cooling system for a rotor operates on the same principle as the evaporative cooling system for stator, utilizing the latent heat of phase change to cool heat sources. The key difference lies in the establishment of the self-circulation driving head. The self-circulation driving head in the evaporative cooling system for stator is influenced by gravity, whereas the evaporative cooling system for a rotor operates under rotational motion, causing its self-circulation driving head to be primarily governed by centrifugal force, with negligible effects from gravitational acceleration. Under high centrifugal acceleration, the circulation and heat transfer characteristics of the self-circulating evaporative cooling system will differ significantly from those in a gravitational field.

The fundamental circulation principle of the evaporative cooling system for a rotor is as follows: The coolant in the heating section absorbs heat generated by the rotor windings, gradually increasing in temperature. Upon reaching saturation temperature, the coolant begins to boil. The resulting vapor-phase coolant, carrying liquid droplets, enters the gas outlet section. The vapor-phase coolant then rapidly condenses back into liquid-phase coolant in the condenser by releasing heat. The density difference between the single-phase medium in the liquid return section and the two-phase medium in the gas outlet section creates a self-circulating driving head within the centrifugal field. This driving head overcomes various resistance losses in the loop, ensuring continuous coolant supply from the liquid return section to the windings. Fig. 2 illustrates the circulation process of the evaporative cooling system for a rotor. In the figure, a_r represents the centrifugal acceleration of the heating section; g denotes gravitational acceleration; ω is the angular velocity of rotation.

Notably, under high centrifugal acceleration, the gas outlet section of the system experiences a sharp drop in static pressure. During this process, as the coolant flows through the gas outlet section, the significant reduction in static pressure causes the originally subcooled saturated liquid to transition into a superheated state, triggering intense vapor-liquid phase change phenomena—referred to as flash evaporation [18]. The presence of flash evaporation complicates the two-phase flow dynamics within the gas outlet section of the evaporative cooling system for a rotor.

C. Overall Structure of the Pipe-cooled Evaporative Cooling System for the Rotor

The pipe-cooled evaporative cooling structure designed based on the aforementioned rotor model of a high-speed vertical hydrogenerator is shown in Figs. 3 and 4. This evaporative cooling structure for a rotor mainly consists of a gas collecting pipe (corresponding to the gas outlet section in Fig. 2), a liquid return pipe (corresponding to the liquid return section in Fig. 2), cooling pipes (used to cooling the coil, corresponding to the heating section in Fig. 2), a rotating water-cooled condenser installed inside the hollow shaft, and a water supply pipe for the condenser. The gas collecting pipe and the liquid return pipe are connected to the condenser by penetrating the interior of the shaft through holes drilled in the upper shaft and the lower shaft. The cooling pipes are embedded in the magnetic pole coil in a multi-parallel configuration and are filled with an evaporative coolant. The condenser is placed inside the hollow shaft, supported by a condenser bracket, and rotates synchronously with the rotor. The condenser bracket is fixed to the interior of the rotor spider by bolts or welding. Grooves are machined inside the upper shaft to secure the inlet and outlet water pipes of the condenser.

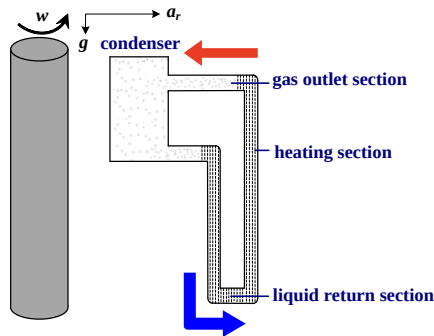


Fig. 2. Circulation process of the evaporative cooling system for a rotor.

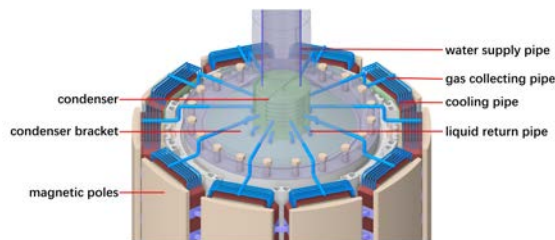


Fig. 3. Model of the pipe-cooled evaporative cooling for the rotor.

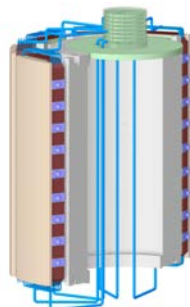


Fig. 4. Model of the rotor spider of the pipe-cooled evaporative cooling for the rotor.

In the evaporative cooling structure for the rotor suitable for a high-speed vertical hydrogenerator, the cooling pipes cool the coil in a multi-parallel configuration. During the evaporative cooling cycle, the liquid coolant is distributed from a single liquid return pipe to each row of cooling pipes, while the gaseous cooling medium is collected from the multiple rows of cooling pipes into a single gas collecting pipe. The connection between the multiple rows of parallel cooling pipes and the gas collecting pipe is shown in Fig. 5: Slots matching the size of the cooling pipes are machined in the gas collecting pipe, and the cooling pipes are embedded into these slots. The multiple cooling pipes and the single gas collecting pipe can be fixed by welding. The connection between the liquid return pipe and the cooling pipes is similar.

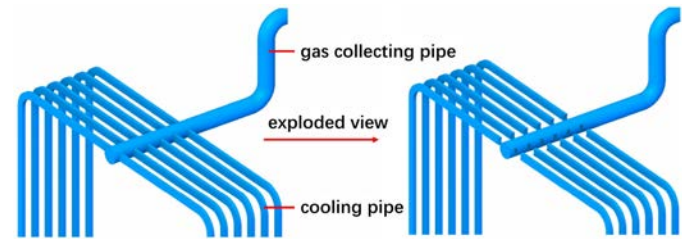


Fig. 5. Connection between the gas collecting pipe and the cooling pipes.

It should be noted that in an actual evaporative cooling system installed in a generator rotor, the component that directly contacts the heat source and achieves cooling is called the “cooling pipe”. However, since the experimental platform in this paper uses electrical heating to simulate the heat source, it is more appropriate to refer to this component as the “heating section”. The two have the same physical function, but to distinguish between engineering application and experimental research, the terminology is not forcibly unified in this paper (the same applies to the gas collecting pipe and the liquid return pipe, which correspond to the gas outlet section and the liquid return section of the experimental platform, respectively).

D. Experimental Platform of the Pipe-cooled Evaporative Cooling System for a Rotor

This research is at an early stage of investigating the two-phase flow characteristics of a rotor evaporative cooling system, with the priority of clarifying the basic two-phase flow behavior without the influence of parallel branches. Therefore, the experimental platform is designed as a single-branch pipe-cooled evaporative cooling system to avoid the complex interference caused by multi-branch coupling.

The model of the experimental platform of evaporative cooling system for rotor is shown in Fig. 6. Two sets of evaporative cooling circulation devices are symmetrically installed on both sides, and the rotating shaft is driven by a drive motor. The evaporative cooling circulation device includes core components essential for principle verification, such as the heating section, gas outlet section, condenser, and liquid return section. A pressure sensor is installed at the inlet of the gas outlet section.

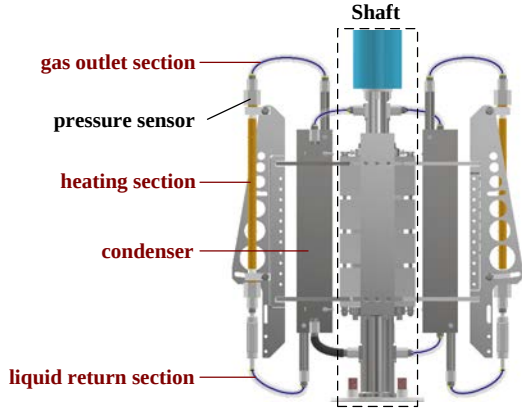


Fig. 6. Experimental platform model of the evaporative cooling system for rotor.

It is worth emphasizing that the pipe of gas outlet section is designed as a transparent flexible hose to facilitate observation.

III. REAL-TIME IMAGE ACQUISITION SYSTEM

Currently, studies employing centrifugal accelerators to simulate hypergravity environments generally utilize experimental platforms with large spatial and operate at relatively low rotational speeds. In contrast, the implementation of visualization methods for the experimental platform of evaporative cooling system for rotor presents additional challenges:

1) The experimental platform of evaporative cooling system for rotor has a more compact structure, requiring cameras with sufficiently small dimensions. This makes it difficult to deploy high-resolution and high-frame-rate cameras.

2) The experimental platform of evaporative cooling system for rotor requires higher centrifugal acceleration. Given its limited size, achieving the necessary acceleration demands higher rotational speeds, thereby imposing stricter reliability requirements on the image acquisition system.

Additionally, to better monitor the system's operational status and enhance automation and integration, real-time transmission of the acquired image signals must also be achieved.

In summary, the image acquisition system for the experimental platform of evaporative cooling system for rotor must meet the following requirements: compact size, lightweight design, high reliability, real-time image signal transmission, and sufficiently high image clarity. Guided by these requirements, a real-time image acquisition system is designed as follows.

To achieve lightweight and compact design of the image acquisition system while enabling real-time image transmission, this study employs the Walksnail Moonlight Kit wireless transmission system along with its matching Moonlight camera for image capture and data transfer. The Walksnail Moonlight Kit consists of two components: a transmitter module mounted on the rotating platform of the evaporative cooling system, and a stationary receiver

connected to the display monitor. Table I presents a comparison of the size and weight between the selected transmitter module and camera in this study and the Canon MVX150i camera used in [12]-[14].

TABLE I
COMPARISON OF SIZE AND WEIGHT BETWEEN THE SELECTED DEVICE IN THIS STUDY AND THE DEVICE IN [12]-[14]

	Name	Size/mm	Weight/g
Walksnail Moonlight Kit	Transmitter module	15.3×34.5×34.5	38.5
	Camera	19.6×19×24	8.9
	Total	—	47.4
	Canon MVX150i camera	75×186×92	625

Additionally, to achieve the goal of high-definition image acquisition, the following solutions are proposed to enhance image quality: 1) Adjust the imaging optical path of the image acquisition device to ensure precise focal length matching, thereby optimizing image quality. 2) Arrange the positions of the image acquisition device and the lighting device reasonably to improve the lighting conditions.

1) Optical path adjustment

When parallel light rays enter the camera lens, an ideal lens converges all rays at the focal point. Fig. 7 illustrates the schematic diagram of the imaging optical path. Here, the lens focal length is f_1 ($f_1 = |OF_1|$); the image distance is v ($v = |OB_1'|$); the object A_1B_1 at object distance u_1 ($u_1 = |OB_1|$) forms a clear, inverted miniature image $A_1'B_1'$ on the photosensor.

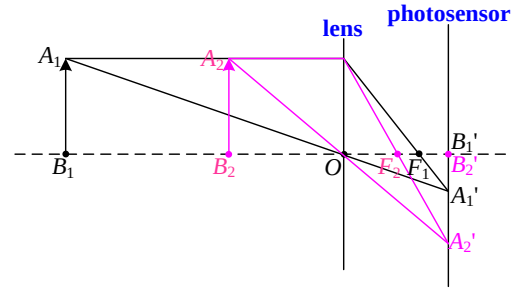


Fig. 7. Schematic diagram of the imaging optical path.

By superimposing a convex lens with focal length f_2 in front of the original lens (as shown in Fig. 8), the combined focal length can be adjusted to f' . This enables the object A_2B_2 at a new object distance u_2 to form a clear, inverted miniature image $A_2'B_2'$ on the photosensor, where the object distance $u_2 = |OB_2|$, and adjusted focal length $f' = |OF_2|$.

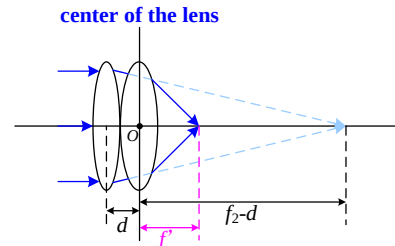


Fig. 8. Schematic diagram of the optical path adjustment.

To maintain a sharp image on the photosensor, the image distance v must remain constant. Thus, f' must satisfy (1):

$$\frac{1}{f'} = \frac{1}{u_2} + \frac{1}{v} \quad (1)$$

Furthermore, the required focal length f_2 of the convex lens for adjusting the optical path can be calculated using (2):

$$\frac{1}{f'} = \frac{1}{f_1} + \frac{1}{f_2 - d} \quad (2)$$

where d is the center-to-center distance between the two lenses.

2) Illumination condition optimization

The lighting conditions significantly impact the quality of experimental video images captured. This study investigates the effects of three typical illumination methods (front lighting, side lighting, and back lighting) on gas-liquid two-phase flow imaging. The results demonstrate that back lighting significantly enhances the contrast of gas-liquid interfaces, making the two-phase boundaries clearly distinguishable. The experiment adopts a backlighting scheme for image acquisition.

Fig. 9 shows the structural schematic diagram of the constructed real-time image acquisition system.

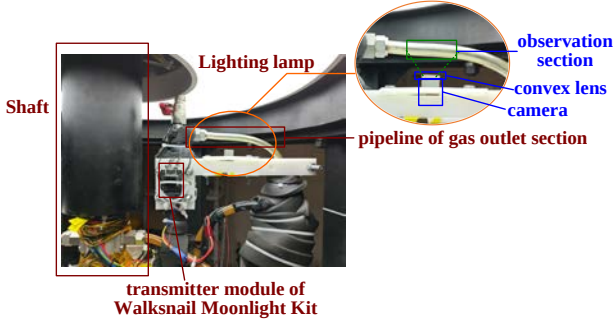


Fig. 9. Structure of the real-time image acquisition system.

In order to obtain a larger field of view, two observation points are designed in total. The specific locations are shown in Fig. 10.

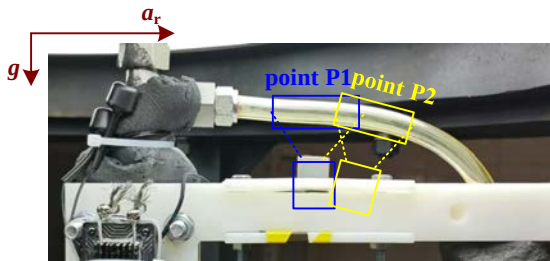


Fig. 10. Setting of observation points.

IV. EXPERIMENT AND ANALYSIS

A. Experimental Platform Parameters and Design of Test Conditions

1) Dimensional design of the rotating experimental platform

Taking a typical large-capacity generator, the Baihetan 1000 MW Francis turbine unit [19], as an example, its rotor has an outer diameter of 16.2 m, a height of 4.1 m, and a rotational speed of 111.1 r/min, corresponding to a centrifugal

acceleration at the rotor outer rim of approximately 110g. For a typical high-speed generator, the Changlongshan 350 MW impulse pump-turbine unit [20] has a rotor height of 10.9 m, a diameter of 4.7 m, and a speed of 600 r/min, giving a centrifugal acceleration at the rotor outer rim of about 926.8g. Due to limited laboratory space, it is impractical to build a 1:1 full-scale model. Therefore, the dimensions of the experimental platform are scaled down based on the aspect ratio of the pump-turbine unit, following the criteria of equal Reynolds number and equal Froude number to maintain mechanical and thermodynamic similarity with the actual machine. The rotating experimental platform of the pipe-cooled self-circulating evaporative cooling system has an outer diameter of 0.8 m for its rotating components. The specific dimensions of the circulation pipes are listed in Table II.

TABLE II
DIMENSIONAL PARAMETERS OF THE CIRCULATION PIPELINE

Parameter	Value
Inner diameter of the pipe of heating section/mm	6
Length of the pipe of heating section/mm	400
Inner diameter of the pipe of gas outlet section/mm	8
Radial length of the pipe of gas outlet section/mm	193.5

The selection principle for the inner diameter of the pipes is as follows: To ensure power generation efficiency and control excitation losses, the rotor current density should not exceed the stator current density, and is typically taken as 4.5–6 A/mm² [21]. According to current technology, a large-capacity high-speed vertical generator reaches a limiting index at a speed of 250 r/min and a capacity of 700 MW, where the maximum width of the rotor coil is 120–125 mm. From handbooks, the corresponding height of a standard soft copper busbar with a width of 125 mm is 8–10 mm [22]. Considering the above constraints, the inner diameter of the cooling pipe in this study is designed to be 6 mm. A schematic cross-section of the hollow copper busbar is shown in Fig. 11, where w is the busbar width; h is the busbar height; d is the cooling pipe diameter.

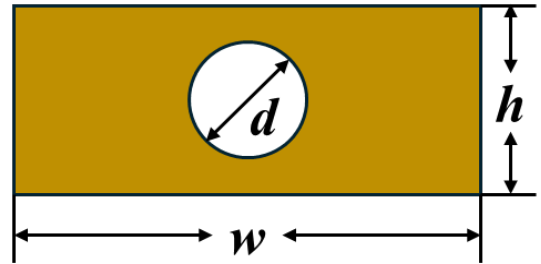


Fig. 11. Structure of the real-time image acquisition system.

2) Safety constraints

High centrifugal acceleration imposes stringent requirements on the installation reliability and structural strength of rotating components. If any component fails due to insufficient strength, the consequences would be severe. Because this rotating experimental platform contains many small

components, the centrifugal acceleration should not be set too high. Tests have shown that operation in the range of $30g$ to $60g$ ensures safety. Therefore, experiments and data collection are planned within this centrifugal acceleration range.

3) Thermal load design

If the current density J is taken as 6 A/mm^2 , the maximum thermal load per unit length $P_{0,\max}$ is calculated as (3):

$$\begin{aligned} P_{0,\max} &= I_{\max}^2 R = J^2 S_{\max}^2 \frac{\rho_1}{S_{\max}} = J^2 S_{\max} \rho_1 \\ &= J^2 \left(w_{\max} h_{\max} - \frac{\pi d^2}{4} \right) \rho_1 \\ &= 6^2 \times \left(125 \times 10 - \frac{\pi \times 6^2}{4} \right) \times 0.018 \\ &\approx 1107.5 \text{ W} \end{aligned} \quad (3)$$

where $I_{\max}$ is the maximum current in a single copper busbar; R is the resistance per unit length of the busbar; $S_{\max}$ is the maximum value of the cross-sectional area of a single busbar can take; ρ_1 is the resistivity of the copper busbar; $w_{\max}$ and $h_{\max}$ are the maximum width and height of a single busbar, respectively.

Given that the heating section pipe length of the rotating experimental platform is 0.4 m , the thermal load on the heating section pipe does not exceed $0.4 \times 1107.5 = 443 \text{ W}$. Accordingly, a series of experiments are planned within the ranges of centrifugal acceleration $a_r = G \cdot g$ (where G is the multiple of gravitational acceleration, dimensionless) from 30 to $60g$, and thermal load P from 100 to 500 W , to observe and record two-phase flow in the gas outlet section. During the experiments, after the system reaches steady state, images of the two-phase flow in the gas outlet section are captured and pressure data at the inlet of the gas outlet pipe are recorded.

4) Coolant filling volume and initial system pressure design

The coolant filling volume directly affects the outlet vapor quality of the heating section. If the filling volume is too small, the outlet vapor quality becomes too high, which may cause operational safety issues. If the filling volume is too large, the outlet vapor quality becomes too low, resulting in insufficient circulation driving force. Considering the dimensions of the circulation pipes and the test conditions, a coolant filling volume of 60 mL is selected in this study. Key thermophysical properties of the coolant are listed in Table III.

TABLE III
KEY THERMOPHYSICAL PROPERTIES OF THE COOLANT

Parameter	Value
Normal boiling point/ $^{\circ}\text{C}$	47.6
Critical temperature ($^{\circ}\text{C}$)	214.1
Critical pressure/MPa	3.39
Saturated liquid density at $20^{\circ}\text{C}/(\text{kg}/\text{m}^3)$	1549
Liquid thermal conductivity at $20^{\circ}\text{C}/(\text{W}/(\text{m}\cdot\text{K}))$	0.068
Liquid thermal conductivity at $75^{\circ}\text{C}/(\text{W}/(\text{m}\cdot\text{K}))$	0.060
Surface tension at $20^{\circ}\text{C}/(\text{mN}/\text{m})$	18.8
Surface tension at $75^{\circ}\text{C}/(\text{mN}/\text{m})$	13.5
Safety class	A1 (non-toxic, non-flammable, non-corrosive)

To ensure that the operating pressure of the evaporative cooling system remains within a safe range, the system is typically evacuated to a certain negative pressure before the start of an experiment. The initial system pressure is designed to be -30 kPa .

5) Design of the dimensions for experimental result analysis

This chapter presents and analyzes the two-phase flow images in the gas outlet section from two perspectives: variations in thermal load and changes in centrifugal acceleration. Furthermore, to investigate the unique two-phase flow characteristics specific to the evaporative cooling system for the rotor, a comparative experimental platform with a conventional gravity-driven evaporative cooling system is established. By adjusting the pipe diameter of the gas outlet section, the thermal load of the heating section, and the liquid filling volume of the coolant, the two-phase flow images in the gas outlet section are observed and recorded for comparative analysis.

B. Variations in Thermal Load

Fig. 7 shows the two-phase flow images in the gas outlet section under different thermal load conditions when the centrifugal acceleration of the heating section is $a_r = 30g$. Specifically, Fig. 12(a) presents the actual captured images at observation point P1, while Fig. 12(b) displays the images at observation point P2.

It can be seen from the images that when the thermal load is relatively small, the two-phase flow in the gas outlet section is a highly unstable intermittent flow, characterized by alternating gas slugs and liquid slugs with complex and variable flow morphology. Because its characteristics differ to some extent from the various defined intermittent flow patterns in normal gravity environments, this study directly refers to it as intermittent flow without further classification. With the increase of thermal load, the flow transitions to a relatively less unstable churn flow, where bubble breakup intensity and gas-phase content vary over time. Detailed characteristics of these two flow patterns are summarized in Table IV.

It should be noted that the instability discussed in this study refers to macro-scale instability. The intermittent flow, with its periodic alternation of gas and liquid slugs, induces significant global pressure and flow rate oscillations, which can impose substantial mechanical stress on the pipe system—hence it is classified as strongly unstable. In contrast, although the churn flow exhibits highly fragmented gas-liquid interfaces (indicating intense micro-scale turbulence), its dispersed energy distribution results in weaker system impact, justifying its classification as “weakly unstable”.

C. Changes in Centrifugal Acceleration

Table V summarizes the two-phase flow patterns of the system under different operating conditions. It can be seen that when the centrifugal acceleration is relatively low, the system is more likely to achieve a relatively stable flow state. As the centrifugal acceleration increases, the pressure gradient

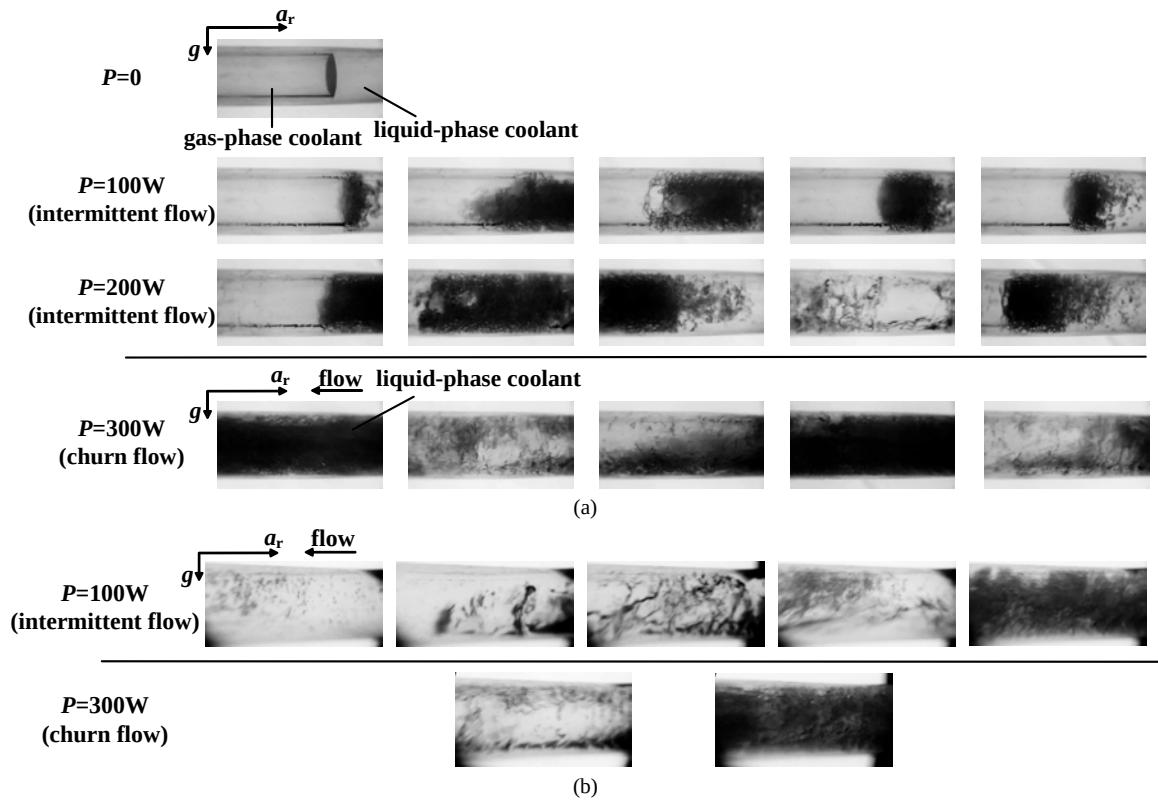


Fig. 12. Two-phase flow images when the centrifugal acceleration of the heating section is $a_r = 30g$. (a) Images of observation point P1. (b) Images of observation point P2.

TABLE IV
CHARACTERISTICS OF INTERMITTENT FLOW AND CHURN FLOW

Flow pattern	Characteristics
Intermittent flow	1) Exhibits high instability. 2) Alternating gas slugs and liquid slugs with weak periodicity (distinctly less regular than slug or plug flows). 3) Gas slugs occupy nearly the full pipe diameter with irregular interfaces, while liquid slugs may contain small bubbles and exhibit sudden velocity surges.
Churn flow	1) Demonstrates comparatively lower instability. 2) Vigorous gas-liquid mixing without stable Taylor bubbles. 3) Large bubbles fragment into smaller ones, creating chaotic flow patterns.

TABLE V
TWO-PHASE FLOW PATTERNS OF THE SYSTEM UNDER DIFFERENT OPERATING CONDITIONS

$a_r (g)$	Flow patterns				
	$P = 100\text{ W}$	$P = 200\text{ W}$	$P = 300\text{ W}$	$P = 400\text{ W}$	$P = 500\text{ W}$
30	Intermittent flow	Intermittent flow	Churn flow	Churn flow	Churn flow
35	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow	Churn flow
40	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow	Churn flow
50	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow
60	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow	Intermittent flow

along the gas outlet pipe (radial direction) rises significantly, and the system pressure and the boiling point of the working fluid also increase (with the exception of the condensing pressure: Since the condenser is installed inside the rotating shaft, the centrifugal acceleration it experiences is negligible, so the condensation pressure does not vary with rotational speed), further intensifying the two-phase flow instability of the system.

Taking the pipe inlet of the gas outlet section at $P = 300\text{ W}$

as an example, the pressure at this location and the boiling point temperature of the coolant selected in this study are shown in Fig. 13. At this condition, the condensing pressure inside the condenser is approximately -18 kPa and hardly changes with centrifugal acceleration. At this condition, the condensing pressure inside the condenser is approximately -18 kPa . Linear fitting of the experimental data shown in Fig. 13 yields the following correlation:

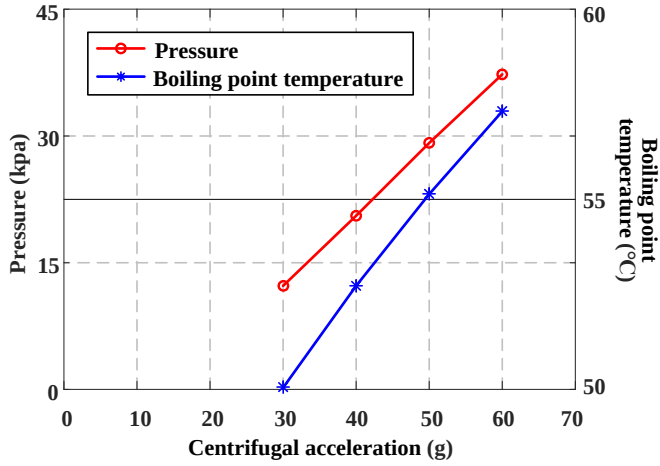


Fig. 13. Pressure and boiling point temperature of the pipe inlet of the gas outlet section when $P = 300$ W.

$$P_{in} = 0.8325G - 12.799 \quad (4)$$

where P_{in} is the pressure at the inlet of the gas outlet section (kPa) in this experiment; G is the multiple of gravitational acceleration (dimensionless); the goodness of fit is $R^2 = 0.9998$, indicating a very strong linear relationship between the two variables. The corresponding boiling point temperature under a given pressure can be obtained from the thermophysical properties of the working fluid, as shown on the right vertical axis of Fig. 13.

The physical mechanism by which an increase in centrifugal acceleration leads to a rise in system pressure originates from the large radial pressure gradient established under a strong centrifugal force field. Under this gradient, the liquid phase working fluid is effectively “compressed”, resulting in an increase in the overall static pressure of the system. The pressure rise directly increases the saturation temperature of the working fluid, reducing the superheat for boiling under the same thermal load and suppressing bubble nucleation. However, once boiling occurs, the high centrifugal force field significantly enhances the local Weber number by increasing the velocity difference between the gas and liquid phases, thereby intensifying the breakup of large bubbles into smaller ones. The competition between “nucleation suppression” and “breakup enhancement” ultimately manifests as an overall increase in system pressure and intensified flow instability.

Furthermore, as the pressure gradient increases, the flash evaporation intensity also increases. The driving force for flash evaporation comes from the liquid superheat ΔT_{sup} caused by a sudden pressure drop:

$$\Delta T_{sup} = T_L - T_{sat}(P) \quad (5)$$

where T_L is the working fluid temperature and T_{sat} is the saturation temperature. The larger the pressure gradient dP/dx , the higher the rate of decrease of saturation temperature per unit distance, $dT_{sat}/dx = (dP/dx) \cdot (dT_{sat}/dP)$, and the instantaneous superheat obtained by the liquid increases accordingly, exacerbating the thermodynamic non-equilibrium. Meanwhile, according to classical nucleation theory, the nucleation rate J satisfies the relationship:

$$J \propto \exp\left(-\frac{\Delta G_{crit}}{k_B T}\right) \quad (6)$$

In the equation, ΔG_{crit} is the critical free energy barrier, i.e., the energy barrier that must be overcome for a bubble nucleus to form and grow stably in the liquid; k_B is the Boltzmann constant (a physical constant); T is the absolute temperature (thermodynamic temperature). The critical free energy barrier decreases as the pressure gradient increases, further causing the nucleation rate J to rise exponentially. The enhancement of flash evaporation directly aggravates the two-phase flow instability of the system. This is mainly manifested by a sudden increase in local void fraction due to the generation of a large amount of vapor in an extremely short time, forming alternating high- and low-void regions in the flow channel, which induces pressure pulsations and flow oscillations.

Furthermore, under a high centrifugal force field, the small bubbles generated by flash evaporation also interact with the breakup process of original large bubbles, making the two-phase flow behavior more complex and significantly worsening the instability.

D. Comparison of Flow Pattern Characteristics with the Conventional Gravity-driven Evaporative Cooling System (a Qualitative Reference Discussion)

In conventional gravity-driven evaporative cooling systems, the random mixing of gas-liquid two-phase coolant is suppressed by surface tension and wall effects in narrow pipes. As a result, it is difficult to form churn flow in the pipe of gas outlet section with an inner diameter of 8 mm. In contrast, in wider pipes, turbulence and inertial forces disrupt the stability of the gas-liquid interface, facilitating the formation of churn flow. Fig. 14 records the two-phase flow patterns in the pipe of gas outlet section under different pipe diameters, liquid filling volume of the coolant, and thermal loads in the conventional gravity-driven evaporative cooling system.

In comparison, in an evaporative cooling system for rotor under a strong centrifugal force field, the two-phase flow in the pipe of gas outlet section exhibits stronger turbulent characteristics, and bubbles are more prone to break up. Therefore, even in pipe of gas outlet section with smaller diameters, churn flow can be observed.

E. Characteristics and Laws of Two-phase Flow in the Pipe-cooled Evaporative Cooling System for a Rotor

Based on the two-phase flow images captured from the pipe of gas outlet section of the pip-cooled evaporative cooling system for a rotor, the characteristics and patterns of the two-phase flow are analyzed, leading to the following conclusions:

1) Two-phase flow patterns

Within the operating range of a heat loads of 100–500 W and a centrifugal acceleration of 30–60g in the pipe of heating section, the two-phase flow in the evaporative cooling system for rotor exhibited significant instability. Two primary flow patterns are observed: intermittent flow and churn flow.

2) Influence of thermal load

As the Thermal load increased, the instability of the two-phase flow decreased, and the flow pattern transitioned

from highly unstable intermittent flow to relatively stable churn flow.

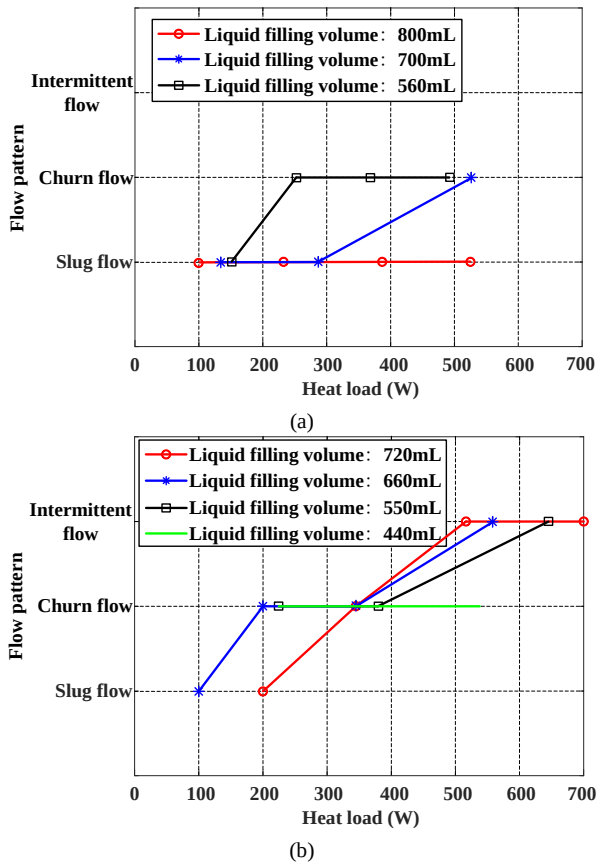


Fig. 14. Flow pattern statistics of the conventional gravity-driven evaporative cooling system. (a) Pipe inner diameter of the gas outlet section is 8 mm. (b) Pipe inner diameter of the gas outlet section is 16 mm.

3) Influence of centrifugal acceleration

With increasing centrifugal acceleration, the system pressure rose significantly, accompanied by an increase in the boiling point temperature of the coolant and the pressure drop gradient along the pipe of gas outlet section. These changes intensified the instability of the two-phase flow.

4) Differences from conventional gravity-driven evaporative cooling system

Under strong centrifugal force, the flow in the evaporative cooling system for rotor became more turbulent and complex, with bubbles more likely to break and churn flow more likely to form compared to conventional gravity-driven evaporative cooling system.

V. CONCLUSION AND OUTLOOK

A. Conclusion of the Research

To clarify the two-phase flow characteristics of the evaporative cooling system for rotor, this study developed a real-time image acquisition system based on a self-built experimental platform of pip-cooled evaporative cooling system for a rotor. Based on these platforms, experimental studies are conducted with heat loads ranging from 100 to 500 W and the centrifugal accelerations of the pipe of heating section ranging from 30 to 60g. The two-phase flow patterns

in the pipe of gas outlet section are observed, collected, and organized, and the pressure data at the inlet of the pipe of gas outlet section are recorded. Meanwhile, a comparative analysis is performed with the flow pattern characteristics of a conventional gravity-driven evaporation cooling system. The results indicate that under the influence of a strong centrifugal force field, the two-phase flow in the system exhibits stronger turbulent characteristics, with bubbles more likely to break up and form a churn flow. An increase in thermal load can reduce the instability of the two-phase flow in the system, while an increase in centrifugal acceleration raises the system pressure and exacerbates the instability of the two-phase flow.

B. Outlook

As an initial experimental investigation into the two-phase flow characteristics of a rotor evaporative cooling system, this work is still at an exploratory stage. Future efforts are warranted in the following aspects.

First, develop methods for measuring two-phase flow parameters in a self-circulating evaporative cooling system operating under high-speed rotation and confined space conditions. Currently, key parameters such as the circulation flow rate, phase fraction, and continuous pressure distribution inside the evaporative cooling system under such conditions are difficult to measure directly, which severely limits quantitative understanding of the flow and heat transfer mechanisms. The development of non-invasive measurement techniques (e.g., visual image-based measurement using computer vision technology) is a prerequisite for obtaining reliable experimental data.

Second, on the basis of the above data, explore fundamental theories of two-phase flow and heat transfer under ultra-gravity conditions of the order of 100g. The centrifugal acceleration generated by a high-speed rotor can reach hundreds of times gravitational acceleration. Whether existing two-phase flow theories under normal gravity remain applicable in this context is still unclear. Future systematic experiments can investigate the effects of ultra-gravity on flow pattern transitions, interphase forces, bubble behavior, and boiling heat transfer, gradually building a theoretical framework suitable for this scenario.

Third, based on theoretical developments, carry out research on simulation and modeling methods. Reliable simulation tools are of great significance for the design of cooling systems in practical engineering products, but they require appropriate constitutive relationships and validation data. Future work can develop numerical simulation methods suitable for rotating evaporative cooling systems based on experimentally refined theoretical models and validate them against experimental results.

Fourth, extend from a single-branch to a multi-branch parallel system. This study adopted a single-branch experimental platform to exclude mutual interference between branches, whereas practical rotor cooling systems mostly adopt multi-branch parallel configurations. Future research, based on a clear understanding of the single-branch mechanism, can investigate issues such as flow distribution,

pressure coupling, and flow instability among parallel branches.

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