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ARTICLE





# Numerical Study on Natural Circulation System under Various Cooling Mediums

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**ABSTRACT:** Aiming at the global design issue of transpiration cooling thermal protection system, a self-driven circulation loop is proposed as the internal coolant flow passage for the transpiration cooling structure to achieve adaptive cooling. To enhance the universality of this internal cooling pipe design and facilitate its application, numerical studies are conducted on this system with four commonly used cooling mediums as coolant. Firstly, the accuracy of the numerical method is verified through an established experimental platform. Then, transient numerical simulations are performed on the flow states of different cooling mediums in the new self-circulation system. Based on the numerical result, the flow, phase change, and heat transfer characteristics of different cooling mediums are analyzed. Differences in fluid velocity and latent heat of phase change result in significant variation in heat exchange capacity among different cooling mediums, with the maximum difference reaching up to 3 times. Besides, faster circulation speed leads to greater heat transfer capacity, with a maximum of 7600 W/m<sup>2</sup>. Consequently, the operating mechanism and cooling laws of the natural circulation system is further investigated, providing a reference for the practical application of this system.

KEYWORDS: Transpiration cooling; natural circulation loop; phase change; heat transfer capability; flow state

## **1** Introduction

As the flight speed and cruise time of aircraft significantly increase, the thermal environment faced by critical components, especially the leading edge of aircraft, becomes increasingly harsh [1,2]. There is an urgent need to develop more efficient and lightweight active cooling technologies [3]. Transpiration cooling with porous materials as the basis material could achieve good cooling effects while consuming less coolant, making it extremely potential for application in high speed aircraft [4,5]. During the application of transpiration cooling, the uneven distribution of thermal loads across the hot-end components can result in localized areas experiencing either inadequate or excessive cooling [6]. This imbalance may subsequently lead to cooling failure in some regions or significant wastage of coolant in others [7,8]. To improve the steadiness of cooling system, it is necessary to design transpiration cooling into engineering applications, it is crucial to consider the internal cooling requirement of the aircraft, fully utilizing the heat sinks of the coolant and further reducing the weight of the cooling system [10,11]. Therefore, the heat transfer between the transpiration cooling structure and the substructure, as well as design schemes for the arrangement of internal coolant pipes should be considered [12–15] to achieve comprehensive thermal management.



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The Natural Circulation System (NCS) capitalizes on the density variation between cooling mediums at the heat source and radiator to facilitate the circulation of single-phase or two-phase fluids within the system [16–20]. Given its requirement of no pumps or additional driving equipment, the NCS boasts a straightforward setup and finds applications across diverse domains, including nuclear reactor cooling, electronic cooling, and solar energy conversion [21–24]. The design and implementation of natural circulation loops hinge significantly on their startup characteristics and flow instability [25,26], garnering considerable attention. Furci et al. [27] introduced a simplified two-phase flow model to examine the transient thermal-hydraulic behavior of a loop. Li et al. [28], meanwhile, conducted numerical research on the combined impact of system oscillations on transient response traits using an enhanced RELAP5 code. Nevertheless, the transient response traits of circulation loops with varying structural setups and cooling mediums remain incompletely comprehended and necessitate further exploration.

A novel open natural circulation system, inspired by the design concept of a low-pressure open NCS, is introduced. It comprises solely an open reservoir and U-shaped piping. Unlike other open natural circulation systems, all piping in this system is enveloped by heat sources, albeit with varying heat flux along different segments, aligning more closely with the heat exposure patterns of aircraft internal structures. During operation, the heat flux difference results in density variation of fluid in different parts of the pipe, driving the fluid to circulate within the loop for cooling purposes. Simultaneously, the liquid coolant undergoes a phase change by absorbing heat and transforms into a gaseous state, and the gaseous coolant flows out from the pipe orifice to be supplied as a coolant for upper-layer transpiration cooling. Consequently, this design not only ensures the safe operation of components with lower temperature tolerance by cooling internal structure but also circumvents issues associated with direct transpiration cooling of liquid coolant in porous media. Additionally, it eliminates the need for driving equipment, reducing the additional load on the aircraft.

However, previous work only conducted qualitative experimental research on the new natural circulation system using liquid water as the coolant, lacking quantitative analysis of the flow state and heat transfer capacity. To enhance the universality of the system design and promote its application, multiple common coolants are selected to investigate the flow and heat transfer characteristics of the system. Firstly, an experimental platform was built to verify the accuracy of the numerical method. Then, Subsequently, a novel direct computational fluid dynamics simulation approach was developed by integrating a customcoded program into commercial software. Finally, through the numerical simulations of the circulation loop transient performance, the startup characteristics and flow states of different cooling mediums were intuitively obtained. The circulation capacity and cooling effects of different coolants were comparatively analyzed, providing a reference for the design of internal flow passages for transpiration cooling.

#### 2 Physical Model

The schematic representation of the natural circulation system is illustrated in Fig. 1. Additionally, the three-dimensional configuration is depicted in Fig. 2. This system primarily comprises a rectangular water tank, two open pipe orifices, and U-shaped piping. The two pipe orifices ensure the continuity of injected liquid coolant and the continuous delivery of steam. The water tank dimensions are  $L 500 \times W 80 \times H 100 \text{ mm}^3$ , directly connected to the atmospheric environment, allowing the system to operate under ambient pressure. The U-shaped piping has a circular cross-section with an internal diameter of  $\Phi = 10 \text{ mm}$  and a solid pipe wall (lightweight aluminum alloy) thickness of  $\delta = 1 \text{ mm}$ . The entire piping consists of five parts: two vertical pipes ( $l_1$ ,  $l_5$ ), a horizontal pipe ( $l_3$ ), and two elbow pipes ( $l_2 \ l_4$ ), where the vertical pipe length is  $l_1 = l_5 = 544 \text{ mm}$  and R = 56 mm, respectively. The U-shaped piping is centrally installed at the bottom of the water tank. Four commonly selected liquid cooling mediums are selected: liquid water, ethanol, fuel oil, and R113.



Figure 1: Schematic diagram of the natural circulation system



Figure 2: Schematic diagram of the U-shaped loop

## 3 Numerical Method

#### 3.1 Mathematical Model

Due to the involvement of phase change in different cooling mediums, using a three-dimensional model for calculations would consume a significant amount of computational resource and time. Therefore, to quickly obtain the flow and heat transfer characteristics of the two-phase fluid within the loop, a two-dimensional model is adopted for transient calculations. Given the significant temperature gradient observed within the pipe, it is crucial to account for both heat and mass transfer in these regions. The entire phase change process is primarily computed using the Volume of Fluid (VOF) multiphase flow model, which could capture the distinct water/air interface in reservoirs and track the generation, flow, and evolution of bubbles in the circuit. Besides, the self-programmed code serving as auxiliary tools to enable mass and energy exchange between liquid and gaseous coolants during phase transitions. The equations used in the fluid domain are detailed as follows:

Conservation of mass [29]:

$$\frac{\partial \alpha_q}{\partial t} + \overrightarrow{v} \nabla \alpha_q = \frac{S_{\alpha_q}}{\rho_q} \sum_{q=1}^n \alpha_q = 1$$
(1)

where the  $\rho_q$  represents fluid density,  $\alpha_q$  represents volume traction.

Mass source term:

$$S = \begin{cases} -r \cdot \alpha_l \rho_l \frac{T - T_s}{T_s}, T \ge T_s \\ r \cdot \alpha_g \rho_g \frac{T_s - T}{T_s}, T < T_s \end{cases}$$
(2)

where the *T* represents temperature, the *r* represents the local stress tensor caused by fluid viscosity.

Momentum conservation [29]:

$$\frac{\partial}{\partial t} \left( \rho \overrightarrow{U} \right) + \nabla \cdot \left( \rho \overrightarrow{U} \overrightarrow{U} \right) = -\nabla p + \nabla \cdot \overline{\overline{\tau}} + \frac{\partial}{\partial x_j} \left( -\rho \overline{u'_i u'_j} \right)$$
(3)

where the U represents the velocity, the  $\overline{\tau}$  represents the local stress tensor caused by fluid viscosity.

SST *k*- $\omega$  turbulence model [30] is used to solve the Reynolds stress term:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_{j}}(\rho k u_{j}) = \frac{\partial}{\partial x_{j}}\left[\left(\mu + \frac{\mu_{t}}{\sigma_{k}}\right)\frac{\partial k}{\partial x_{j}}\right] + G_{k} + G_{b} - \rho\varepsilon - Y_{M} + S_{k}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_{j}}(\rho\varepsilon u_{j}) = \frac{\partial}{\partial x_{j}}\left[\left(\mu + \frac{\mu_{t}}{\sigma_{\varepsilon}}\right)\frac{\partial\varepsilon}{\partial x_{j}}\right] + \rho C_{1}S\varepsilon - \rho C_{2}\frac{\varepsilon^{2}}{k + \sqrt{\nu\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_{b} + S_{\varepsilon}$$
(4)

the specific definitions of each term in the Standard k- $\omega$  turbulence model are found in ANSYS Fluent documents [31].

Energy conservation [29]:

$$\frac{\partial}{\partial t}\left(\rho E\right) + \frac{\partial}{\partial x_{i}}\left[u_{i}\left(\rho E + p\right)\right] = \frac{\partial}{\partial x_{j}}\left[k\frac{\partial T}{\partial x_{j}} + u_{i}\left(\tau_{ij}\right)_{eff}\right] + S_{h}$$
(5)

where the *E* represents the internal energy of the fluid, and *k* represents the heat conductivity of the fluid.

The equation used in the solid domain is mainly the energy equation as follows [32]:

$$\nabla \cdot (k_s \nabla T) = \frac{\partial}{\partial t} \left( \rho E \right) \tag{6}$$

#### 3.2 Parameter Setting

The NCS is driven by the density variation of fluid within different sections of the loop. Therefore, the changes in the physical properties of gas-liquid two-phase fluid, especially liquid fluid, cannot be ignored. Throughout the cooling process, the physical properties of liquid fluid do not vary significantly with pressure changes. Thus, the influence of temperature is mainly considered here. Within the temperature variation range of liquid fluid, enough temperature points are selected, and the density  $\rho$ , dynamic viscosity  $\mu$ , thermal conductivity k, and specific heat capacity  $C_p$  of the liquid fluid at each temperature are calculated using piecewise linear fitting. The data source of physical properties for four mediums are obtained from the NIST, and their properties are shown in Tables 1–4.

$H_2O$	T/K	ρ/kg⋅m⁻³	$\boldsymbol{\mu} \times 10^{3}/kg {\cdot}m^{-1}{\cdot}s^{-1}$	$k/W \cdot m^{-1} \cdot K^{-1}$	$Cp/J \cdot kg^{-1} \cdot K^{-1}$
1	273	999.9	1.788	0.551	4218.3
2	283	999.7	1.354	0.574	4195.4
3	293	998.2	1.004	0.599	4184.2
4	303	995.7	0.802	0.618	4179.9
5	313	992.2	0.653	0.635	4179.4
6	323	988.1	0.549	0.648	4181.3
7	333	983.1	0.470	0.659	4184.9
8	343	977.8	0.406	0.668	4190.0
9	353	971.8	0.355	0.674	4196.6
10	363	965.3	0.312	0.680	4205.1
11	373	958.4	0.283	0.683	2077.3

Table 1: Physical property parameters of H<sub>2</sub>O

Table 2: Physical property parameters of C<sub>2</sub>H<sub>5</sub>OH

C <sub>2</sub> H <sub>5</sub> OH	T/K	$\rho/kg \cdot m^{-3}$	$\boldsymbol{\mu} \times \boldsymbol{10}^{3}/kg \cdot m^{-1} \cdot s^{-1}$	$k/W \cdot m^{-1} \cdot K^{-1}$	$Cp/J \cdot kg^{-1} \cdot K^{-1}$
1	273	805.3	1.55	0.169	2278.6
2	283	797.9	1.46	0.166	2323.5
3	293	790	1.19	0.1645	2396
4	303	780.8	0.983	0.1625	2474.5
5	313	772	0.82	0.16	2558.8
6	323	763	0.689	0.159	2648.7
7	333	754	0.584	0.157	2743.6
8	343	744.6	0499	0.1557	2843.4
9	351.2	736.6	0.447	0.1544	2929

**Table 3:** Physical property parameters of C<sub>19</sub>H<sub>30</sub>

$C_{19}H_{30}$	T/K	ρ/kg⋅m⁻³	$\boldsymbol{\mu} \times \boldsymbol{10}^{3}/kg \cdot m^{-1} \cdot s^{-1}$	$k/W \cdot m^{-1} \cdot K^{-1}$	$Cp/J \cdot kg^{-1} \cdot K^{-1}$
1	273	931.8	8.68	0.11	1457.2
2	303	914	5.09	0.11	1542.3
3	333	896.3	3.19	0.109	1633.2
4	363	878.8	2.11	0.11	1728
5	393	861.4	1.45	0.108	1922.9
6	423	844	1.031	0.107	2024.2
7	453	827.1	0.76	0.107	2141
8	483	812	0.65	0.105	2244
9	513	801.1	0.47	0.103	2361.3
10	523	795.4	0.343	0.101	2422.3

R113	T/K	$\rho/kg \cdot m^{-3}$	$\boldsymbol{\mu} \times 10^{3}/kg \cdot m^{-1} \cdot s^{-1}$	$k/W \cdot m^{-1} \cdot K^{-1}$	$Cp/J \cdot kg^{-1} \cdot K^{-1}$
1	273	1621.9	0.92	0.0778	1621.5
2	283	1598.8	0.79	0.0755	1598.8
3	293	1575	0.69	0.0734	1575
4	303	1551	0.614	0.0712	1551
5	313	1527	0.546	0.0691	1527
6	320.5	1508	0.5	0.0675	1508

Table 4: Physical property parameters of R113

#### 3.3 Initial and Boundary Condition

Prior to heating and startup, the system maintains a static condition, featuring a cooling mediums level of h = 13 mm in the tank and an initial system temperature of  $T_{sur}$ . Given that the tank is directly exposed to the atmospheric environment, the upper surface of the tank is designated as a pressure outlet boundary with an outlet pressure of  $P_{out} = 1$  atm. Upon heating and startup (t > 0 s), a constant heat flux ( $Q_1, Q_2$ ) is employed for the external wall surface of the U-shaped piping.

#### 3.4 Spatial and Temporal Step Sizes

#### 3.4.1 Spatial Step Size

Taking a two-dimensional model of a NCS as the focus of research, unstructured tetrahedral grides are generated for the model. Local grid refinement is implemented in areas near wall surfaces or at curved pipe sections where physical parameters undergo significant variations, as illustrated in Fig. 3a. To verify grid independence, calculations are conducted using three sets of grids with differing initial grid counts. Ensuring the accuracy of calculations is crucial. The height of the first grid layer is determined based on the constraint  $Y + \leq 1$ . There is a balance to be struck between grid size and computational efficiency: smaller grid distances can enhance the precision of the numerical solution but come at the cost of increased computational demands; larger grid distances may compromise precision but offer improved computational speed. Taking these considerations into account, the grid heights of the first layer adjacent to the wall are set at  $\Delta x = 1 \times 10^{-5}$  m (grid 1),  $\Delta x = 2 \times 10^{-5}$  m (grid 2) and  $\Delta x = 5 \times 10^{-5}$  m (grid 3), respectively. The corresponding initial total grid counts are 1,200,000, 800,000, and 400,000, respectively. Under the operating conditions of  $T_{sur} = 284$  K,  $Q_1 = Q_2 = 10$  kW/m<sup>2</sup>, the solid outer wall temperature variation at point M of the U-shaped piping is calculated using the three sets of grides with a consistent time step ( $\Delta t = 0.01$  s), as shown in Fig. 3b. It is evident that the temperature change trends obtained using grid 2 and grid 3 are very similar. Therefore, to optimize both computational time and accuracy, the division strategy of grid 2 is ultimately chosen for subsequent calculations.

#### 3.4.2 Temporal Step Size

In transient VOF calculations, the Courant number plays a crucial role in determining both the precision and the computational efficiency of the results. It quantifies the relationship between the temporal step size and the spatial step size, defined as the product of fluid velocity (*u*) and temporal step size ( $\Delta t$ ) divided by the grid size ( $\Delta x$ ). For a fixed grid configuration, an increased Courant number corresponds to a larger temporal step size, accelerating the progress of the transient calculation. However, this can adversely affect computational stability and lead to larger deviations between the calculated results and the true values, failing to accurately capture the flow details. To ensure a reasonable value of the Courant number, repeated

adjustments are made to the temporal step size based on the fluid velocity. Finally, a temporal step size of  $\Delta t = 1 \times 10^{-2}$  s is selected for the stage without phase change, and during the phase change stage, the temporal step size is adjusted to  $\Delta t = 1 \times 10^{-5}$  to  $1 \times 10^{-4}$  s according to the intensity of fluid velocity changes.



Figure 3: (a) grid division and (b) grid independence verification

## 3.5 Experimental Verification

The study mainly focuses on numerical simulation, hence, the accuracy of the mathematical model and numerical calculation method need to be validated using experiment. Experimental methods and data from previous studies [33] were used for validation, and the experimental device is shown in the Fig. 4a. In the experiment, the model was heated using a heating plate, and the heating plate was split into two fully symmetrical sections, each individually linked to the control unit to guarantee independent heating capacity and temperature regulation for both plates. The transient temperature changes of the pipe wall were monitored and recorded using an infrared thermal imager. A 5.0 W continuous laser and a highspeed camera were employed to record flow of cooling mediums. With the operating condition of  $T_{sur}$  = 284 K and  $Q_1 = Q_2 = 10$  kW/m<sup>2</sup>, the comparison of the trend of pipe wall temperature and the flow states at selected moments between experimental and numerical results is shown in Fig. 4. It can be observed that under the same operating condition, the calculated numerical results are in good agreement with the experimental results in terms of flow state change and temperature change, with a relative deviation of less than 3%. Therefore, it is concluded that the mathematical model and calculation method adopted in this chapter are effective.



**Figure 4:** (a) Experiment system and the trends of pipe wall temperature obtained by simulation and experiment [29], (b) The flow states at selected moments

## **4** Results and Discussions

## 4.1 Heat Transfer Deterioration Phenomena under Symmetrical Heat Flux Condition

To investigate the change in the flow state of different cooling mediums within the loop under constant heat flux condition on the pipe wall, transient numerical simulations are conducted under operating condition of  $T_{sur} = 300$  K and  $Q_1 = Q_2 = 8$  kW/m<sup>2</sup>. Based on previous research findings, the evolution of the flow state of liquid water within the loop under uniform heat flux condition could be divided into three stages: expansion stage, localized circulation stage, and phase change stage. Consistent with the flow state of water, when the heating duration is brief (t < 1 s), resulting in a minimal increase in wall temperature  $(\Delta T < 2 \text{ K})$ , the fluid inside the pipe absorbs heat from the wall, causing a slight reduction in its density. During this initial phase, the fluid expands outwards along the pipeline. Notably, the fluid velocity close to the pipe wall is notably higher than that at the center. Specifically, across the pipe's cross-section in the radial direction, fluid velocity increases from the boundary layer and subsequently decreases towards the pipe's center. This velocity gradient arises due to the rapid temperature rise in the fluid adjacent to the pipe wall during the short heating period. Different cooling media exhibit varying temperature rise rates, which influence the duration of the expansion phase.

Fig. 5 displays the pressure, temperature, velocity, and phase distribution of four cooling mediums at t = 1 s. Among them, the R113 has the highest pressure within the pipe, exceeding 6000 Pa in the horizontal segment pipe, followed by the water, while the ethanol has the lowest pressure within the pipe, about 5000 Pa. This is related to the fluid's density and heat absorption capacity. During the initial heating stage, the temperature rise of the fluid is small, resulting in insignificant changes in its density. Therefore, the greater density of the fluid, the higher the pressure observed in the horizontal segment of the pipeline. At this point, the liquid water is still in an expansion state. The three additional cooling media, ethanol, fuel oil, and R113 demonstrate increasing axial and radial temperature differences inside the pipe as a result of their rapid temperature increase and significant density variations with rising temperature. Gravity and buoyancy working together create a localized circulatory flow inside the vertical pipe, where fluid descends close to the center and ascends near the wall. This circulation initiates in the pipe's midsection and gradually spreads towards both the orifice and the elbow over time. As a result, the turbulence intensity in the fluid flow surrounding the orifice is increased. The specific heat capacity of R113 is less than half that of water and thus experiences the fastest temperature rise, combined with low fluid viscosity, resulting in the highest local circulation speed. The fuel oil does not form a circulation in the elbow and the bottom horizontal pipe due to its greater flow resistance, for its viscosity is 5-8 times higher than other cooling mediums at low temperatures. Additionally, the elbow increases the turbulence level of the fluid within the pipe, resulting in irregular flow. During this stage, the flow velocities of the four cooling mediums are very slow, ranging from  $1 \times 10^{-4}$  to  $1 \times 10^{-6}$  m/s, with the maximum velocity occurring near the vertical pipe orifice. In summary, due to differences in temperature rise rates and viscosities among different cooling mediums, the time cycles for the evolution of their flow states exhibit slight differences.

As the temperature of the solid wall further increases, Fig. 6 depicts the velocity streamline diagrams and the flow law of four cooling mediums within the pipes at t = 10 s, respectively. Fig. 7 displays the heat flux distribution on the pipe wall at the same time. At t = 10 s, the pressure distributions of the four cooling mediums undergo minimal changes, and the temperature increase is also insignificant. At this point, it can be observed that liquid water has entered a localized circulation state. Liquid water has high thermal conductivity, and its flow speed is accelerated with the occurrence of localized circulation, so it exhibits strong heat exchange capability with the wall. Consequently, the wall heat transfer rate reaches up to 6100 W/m<sup>2</sup>. Ethanol flows rapidly at the pipe orifice and remains in a localized circulation state. This is because as the fluid temperature within the pipe rises, the fluid exits the orifice and exchanges with fluid in the reservoir. However, within the vertical pipe, the flow velocity is lower, and most of the fluid flows in the opposite direction, resulting in an overall wall heat flux that is only 1/3 of that of liquid water. The fuel oil experiences a rapid temperature rise because of the smaller specific heat capacity. Due to its high viscosity and significant flow resistance, the buoyancy force generated is insufficient to overcome the shear force in the boundary layer. Therefore, it exhibits irregular flow within the pipe. Furthermore, the fuel oil has low thermal conductivity and low overall flow velocity, with the overall flow velocity below 0.001 m/s except at the elbow. Consequently, the heat exchange capacity between the fuel oil and the solid wall is relatively weak, showing a wall heat transfer rate of 2800 W/m<sup>2</sup>. The R113 has rapid temperature increase and low viscosity, experiencing low flow

resistance and showing an overall flow velocity less than 0.0025 m/s. Moreover, the thermal conductivity of the R113 is only about 1/10 of that of liquid water. Therefore, there exists localized circulation with greater turbulence intensity within the pipe, enhancing its heat exchange capacity. The overall wall heat transfer rate reaches 4300 W/m<sup>2</sup>. In summary, during the initial heating stage, the viscosity and thermal conductivity of the fluid affect the duration of localized circulation, which in turn increases the flow velocity and enhances the heat exchange between the cooling medium and the solid wall.



**Figure 5:** t = 1 s (a) pressure, (b) temperature, (c) velocity and (d) phase distribution of four cooling mediums (water (C1), ethanol (C2), fuel oil (C3) and R113 (C4) from left to right)



Figure 6: t = 10 s velocity distribution of four cooling mediums (a) water, (b) ethanol, (c) fuel oil and (d) R113



**Figure 7:** t = 10 s heat flux distribution on the right pipe wall of the four cooling mediums

Fig. 8 presents the pressure, temperature, velocity, and phase distribution of four fluids within pipes at t = 60 s. As the fluid temperature further increases, the pressure of all four cooling mediums decreases, with the maximum drop reaching 1000 Pa. R113 experiences the fastest temperature rise and reaches the highest temperature, approximately 320 K. Liquid water, due to its high specific heat capacity, experiences the slowest temperature rise and remains the lowest in temperature. Additionally, Fig. 8b displays the temperature distribution, all four cooling mediums exhibit lower temperatures in part of the fluid at the elbow, which caused by higher turbulence dissipation at these locations. During this stage, due to its slow temperature rise, liquid water continues to flow in localized circulation within the pipe, and its flow velocity is relatively

fast due to the presence of temperature difference. R113 has a relatively low phase transition temperature and a rapid temperature rise and enters the initial phase transition stage, generating tiny bubbles. This further reduces the fluid's density and viscosity, resulting in the fastest flow velocity, with a center velocity exceeding 0.01 m/s. As the bubbles rise, the fluid flows uniformly towards the pipe orifice. Ethanol, the specific heat is half that of water, and the thermal conductivity is more than twice that of water, resulting in fast temperature rise and small temperature difference. Hence, the density difference formed by the temperature difference is not enough to drive the local circulation. Additionally, the fluid density further decreases upon heating because of its lower pressure within the pipe. At the moment depicted in Fig. 8c, the inflow of the cooling medium from the reservoir results in the fluid within the vertical pipe flowing in the opposite direction. In the next moment, as the fluid within the pipe expands, the overall flow within the vertical pipe shifts towards the pipe orifice, exhibiting irregular overall flow. The fuel oil has high viscosity, and thus remains in an extremely irregular flow state, with high turbulence intensity and relatively fast flow velocity at the elbow pipe. As the heating time progresses, the velocities of all four cooling mediums increase, and the maximum velocity reaches  $1 \times 10^{-2}$  m/s. In summary, as temperature rises, the thermophysical properties of cooling mediums have a significant impact on the changes in fluid flow states.



**Figure 8:** t = 60 s (a) pressure, (b) temperature and (c) velocity of four cooling mediums (water (C1), ethanol (C2), fuel (C3) and R113 (C4) from left to right)

Due to the differences in phase change temperatures and temperature rise rates among the four cooling mediums, the time required for each to reach its phase change state is different. The latent heat of phase change and the specific time points of phase change are shown in Table 5. Once the four liquid cooling media reach their respective phase change temperatures, small bubbles initially emerge in the center of the vertical pipe, slowly expanding and coalescing as time passes. These bubbles then swiftly ascend and exit through the pipe orifice due to buoyancy. Fig. 9 illustrates the pressure, temperature, velocity, and phase distribution of the fluids within the pipe 30 s after each of the four cooling media undergoes phase change. The presence of an elbow in the pipe results in complex flow patterns and high turbulence intensity within the horizontal pipe segment. Consequently, despite the upper wall of the horizontal pipe experiencing the quickest temperature increase, bubbles first become apparent in the middle of the vertical pipe because of the relatively lower turbulence levels present. Furthermore, during the liquid-to-gas phase transition, the fluid undergoes rapid expansion, leading to a significant increase in flow velocity, which can exceed  $10^{-1}$  m/s. Observing the velocity vector diagrams at different locations within the pipes of the four cooling mediums reveals that the velocity changes are relatively drastic at the locations where bubbles form, indicating that the fluid motion intensifies due to the phase change of the liquid. The fuel oil has a higher phase change temperature and undergoes the most intense phase change, resulting in a significant reduction in viscosity, and the local fluid velocity of fuel oil reaches as high as 1 m/s. Ethanol has lower density and viscosity, and hence has the second-fastest flow rate after fuel oil, which local fluid velocity reaches to 0.4 m/s. R113 has lowest phase change temperature and highest fluid density, leading to the slowest flow rate, about 0.2 m/s. In summary, the intensity of the phase change is closely related to the fluid's phase change temperature. The higher the phase change temperature, the greater the flow velocity and the more intense the heat transfer. Moreover, in the two-phase state, the viscosity of the liquid fluid has a minor impact on the flow velocity.

<b>Cooling mediums</b>	$H_2O$	$C_2H_5OH$	Oil	R113
Phase change temperature/K	373	351.24	523	320.5
Time/s	180.1	90.4	278.3	51
Latent heat of phase change/kJ·kg <sup>-1</sup>	2258	849.9	374	144

Table 5: Time points of phase transition of cooling mediums

To investigate the heat transfer between the fluid inside the pipe and the solid wall at this point, Fig. 10 presents the distribution of fluid-solid heat transfer and volume fraction of liquid cooling mediums in the right side of vertical pipe. When the liquid cooling medium volume fraction ranges from 0 to 1, the liquid cooling medium undergoes a phase change during this period, releasing substantial latent heat of phase change and enhancing heat exchange with the solid wall. Hence, the maximum wall heat transfer rate for water, ethanol, fuel oil and R113 can reach approximately 11, 12.5, 30, and 10 kW/m<sup>2</sup>, respectively. The magnitude of wall heat transfer is related to the fluid's temperature, velocity, and latent heat of phase change. Fuel oil has the highest phase change temperature and the fastest flow velocity. Although the phase change latent heat of fuel oil is only 374 kJ/kg, its wall heat transfer rate is still the highest. Ethanol has the second-fastest flow velocity and the second-highest heat transfer rate about 7500 W/m<sup>2</sup>, but due to the significantly lower phase change temperature, its wall heat transfer rate is far behind fuel oil. This indicates that flow velocity is one of the main factors influencing heat transfer rate. R113 has the slowest flow velocity and extremely low latent heat of phase change, resulting in the lowest wall heat transfer rate. Although the latent heat of phase change of liquid water is 10 times that of fuel oil, its wall heat transfer rate is not prominent among the four cooling mediums due to differences in phase change temperature and flow velocity. When the liquid cooling medium volume fraction drops to 0 and the wall is completely occupied by the vapor

region, the fluid-solid heat transfer rate drops sharply. At this point, the external heat absorbed by the solid wall (*Q*) will be much greater than the heat absorbed and carried away by the internal cooling medium from the solid wall. This results in a rapid increase in the local wall temperature, leading to localized overheating. Moreover, due to the pipe orifice's connection to the reservoir, the fluid temperature, enhanced by bubble expansion, is greater than the wall temperature at the orifice. Consequently, the heat transfer rate between the fluid and the solid near the outlet (-0.1 < Y < 0 m) drops drastically to a negative value, signifying a shift in heat transfer direction from the fluid to the solid. However, due to the small difference in specific heat capacity between the other three liquid cooling mediums and the quartz glass pipe wall, the temperature difference between the fluid and the solid wall during heating is not significant, and therefore no obvious heat transfer deterioration occurs at the pipe orifice. In summary, the influence of fluid velocity on the heat transfer rate between the fluid and the solid wall is greater than that of the fluid's latent heat of phase change, and by reasonably regulating the thermophysical properties of the fluid, some instances of heat transfer deterioration can be avoided.



**Figure 9:** (a) pressure, (b) temperature, (c) velocity and (d) phase distribution of four fluid cooling mediums (water (C1), ethanol (C2), fuel oil (C3) and R113 (C4) from left to right) 30 s after phase transformation



**Figure 10:** Distribution of heat flux on the inner wall of the vertical pipe and liquid volume fraction 30 s after phase change: (a) water, (b) ethanol, (c) fuel oil, and (d) R113

#### 4.2 Circulation Flow Phenomena under Asymmetric Heat Flux Condition

To address the issues of heat transfer deterioration and localized temperature rise caused by bubble blockage under uniform heat flux condition, an investigation is conducted on asymmetric heat flux condition, i.e., enhancing the heat flux on the right-side  $l_2$  pipe wall. Transient calculations are performed under the condition of  $T_{sur} = 300$  K,  $Q_1 = 8$  kW/m<sup>2</sup>, and  $Q_2 = 8.01$  kW/m<sup>2</sup>. Fig. 11 illustrates the wall heat flux on the right-side pipe wall of four cooling mediums after 10 s of heating. The figure reveals that the relative strength of heat transfer among the four cooling mediums remains unchanged. However, the difference of heat flux between the pipes on both sides causes a density difference in the fluid, which subsequently forms a pressure difference, increasing the fluid flow velocity. Consequently, the heat transfer rate of all four cooling mediums at this point is greater than that when the heat flux is uniform on both sides of the pipe, with a maximum increase of 1100 W/m<sup>2</sup>. Fig. 12 depicts the velocity vector diagrams of the four cooling mediums within the vertical pipe at 30 s of heating. Due to the pressure difference between the pipes on both sides, the heat exchange between hot cooling medium within the pipe and the cold cooling medium in the reservoir is enhanced. Therefore, the temperature of cooling mediums within the pipe rises slowly. Liquid water and R113 exhibit varying degrees of localized circulation. The R113 has low viscosity and significant density change with temperature, and hence experiences violent localized circulation within the pipe, with faster flow velocity

and higher wall heat transfer rate up to 3000 W/m<sup>2</sup>. Ethanol has rapid temperature rise rate, low density, and significant density change with the increase of temperature, and thus the pressure difference arising from density difference can overcome its own gravity, forming a counterclockwise overall circulation. The fuel oil has fast temperature rise rate and high viscosity, and thus does not exhibit localized circulation. The fluid within pipes on both sides expands towards pipe orifice, and flow velocity in the right pipe is twice as fast as that in the left. However, due to the small difference in heat flux between the pipes on both sides, the pressure difference is insufficient to overcome the force of gravity, preventing the formation of an overall circulation within the pipe. In summary, different fluid requires different heat flux differences to drive overall circulation due to their distinct densities and magnitude of the density change with the increase of temperature.



Figure 11: t = 10 s heat flow distribution on the wall of four kinds of cooling mediums

When the fluids are heated to the phase change temperature, the distributions of fluid-solid heat transfer and liquid volume fraction on the inner wall of the right vertical pipe of the four cooling mediums are shown in Fig. 13. Due to the presence of a heat flux difference and an increase in heat flux, the times for liquid water, ethanol, fuel oil, and R113 to reach their phase change points advance to 160, 79, 235, and 39 s, respectively. Meanwhile, the existence of the heat flux difference enhances the flow velocities of all four cooling mediums, resulting in higher heat transfer rates during the phase change state compared to uniform heat flux condition. Hence, the maximum wall heat transfer rate for water, ethanol, fuel oil, and R113 can reach 12, 17, 31, and 13  $kW/m^2$ , respectively. Furthermore, the phase changes of the four cooling mediums become more intense, with a relative decrease in the liquid volume fraction and a narrowing of the heat flux fluctuation range. Additionally, the heat exchange between the fluid in the reservoir and the fluid at the pipe orifice increases due to the pressure difference, mitigating the heat transfer deterioration at the pipe orifice. Therefore, even though the magnitude of the heat flux difference is insufficient to drive an overall circulation of the fluid, it can still enhance fluid-solid heat transfer and reduce the phenomenon of heat transfer deterioration.



Figure 12: t = 30 s velocity vector diagrams of four cooling mediums (a) water, (b) ethanol, (c) fuel oil and (d) R113

To facilitate the directional circulation of the cooling medium solely through the pressure differential on both sides, a heat flux disparity of 100  $W/m^2$  arises between the fluids in the pipes on both sides. Throughout the heating process, the evolution of the four cooling mediums can be categorized into three phases: liquid expansion, localized flow, and comprehensive circulation. Before any phase transition occurs in the fluid, the temperature divergence of fluids in the vertical pipes on either side steadily escalates with the duration of heating. Impelled by this pressure differential, the cooling medium establishes a comprehensive counterclockwise flow within the loop, transitioning into the comprehensive circulation phase, as depicted in Fig. 14. During this phase, the temperature and density contrast, axial velocity of the fluids in the vertical pipes on both sides attain their peak values, with the axial flow velocity surging to approximately 1.1 m/s. The density contrast  $(\Delta \rho)$  between the fluids on both sides functions as the primary impetus for this comprehensive circulation. The essential role of this driving force is to alter the fluid flow in the opposite direction (clockwise) to the overall flow direction during the localized circulation stage, namely the upward flow near the left vertical pipe wall and the downward flow at the center of the right vertical pipe. Fig. 15 illustrates the fluid flow direction transition within the right vertical pipe as an example. The longer the duration of the localized circulation stage, the greater the maximum Y-direction velocity v of the local flow opposite to the overall flow direction, which means a larger external force is required to overcome this

flow direction. Furthermore, the higher the fluid density, the greater the buoyancy force required for overall circulation, and hence the larger density difference needed. Ethanol has a low density and shorter localized circulation time, which facilitates the formation of overall circulation. Fuel oil exhibits earlier overall circulation when the heat flux difference is sufficiently large due to its high viscosity and short localized circulation time. This shortens the duration of the localized circulation stage and correspondingly reduces the local reverse flow velocity v, thus requiring a smaller driving force  $\Delta \rho$  for overall circulation. Fig. 16 shows the wall heat flux distribution of the four cooling mediums at this time. After fuel oil enters overall circulation, the flow velocity increases and the wall heat transfer rate rises to 7600 W/m<sup>2</sup>. Ethanol has a long overall circulation time and high fluid velocity, and thus its wall heat transfer rate also increases to a level similar to that of fuel oil, about 7500 W/m<sup>2</sup>. The wall heat transfer rate of water increases to 7000 W/m<sup>2</sup> for its high thermal conductivity and specific heat capacity. Additionally, the wall heat transfer rate of R113 significantly drops to 5150 W/m<sup>2</sup> due to the shorter duration of overall circulation. At this time, fluid flow is mainly driven by pressure difference, and viscosity has a minor impact on liquid flow velocity. It can be concluded that overall circulation significantly increases fluid flow velocity and enhances fluid-solid heat transfer in the loop.



**Figure 13:** Distribution of heat flux density on the inner wall of the vertical pipe and liquid volume fraction 30 s after phase change: (a) water, (b) ethanol, (c) fuel oil, and (d) R113



Figure 14: Velocity vector diagrams of four cooling mediums (a) water, (b) ethanol, (c) fuel oil, and (d) R113



Figure 15: Schematic diagram of the fluid flow direction transition in the right pipe

To investigate the heat transfer efficiency of the fluid during the phase change circulation phase, additional heat is applied to the system. As the pipe wall's temperature approaches the fluid's boiling point, the cooling mediums undergo phase change at high-temperature regions while simultaneously flowing with the circulation, resulting in no significant bubble formation. However, thereafter,  $T_{max}$  does not show a significant increase. After reaching the phase change point 30 s of four cooling mediums, the heat flux on the right pipe wall surface is as illustrated in Fig. 17. Due to the high phase change temperature and long

heating duration results in a large pressure difference and rapid flow rate, the wall heat flux of the fuel oil remains the highest, with an average of 15,000 W/m<sup>2</sup>. Furthermore, the emergence of multiple tiny bubbles close to the exit of the right vertical pipe swiftly increases the density disparity ( $\Delta \rho$ ) of the fluids on both sides, resulting in a notable acceleration of the fluid's overall circulation rate. The bubbles quickly flow out of the pipe orifice, preventing bubble accumulation, pipe blockage, and deterioration of wall heat transfer. Furthermore, whenever localized high temperatures appear in the loop, the fluid undergoes phase change to absorb heat, ensuring the maximum system temperature ( $T_{max}$ ) close to the boiling point of the cooling medium. The calculation results indicate that when the heat flux difference ( $\Delta Q$ ) loaded on the pipe wall on both sides is sufficiently large, all four cooling mediums undergo overall circulation. Moreover, as the duration of overall circulation increases, the fluid flow rate accelerates, increasing overall heat transfer. The influence of the viscosity of the cooling medium on the flow rate and heat transfer becomes insignificant. In addition, although there are differences in heat transfer capacity among the natural circulation systems using different cooling mediums, they all possess self-adaptive cooling capabilities by autonomously adjusting their circulation states based on external heat flux differences.



Figure 16:  $t_2 = 60$  s heat flow distribution on the wall of the four cooling mediums



**Figure 17:** Distribution of heat flux on the inner wall of the vertical pipe and liquid volume fraction 30 s after phase change: (a) water, (b) ethanol, (c) fuel oil, and (d) R113

## **5** Conclusions

The innovative open natural circulation system functions as the internal cooling pathway for transpiration cooling. It harnesses variations in heat flux across different sections of the piping to propel the cooling medium through the loop, facilitating the release of steam in the process. Four commonly used cooling mediums: liquid water, ethanol, fuel oil, and R113 are selected to look for physics-based laws. Through numerical simulation of natural circulation flow state within the loop, the flow, phase change, and heat transfer characteristics of different cooling mediums, as well as the influence of fluid thermal properties on the fluid flow state are analyzed. The main conclusions are as follows:

(1) When the walls of the U-shaped pipe are subjected to uniform heat flux, the fluid state of other cooling mediums is consistent with that of liquid water, which can also be sequentially categorized three stages: cooling medium expansion, localized circulation, and phase change. However, due to differences in temperature rise rates and fluid viscosity, the time required for different cooling media to experience these fluid states is different.

(2) As heating time increases, the time difference between different cooling mediums to reach phase transition state is more than 5 times, leading to different fluid flow velocities. Differences in fluid velocity and latent heat of phase change result in significant variations in heat exchange capacity among different

cooling mediums, with the maximum difference reaching up to 3 times. Furthermore, under phase change conditions, the influence of fluid velocity on heat exchange capacity is much greater than that of latent heat of phase change and fluid property. Additionally, heat transfer deterioration at the pipe orifice can be mitigated by appropriately adjusting the properties of the cooling medium.

(3) Boosting the heat flux on the pipe wall's right side prompts the fluid to flow counterclockwise within the loop. However, the threshold for heat flux disparity necessary between the pipe's two sides to commence full circulation varies across cooling mediums, influenced by the fluid density and the change of density with temperature increase. A heat flux difference of 100 W/m<sup>2</sup> suffices to induce overall circulation in all four cooling mediums. Faster circulation speed leads to greater heat transfer capacity, with a maximum of 7600 W/m<sup>2</sup>.

In conclusion, this study further elucidates the operation mechanism and physics-based cooling laws of the natural circulation system, providing valuable insights for its practical applications. Subsequent work can combine the natural circulation system with the upper transpiration cooling structure, designing a coupled transpiration cooling thermal protection system incorporating self-driven cooling pipes for comprehensive heat transfer analysis.

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