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The Experiment Study of a Passive Containment Cooling System on Heat Transfer Property under DBA Accident

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Abstract

Owing to a passive containment cooling system (PCCS) design in the AP/CAP series nuclear power plants, the containment can be cooled within 72 h after the accident using a passive containment cooling water storage tank (PCCWST). However, after 72 h, if the top tank is not able to replenish water in time, it is difficult to take all of the waste heat away by the containment itself; thus, the containment is still prone to overpressure risk. Aiming at the shortboard with a limited time of exporting the waste heat of the containment in a nuclear power plant, this paper studied in-depth a set of innovative passive heat removal system for the containment. In addition, an experimental system was set up to study the thermal performance of the system at different pressures, temperatures, and gas compositions under the condition of design basis accident. The results show that the heat capacity of the system completely meets the design requirements.

Keywords: Containment; Passive cooling system; Heat removal

1. Introduction

Since the Fukushima accident, two requirements are being considered in the development of new nuclear power projects in China: the most advanced technology and the most stringent standards. At the same time, new nuclear power plants are required to be designed in a way that eliminates the release of large quantities of radioactive materials. Containment, a set of general heat tubes arranged, is the last safety option to prevent the release of radioactive fission products from a nuclear power plant. How to further improve the technical scheme of a containment heat removal system to ensure the safety of the containment under accident conditions is also a new requirement being considered. Ensuring the integrity of the containment is a new and common challenge after the Fukushima nuclear accident.

Many scholars both at home and abroad have carried out relevant researches on the passive heat transfer of the containment, and have put forward multiple passive heat transfer schemes for containment after accidents. Razzaque et al.[1] proposed a heat-tube passive heat transfer system for a containment, in which containment, a set of general heat tubes are arranged, and inert gases are used as a buffer layer. Once the reactor is shut down (under normal and accident conditions), the core decay heat would cause the inert gas in the containment to form a natural circulation loop, and the heat tube removes the heat from the containment so that the temperature in the containment will not exceed the temperature limit. However, when the water cooling system is used as a cold source, it is necessary to fill the cooling pool with water. Sugawara and Lanchao Lin et al.[2] proposed a cooling scheme that uses a separated heat tube to drain heat into a pool outside the containment. Because heat transfer systems designed based on the separated heat tube technique adopt open cycle and function realization depends on an external tank, once water in the tank runs out, it still needs to be supplemented using an active system; thus, the system cannot meet long-term, safe, and reliable cooling requirements. Sviridenko et al.[3] proposed a three-cycle heat-tube passive heat transfer system, between which are an evaporator and a condenser; there is a clapboard separating them to ensure that the radioactive material does not leak out of the containment through a heat exchanger. However, the multi-loop design reduces the heat transfer ability and reliability of the whole system.

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The AP1000 or CAP1400 nuclear power plant designed PCCS. When an accident triggers its action, the PCCWST at the top of the containment can maintain passive operation by spraying water to the exterior of the steel containment, and the air baffle structure between the steel containment vessel and concrete containment structure forms a natural air circulation cooling channel within the shielded structure[4]. Through these approaches, the core waste heat can be effectively discharged, and the operator does not need to intervene within 72 h. Despite this, there is a decay heat generated after 72 h, and the heat transfer capacity of the containment system alone cannot fully export these decay heat. Therefore, human intervention is still needed to fill the top water tank to effectively export the residual decay heat. Table 1 compares the residual decay heat for different reactor types. Taking the AP1000 nuclear power plant as an example, after closing the spray, the containment cooling capacity was about 6 MW. However, after 72 h, the core decay heat is still 17 MW, i.e., 11 MW of decay heat was still needed to be derived. Therefore, after 72 h, we need to replenish water to the top spray tank to ensure continued heat export. Considering the need for long-term cooling of the containment, to enhance safety to a greater extent, the second passive containment cooling system (PCCS2)[5] for AP/CAP series reactors was designed to export waste heat. Fig. 1 shows the schematic diagram of the system.

On the basis of the digestion and absorption multiple schemes of the passive residual heat removal system of containment at home and abroad, PCCS2 continued its passive design concept. During operation, mixed fluid in the containment loop heats up the loop through condensation and natural convection, and the densities of the heated fluid in the heating section and the cold fluid in the external air cooler are different. As a result, a natural circulation is formed in the circuit, and the heat in the containment is taken out and finally exported to the atmosphere through an air circuit.

The system runs entirely on natural circulation principle, does not need an external power source support, can fundamentally solve the weak link of water replenishment need in other technical solutions, improves the ability to deal with power plant accidents, and helps in enhancing further the safety of the nuclear power plant.

Table 1. Comparison of residual decay heat of different reactor types

Time after shutdown（Ratio of residual heat）	Heat power（MW）	
Reactor type	AP1000	CAP1400
Full power（100%）	3400	4044
72 h（0.5%）	17	20
7d（0.35%）	12	14
30d（0.2%）	6.8	8.1

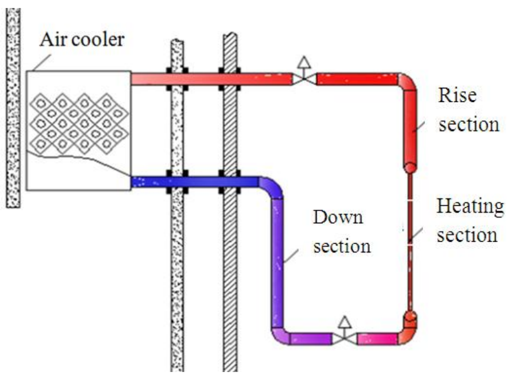

Fig. 1. Schematic diagram of PCCS2

2. Test System

PCCS2 was designed to generate a heat power of 1.5 MW. Considering the research conditions, funding, and other factors, the test system heat capacity can do a 1:2 zoom. A comprehensive performance test platform was built to study the effects of atmospheric pressure, temperature, and gas composition on thermal performance and operational stability of the whole system. Table 2 shows the design parameters of the test system.

Table 2. Design parameters of the test system

Parameters	Design value
Design pressure of Containment simulator（MPa）	0.8
Heat transfer power of the system（kW）	750
Steam temperature in the Containment simulator（℃）	145
Mass fraction of Non-condensable gas（%）	20-80
Flow rate of circulation（t/h）	72
Design temperature of cooling water circuit（℃）	120
Wall subcooling（℃）	5-30
Internal design pressure of heat exchanger（MPa）	0.6

2.1. Overall introduction of the system

Fig. 2 shows a picture of the test system. The test system has two operating modes: natural circulation and forced circulation. Between these systems, the natural circulation mode is mainly used to study the start-up characteristics of systems and heat transfer performance. The forced circulation mode is used to study the heat transfer performance of a single-unit system and a system under constant flow conditions. This paper focuses on the natural circulation mode. The test system includes many auxiliary systems such as air supply, steam supply, helium supply, plant water supply and circulation, containment environment simulation, gas sampling and analysis, and data monitoring and recording systems. The thermal process of the test system consists of three parts: condensation outside the steam pipe in the containment, heat exchange between the inner heat exchanger and external air cooler, and air convection heat exchange in the air duct.

The natural circulation system includes a containment simulator, an internal heat exchanger, an external air cooler, an air duct, an axial flow fan, an air heater, a pressure stabilizing tank, a water storage tank and related pipes, instruments, and so on. Fig. 3 shows a schematic diagram of the loop circuit.

A mixture of steam and the non-condensable gas mixture is formed in the containment simulator. The internal heat exchanger, the outer air cooler, and the connecting pipe are filled with water. The mixed steam in the containment simulator heats up the internal heat exchanger, thereby leading to a temperature difference between the fluid in the heat exchanger and the external air cooler. Because of density and height differences, the natural circulation is formed in the loop, and air in the outer air cooler and the air duct forms the heat exchange. Through this process, the wind speed of the air duct can be adjusted by controlling the speed of the fan; the air temperature of the inlet of the air duct is controlled by adjusting the power of the air heater. The fluid in the loop is a single phase during the whole cycle.

Fig. 2. Picture of the test system

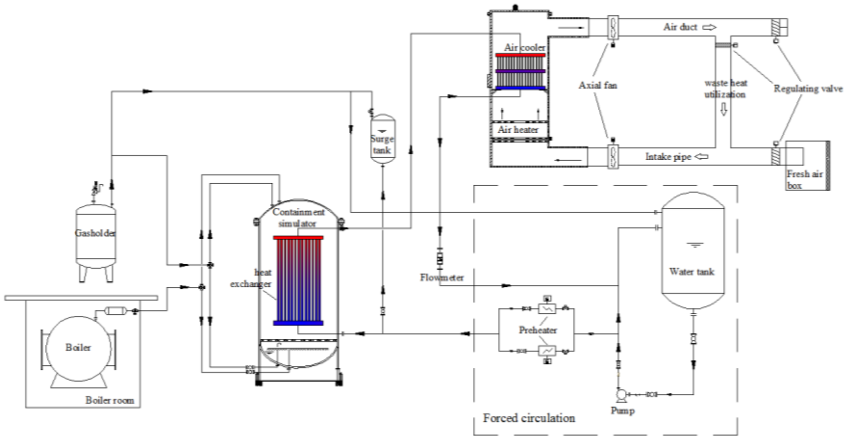

Fig. 3. Schematic diagram of the loop circuit

2.2. Auxiliary systems

2.2.1. Steam supply system

The steam generated by the electric heating boiler provides steam for the containment simulator. The maximum heating power of the boiler is 1.5 MW, and the design pressure is 1.25 MPa, which can provide up to 2 t/h saturated steam at the highest level. According to the specific test conditions, the steam generated by the boiler enters the containment through the vortex flowmeter. After that, thermocouple and pressure sensors are used to measure the parameters of the boiler steam.

2.2.2. Air supply system

The air supply system mainly provides a gas source for the containment simulator, pressure head for the storage tank, voltage regulator, the pneumatic valve, and other instruments. It is mainly composed of a gas storage tank, a pressure regulating valve group, a gas pre-heater, and corresponding pipe, valve, instruments, and so on. The test bench sets the 2 m^3 gas storage tank on site with a maximum inflation pressure of 0.8 MPa and an aeration capacity of 1 m^3/min. The gas distribution pipeline of the plant stops working after enough air is pumped into the gas tank.

2.2.3. Desalinated water supply system

A desalinated water system for the test circuit provides the required demineralized water. Demineralized water connects to the circulating water tank. By monitoring the liquid level and control valve opening to achieve automatic replenishment.

2.2.4. Instrument and control system

The control instruction of the comprehensive test system can be realized using a programmable logic controller (PLC) system, and data acquisition can be done using National Instruments's data acquisition system.

The PLC system consists of an industrial control machine, a PLC master station, a PLC slave station, a pump distribution box, and subsidiary equipment. PLC control logic includes liquid level control, temperature control, flow rate control, flow direction control, and so on. The data acquisition system includes the NI host, data acquisition software, measurement, and control cabinet. Important measurement signals such as temperature and pressure are recorded by the NI data acquisition system. The system transforms the signals generated by the thermocouples, pressure transmitters, and flowmeters into the corresponding physical quantities displayed on a human–machine interface and handles data storage at the same time. The device used is NI PXIe-1085, which includes 13 voltage modules and four current modules. The voltage module is used to measure the temperature signal, and each module has 32 channels. The current module measures current signals such as pressure and flow. Each current module has eight channels.

2.3. Major equipment

2.3.1. Test body

The test body consists of a containment simulator, an internal heat exchanger, and a measuring part. As shown in Fig. 4, the containment simulator is a carbon steel cylindrical vessel with an inner diameter of 4.5 m and a total height of 10 m above the base. The test system can switch up and down intake, which causes the mixed gas to enter the test pressure tank and condense on the test piece. The condensate is gathered at the bottom of the containment simulator. After filtration treatment, it is recovered using the boiler. Near the gas inlet and outlet, two layers of gas (steam) orifice plates are installed to make the mixed gas entering the containment simulator more homogeneous. The internal heat exchanger is composed of a heat pipe and up and down headers. The heat exchanger tube consists of 180 roots in relative symmetry with the header. The heat transfer tube is C-shaped and arranged in a triangular form.

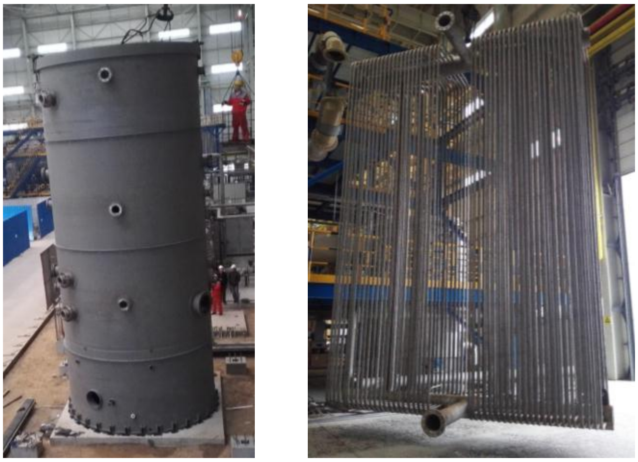

Fig. 4. Picture of the test body

To analyze the thermal performance of the system, the parameters of the containment simulator, such as the thermal environment, wall temperature of the heat exchanger, and temperature of the cooling circuit, need to be measured. Therefore, 371 temperature and some other measurements were set up in the containment simulator. The arrangement form of temperature measurement points are as follows:

There are 105 measuring points on the inner wall of the containment and ambient systems. 18 measuring points are arranged in different large spaces between the current-sharing orifice at the bottom and the dome to measure the spatial temperature distribution. To monitor the spatial temperature and the gradient of the temperature near the heat exchanger, measuring points were arranged in six different circumferential locations near the heat exchange tube bundle. The number of measuring points was 84, and they were 0.1 m away from the inner and outer sides of the heat exchange bundle. Three measuring points were arranged along the inner wall of the containment to measure the wall temperature.

To measure the temperature of the internal heat exchanger, the measuring points were arranged in the inlet and outlet headers of the heat exchanger, the wall of the heat transfer tube, and the vicinity of the heat exchanger. The total number of measuring points was 266. Among them, 14 measuring points were arranged in the import and export headers to measure the temperature. 252 measuring points were arranged in the axial position of the six different axial positions of the heat transfer tube. The layout of the measured points in each position is the same, with a total of 42 measuring points. In each position, a pair of thermocouples was arranged along the axially fixed interval of the heat exchange tube. The wall temperatures of the two heat exchange tubes were measured, and a measuring point was placed at the same location between the inside and outside heat exchanger tubes to measure the temperature of the main body.

2.3.2. External air cooler

The external air cooler and its air duct are very important parts of the timeless passive residual heat removal system and is the characteristic that is different from other containment waste heat removal systems. Air cooling was used outside the outer air cooler. The convection heat transfer coefficient of air is very low. To reduce the volume of the external air cooler, the fin tube was used to strengthen the heat transfer. The structural parameters of the external air cooler are given in Table 3. Fig. 5 shows a schematic diagram of the structure of an external air cooler. The inlet temperature of the air side of the external air cooler needs to be controlled. To reduce the load of the air heater, the air temperature was adjusted by mixing the fresh air and hot air in the design, and a precise temperature control was realized by adjusting the power of the air heater. The speed of the fan can be adjusted using the frequency converter to meet the control demand of the air flow velocity. At the same time, the local parameters can be measured using thermocouples, pressure sensors, and anemometers near the air duct, air cooler, and air heater.

Table 3. Design parameter range of the external air cooler

Parameters	Design value	Parameters	Design value
Total height （m）	2	Heat transfer tube arrangement	Triangular arrangement
Diameter of heat transfer tube （m）	0.02	Heat transfer tube number	180
Design pressure inner the tube （MPa）	0.6	heat transfer tube spacing （m）	0.04
Design flow rate inner the tube （t/h）	90	Inlet temperature inner the tube （℃）	70-120
Air side design flow rate （m/s）	0.3-2	Air side design inlet temperature （℃）	35-55

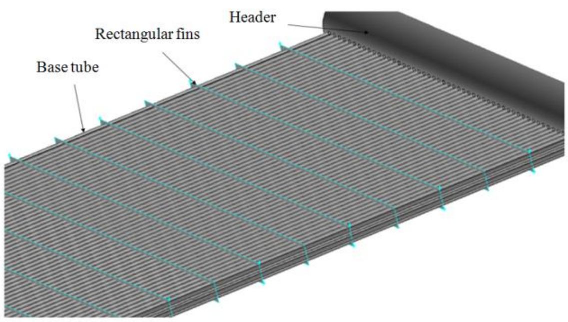

Fig. 5. Schematic diagram of the structure of an external air cooler

3. Data Analysis

At the beginning of the experiment, steam (air) was injected through the bottom inlet, and the fluid in the containment simulator was discharged using the top exhaust valve. The changes in the temperature and pressure values of the space in the containment were monitored until the environment in the containment reached a preset value. At the same time, the preset parameters of air in the air duct were adjusted, and the circulating flow rate was detected. When the circulating flow rate was stable, parameters such as the circulating flow rate, temperature, and pressure were recorded, and the environment inside the containment was kept stable during the whole process. The recorded data were used in the analysis of the thermal performance of the system.

3.1. Data processing

This section discusses how the data generated from the heat capacity test of the whole passive heat conducting system were processed and the calculation of important parameters of the system such as air occupation ratio, and system heat load. Processing of the test data was mainly composed of two parts: one is the determination of the mixture of gases, and the other is the determination of the heat load of the system. Suppose the test has been measured with the total pressure *P* of the mixed gas, the average temperature $\overline{T_b}$ of the gas mixture, and the circulating flow rate of *G*.

Under steady state, the partial pressure of air is calculated using the total pressure of the containment simulator, total pressure of the body, and partial pressure of steam. The latter is calculated using the average temperature of the mainstream of the containment simulator and checked using a steam table. It is assumed that the steam is in a saturated state, and both air and steam are considered as ideal gases. According to Dalton's law of pressure, we have

$$\frac{m_a}{m_v} = \frac{n_a}{n_v} \cdot \frac{M_a}{M_v} = \frac{P_a}{P_v(\overline{T_b})} \cdot \frac{M_a}{M_v} \quad (1)$$

where m is the mass in *kg*, n is the number of molecules in *mol*, *M* is the molecular weight, *a* denotes air, and *v* denotes steam.

Then, the ratio of air mass to steam mass is

$$\frac{m_a}{m_v} = \frac{29(P - P_v(\overline{T_b}))}{18P_v(\overline{T_b})} \quad (2)$$

Therefore, the mass fraction of the air is as follows:

$$W_a = \frac{m_a}{m_a + m_v} = \frac{29P - 29P_v(\overline{T_b})}{29P - 11P_v(\overline{T_b})} \quad (3)$$

where *W* is the mass fraction.

In the test, the simplest way to change the working condition is to exhaust the outside of the containment. After opening the vent valve for a period of time, the pressure of the containment simulator will gradually recover, which enables the mixed gas to rebalance under the new mass ratio. We do not need to measure the mass of the exhaust gas. The mass ratio of the mixed gas can be calculated using the above three steady-state parameters.

Heat transfer is the most direct indicator to measure the design requirements of a complete set of passive heat conduction systems. In this experiment, the heat transfer in the condensation heat transfer process outside the tube was determined by solving the water side heat transfer. The containment simulator was outsourced with a thermal insulation material, so it can be assumed that the heat released from the condensation of the mixed gas is completely carried away by the cooling water in the heat transfer tube. Therefore, the temperature rise and the flow rate of the inlet and outlet cooling water can be used to calculate the heat transfer of the test as follows:

$$Q = G(i_2 - i_1) = \rho V(i_2 - i_1) \quad (4)$$

where *Q* is the heat transfer quantity of condensation in *kW*, G is the mass flow of cooling water in *kg/s*, *V* is the volume flow of cooling water in m^3/s, ρ is the density of cooling water in kg/m^3, i_2 is the outlet enthalpy of cooling water in *kJ/kg*, and i_1 is the inlet enthalpy of cooling water in *kJ/kg*.

During the condensation of steam containing non-condensable gases, the heat released by the steam is given by

$$Q = h\,A \cdot \left(T_b - T_{cw}\right) \quad (5)$$

where *Q* is the heat transfer power in *W*, A is the heat transfer area in m^2, T is the temperature in ℃, h is the outside tube heat transfer coefficient in $W/(m^2 \cdot ℃)$, *b* is the mainstream space, and *cw* denotes condensation wall.

According to the principle of thermal equilibrium under steady conditions, it can be concluded that the condensation rate outside pipes of the internal heat exchanger is the same as the heat absorption rate by the fluid inside pipes. That is, Q in equation (5) is the heat power of the system. Many scholars have carried out relevant researches[6–8] on how to calculate heat transfer coefficient h in equation (5), but the application scope that they considered is different from the one considered in this paper. To design this passive heat transfer system more accurately, our team also set up a single performance test in prophase, carried out researches on condensation heat transfer outside tubes, and gave the experimental correlation of the condensation heat transfer coefficient h with wall super cooling degree, pressure, and the change of air mass fraction as follows:

$$h = \left(T_{\mathrm{b}} - T_{\mathrm{cw}}\right)^{-0.30}\left[-3338.03P + 1014.31 + \left(-121.18 - 19754.61P\right)\mathrm{Log10}\left(W_{\mathrm{a}}\right)\right] \tag{6}$$

Application scope: $0.35 \le W_a \le 0.80$, $0.20 \le P \le 0.40$ *MPa, and* $15\ ℃ \le T_b - T_{cw} \le 58\ ℃$. Through equations (5) and (6), in addition to the relevant parameters obtained in the experiment, the condensation heat exchange outside the tubes can be calculated.

3.2. System heat transfer performance analysis

In this paper, the passive heat transfer system of the containment was mainly aimed at the AP/CAP series reactor which was used to export the residual heat for the long-term cooling stage 72 h after an accident. Therefore, in the system design stage, we modeled the AP1000 reactor as a prototype using mature calculation software (COCOSYS) for modeling of containment and calculated the thermal environment in the containment under a typical double-ended cool leg guillotine (DECLG) accident condition. The result shows that after 72 h under DBA condition, the total pressure in the containment was 0.23 MPa, and the air mass fraction was 62%. According to the actual situation, the test conditions were designed as shown in Table 4. As a base condition, the system was designed to have a total pressure 0.23 MPa. From the point of heat transfer effectiveness, the heat exchange between the containment environment and internal heat exchanger relied mainly on the steam condensation outside tubes. Therefore, high levels of air mass fraction can have a negative effect on heat transfer. Considering the conservatism of the system’s heat transfer performance, the air mass fraction in the range of 42% to 71% was used to fully demonstrate the system’s heat transfer performance. Similarly, after 72 h of the accident, if the top tank was not filled with water in time, only by the thermal conductivity of containment, the pressure in the containment will still increase. In order to improve the coverage of the design, we have also carried out experiments at higher pressure conditions.

Table 4. Design parameters of the test system.

Air mass fraction (%)	Total pressure (MPa)	Steam partial pressure (MPa)	Ambient temperature (°C)
42	0.23	0.159	113
50	0.23	0.142	110
56	0.23	0.128	106.8
61	0.23	0.117	104
66	0.23	0.104	101
71	0.23	0.091	97
61	0.35	0.178	116.5
65	0.35	0.163	113
71	0.35	0.139	109
75	0.35	0.122	105

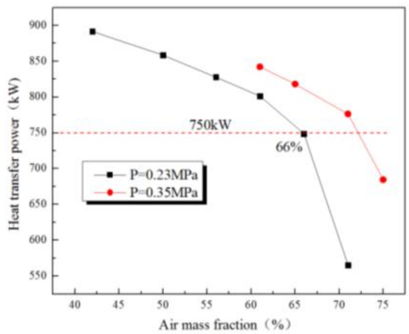

Fig. 6. Curve of heat transfer power

According to the experimental data and equation (4), the heat transfer power of the system under different experimental conditions can be calculated. Fig. 6 shows the variation curves of the system's heat transfer power under different pressure and air mass fraction conditions. Firstly, under the condition of 0.23 MPa total pressure in the containment, the heat transfer capacity of the system increases with a decrease in air mass fraction. When the air mass fraction was 66%, the heat transfer power of the system reached the design value of 750 kW. When the air mass fraction was reduced to 42%, the system's heat transfer power could reach as high as 891 kW. This result shows that after 72 h of DBA accident, if the air mass fraction is lower than 66%, the system can take away 1.5 MW from the containment, thus ensuring that the containment is not over-pressurized.

The relation between heat transfer power and air mass fraction under different pressures in the containment is shown in Fig. 6. The results show that the change trends of heat transfer performance under 0.35 and 0.23 MPa are the same. However, when pressure increases, the partial pressure of the steam increases under the same steam mass fraction, and the corresponding saturated temperature and steam mass flow also increase. Therefore, for the internal exchanger, the temperature difference between the wall of the tubes and the mixed steam is bigger. Under the same gas composition, when the pressure in the containment is higher, the heat transfer performance of the system is better. When the pressure in the containment was 0.23 MPa and the air mass fraction was 71%, the system's heat transfer power was 566 kW. Under the same gas composition, when the pressure in the containment was up to 0.35 MPa, the heat transfer power of the system reached 776 kW, which is more than design. It shows that, even if the system is delayed after the accident, it can still have the desired effect. And under the same gas component condition, higher the pressure is, stronger the heat transfer capacity of the system could be.

It can be seen from Fig. 6 that the pressure of the system and the non-condensable gas content are important factors that affect the system's heat capacity. The effect of the above two factors on the heat transfer coefficient was taken into consideration in equation (6). However, equation (6) was derived from a single tube test. Therefore, the heat carrying capacity deviation of the system caused by the tube bundle effect and structural design of the heat exchanger still need to be verified.

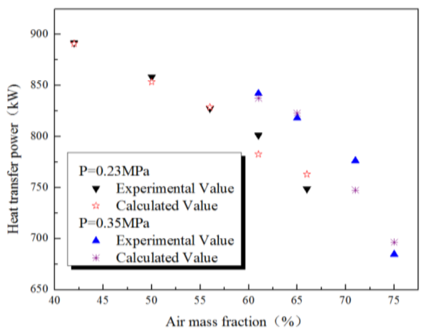

Fig. 7. Comparison between the calculated value of the single tube and the experimental value of the comprehensive system

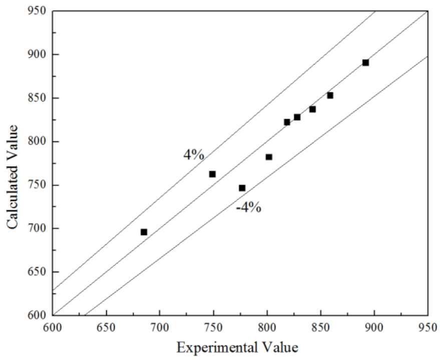

Fig. 8. Comparison between the calculated value and experimental value of the comprehensive system

In this paper, the system's heat transfer power was calculated using equation (5) and equation (6) and then compared with the results calculated using equation (4). The comparison results are shown in Fig. 7. The results show that the experimental results are significantly higher than the calculated results, there are large deviations. Because equation (6) is from the single test case, the applicable range of wall subcooling is *15 ℃* to *58 ℃*. However, in this experiment, owing to the influence of the shell structure and tube bundle effect, the spatial flow and mixing of gas are more significant. Therefore, the degree of wall subcooling outside the tube is just *1.5 ℃–6.5 ℃*, far less than the scope of formula (6). As a result, the deviation of the calculated result is large. To better guide the follow-up optimization of the system, the experimental correlation of the condensation heat transfer coefficient with the subcooling, pressure, and air mass fraction under low subcooling condition is fitted in the form of equation (6) as follows:

$$h=\left(T_{\mathrm{b}}-T_{\mathrm{cw}}\right)^{-0.76}\left[2331.7-2137.7P-\left(4265.1+20755.6P\right)\mathrm{Log10}\left(W_{\mathrm{a}}\right)\right] \tag{7}$$

Application scope: $0.35 \leq W_a \leq 0.80$, $0.20 \leq P \leq 0.40$ *MPa, and* 15 ℃ $\leq T_b - T_{cw} \leq 58$ ℃. Fig. 8 shows the results of a comparison of the numerical values obtained using equation (7) with the experimentally generated values. The results show that the deviation is within ±4%, which is an acceptable value.

4. Conclusion

In this paper, we carried out experiments on different working conditions under the DBA condition and studied the heat transfer performance of the system with different air contents and at different pressures. The conclusions are as follows:

1. Through the experimental verification, the PCCS2 studied in this paper meets the design requirements with sufficient thermal load at DBA conditions and some design margins.

2. The experimental verification of the correlation of the condensation heat transfer coefficient of tubes with non-condensable gases under low wall subcooling conditions is given. The calculated values are in good agreement with the experimental values and can be used in subsequent optimization design of the system.

3. This article only analyzed the heat transfer performance of the system under DBA conditions. In the future, we will carry out research on the start-up characteristics of the system and perform experiments on the conditions of various non-condensable gas components under the condition of beyond design basis accident (BDBA) to improve the overall performance of the system.

Acknowledgments

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