Concept design of primary heat transfer system for CFETR vacuum vessel

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Abstract. The Chinese Fusion Engineering Testing Reactor (CFETR) aims at providing technical support and data reference for future fusion demonstration reactor. The vacuum vessel primary heat transfer system (VV PHTS), playing a vital role in guaranteeing the safety operation of tokamak device, which is a part of the tokamak cooling water system. It will operate in three main modes: plasma operation, water baking and decay heat removing. The design of the VV PHTS should have three functions corresponding to the above modes. Based on the thermal-hydraulic requirements, the system process have been proposed. The concept design of the system has been performed including pipes and pressurizer. For the purpose of obtaining the parameters of other main components and predicting the system operating behaviour, the thermal-hydraulic analysis model has been developed by using AFT fathom mode. The steady-state results of all three operating modes such as velocity and pressure show that the designed VV PHTS and selected methodology are feasible.

1. Introduction
Magnetic confinement fusion is an approach to obtain fusion power by using special magnetic field to confine the ultra-high temperature plasma composed of deuteron, triton and free electrons in a limited volume. It is considered as the most feasible way to solve the energy problem [1-3]. As the next generation of tokamak magnetic confinement device after EAST in China, the Chinese Fusion Engineering Testing Reactor (CFETR) has higher operating requirements of parameters [4].

The Tokamak Cooling Water System (TCWS), Component Cooling Water System (CCWS), Chilled Cooling Water System (CHWS) and Heat Rejection System (HRS) have the capacity to guarantee the safety operation of the device at high temperature and convert fusion power into electric energy. The Vacuum Vessel Primary Heat Transfer System (VV PHTS) is one of the essential part of TCWS, which extracts the heat deposited in vacuum vessel and prevents the structure of the VV from deforming or even cracking. In order to meet the different requirements during the test, the concept design of the VV PHTS should have 3 main functions: cooling, baking and decay heat removing. Moreover, the system should have the ability to switch between different functions without affecting each other which makes the VV PHTS design challenging. Presently, the research work has been...
developed which is relevant to cooling water system applied in fusion engineering [5-7]. Yuya Miyoshi et.al [8] have complete a holistic design of cooling water system for Japan’s DEMO. Donato Lioce et.al [9] have proposed an arrangement of the VV PHTS for International Thermonuclear Experimental Reactor (ITER). Most of the research work is to obtain key parameters through numerical calculation. In this study, the simulation model is built on the basis of calculation, and the thermal-hydraulic analysis is performed to verify the rationality of the concept design.

The content is organized in 5 sections. The system requirement and process are discussed in Section 2. Then the details of thermal-hydraulic analysis including model description, formulation and simulation results are presented in Section 3. The discussion and conclusion are shown in the end.

2. Overview of CFETR VV PHTS design

The design of VV PHTS should satisfy the following two aspects: functional requirements and operational requirements. The functional requirements which has been introduced in the previous section. The operational requirements mean that the system parameters should be maintained at a specific level to achieve its function. Therefore, the VV PHTS is composed of the following parts: cooling line, baking line and decay heat removing (DHR) line to meet the requirements.

2.1. System operational requirement

The primary heat transfer system preliminary design requirements are shown in Table 1. During plasma operation the VV PHTS will remove the heat deposited inside vessel clients by supplying cooling water at the specific flow rate, pressure and temperature through the cooling line. For the purpose of avoiding a large temperature difference between vacuum vessel and blanket module which may affect the structure of blanket, the inlet temperature is set at 200°C. Likewise, it is the baking temperature to achieve the goal of removing the impurities remaining in the vessel. The inlet pressure both in plasma operation and water baking should be 2.8–3.2 MPa which is regarded as an appropriate level for the saturated pressure of the hot water. The flow rate in plasma operation and water baking are 2860 kg/s and 286 kg/s. In addition, in case of loss of off-site power accident or other abnormal operating conditions, 300 kg/s of cooling water is still needed to be pumped into the VV. All current parameters will be updated as further studies are developed.

| Thermal-hydraulic parameter | Plasma Operation | Water Baking |
|----------------------------|-----------------|--------------|
| Thermal Load (MW)          | 64              | 0            |
| Coolant Inlet Temperature (°C) | 200           | 200          |
| Coolant Outlet Temperature (°C) | 205            | 200          |
| Coolant Inlet Pressure (MPa)   | 3.0 (+/- 0.2)  | 3.0 (+/- 0.2) |
| Pressure Loss (MPa)         | 0.3             | ~0.01        |
| Flow Rate (kg/s)            | 2860            | 286          |
| Flow Rate During Decay Heat Removing (kg/s) | 300        |

2.2. System process design

The schematic of the VV PHTS is illustrated in Figure 1. The VV is initially divided into a section at 40°. Therefore, the cooling water is routed through the main distributor which can distribute the coolant into 9 channels from upper ports, and then the coolant which passes through each client can be collected by the main collector located below the tokamak device.

Four parallel centrifugal pumps circulate cooling water through the VV, the nuclear heat will be taken out and transferred to the secondary side by the heat exchanger during plasma operation. The heater in the baking line can heat circulating water to the required temperature. Isolation valves are installed at the upstream and downstream of each client for the purpose of isolating the client during maintenance. The flow control valve in parallel with the heat exchanger can optimize the size of the heat exchanger by sharing part of the coolant. Also the flow control valve connected to the client is
used to ensure the flow balance between each branch. Pressurizer has the function to maintain the pressure at a stable value and to accommodate the coolant expansion and contraction caused by temperature transients. In addition, the VV PHTS is subdivided according to the component categories. For example, the cooling line is composed of primary pump line, primary bypass line and primary heat exchanger (HX) line.

![Figure 1. Schematic diagram of VV PHTS](image)

### 3. Thermal-hydraulic analysis

For the purpose of predicting system behaviour, the thermal-hydraulic analysis of VV PHTS is carried out by formulation and simulation. The results of pipes and pressurizer by calculating will be input into the simulation model as initial parameters. The thermal-hydraulic analysis theory and the details of simulation are introduced in this section.

#### 3.1. Formulation

3.1.1. **Pipe sizing.** The details of pipe selection such as size, type and thickness are shown in Table 2. The dimension of pipes in different parts of the system are determined according to the corresponding flow rate. The determination of the common pipes in different modes is relevant to the most crucial condition. As can be seen from Table 1, the flow rate of the system is maximum in plasma operation mode. Therefore, the dimension of main supply and return pipes is determined in this condition, otherwise the excessive velocity will cause a huge pressure loss when the coolant flows through the pipe. Furthermore, the final results of pipe size need to take both the site limitation and the cost into account. The initial pipe diameter is calculated according to the reference flow rate by equation (1):

\[
D_{in} = 2 \left( \frac{Q_v}{\pi v} \right) \quad (1)
\]

where \(D_{in}\) is the initial inner diameter, \(Q_v\) is the volume flow rate, \(v\) is the reference velocity which is 9 m/s.

The initial thickness is given by equation (2) [10]:

\[
t_m = \frac{\rho D_{in}}{2(S_m+F_Y)} + A \quad (2)
\]

3
where \( t_m \) is the minimum thickness, \( P \) is the design internal pressure, \( S_m \) is the maximum allowable stress strength of the material at the design temperature of the core level 1, \( y \) is a constant of 0.4, \( d \) is the appropriate additional thickness.

The final thickness of the pipe is determined by rounding up the calculated thickness to the standard thickness of the pipe.

**Table 2. Main parameters of pipes**

| Part                        | DN (mm) | \( D_w \) (mm) | Thickness (mm) | Length (m) |
|-----------------------------|---------|----------------|----------------|-----------|
| Primary pump line           | 350     | 336.6          | 20.20          | 44        |
| Main supply and return line| 700     | 679.5          | 20.25          | 158       |
| Primary HX line             | 300     | 304.8          | 10.10          | 12        |
| Baking heater line          | 250     | 254.5          | 9.25           | 12        |
| Primary bypass line         | 650     | 631.9          | 14.05          | 10        |
| DHR pump line               | 250     | 254.5          | 9.25           | 30        |
| DHR HX line                 | 150     | 154.1          | 5.45           | 8         |
| DHR bypass line             | 200     | 202.2          | 8.4            | 8         |
| Tokamak internal line       | 250     | 254.5          | 9.25           | 90        |

**3.1.2. Pressurizer volume sizing.** Due to the instability of plasma the VV bears a fluctuating thermal load. The change of heat load leads to the continuous change of coolant temperature in the VV, which leads to the volume change of coolant in the loop and finally to the pressure change of the system. If the pressure is too high, the pipes and equipment of the system are exposed to the risk of overpressure rupture; if the pressure is too low, it is easy to generate local coolant boiling, and the vapor generated by evaporation will seriously affect the structure of the VV. The tank of the pressurizer is able to accommodate the changes of coolant volume in the loop. Also the electric heater located in the bottom of the pressurizer has the ability of stabilizing the system pressure. In this section, the preliminary design of the pressurizer tank is carried out, and the relevant performance of the electric heater will be developed in the future.

The inner diameter of pressurizer is selected as 3.0 m with a consideration of the workshop space. The internal volume of the pressurizer is mainly composed of the follow 6 parts: transient volume under static power V1, transient volume under superheat condition V2, transient volume under supercooled condition V3, volume of the heating zone V4, volume of liquid level measurement error V5 and volume of operating margin V6.

During VV PHTS operation, the coolant is heated from 200°C to 205°C, resulting in an increase of coolant volume. Part of the coolant flows into the pressurizer caused by the volume expansion and its temperature becomes the saturation temperature at the internal pressure of the pressurizer. The volume change in this process is the transient volume under static power V1 which can be expressed by equation (3):

\[
V_1 = V_0 \left[ \left( \frac{1}{\mu_0} - \frac{1}{\mu_{200}} \right) + \alpha \left( \frac{1}{\mu_{200}} - \frac{1}{\mu_{205}} \right) \right] \mu_B
\]  

Where \( V_0 \) is the total volume of VV PHTS, \( \mu_0, \mu_{200}, \mu_{205} \) are the specific volume of coolant under off-power, 200°C and 205°C, \( \mu_B \) is the specific volume of the water inside pressurizer, \( \alpha \) is the ratio of the volume of hot leg to the total volume of the system.

The transient volume under superheat condition V2 and transient volume under supercooled condition V3 can be got by equation (4) and (5):

\[
V_2 = V_0 \left( \frac{1}{\mu_1} - \frac{1}{\mu_{max}} \right) \mu_B
\]  

\[
V_3 = V_0 \left( \frac{1}{\mu_{min}} - \frac{1}{\mu_1} \right) \mu_B
\]
Where \( \mu_1 \) is the specific volume under normal condition, \( \mu_{\text{min}}, \mu_{\text{max}} \) are the specific volume at the upper limitation and lower limitation of the temperature.

The volume of the heating zone which is divided into the lower head volume \( V_{4a} \) and the volume occupied by heaters extending in the cylinder part \( V_{4b} \) is given as equation (6) and (7):

\[
V_{4a} = \frac{2}{3} \pi \left( \frac{D_{\text{PZR}}}{2} \right)^2 \left( \frac{D_{\text{PZR}}}{4} \right)
\]

(6)

\[
V_{4b} = 0.25 \pi D_{\text{PZR}}^2 * l
\]

(7)

Where \( D_{\text{PZR}} \) is the inner diameter of pressure, \( l \) is the length of the heater in cylinder.

The volume of liquid level measurement error \( V_5 \) and the volume of operating margin \( V_6 \) are set as 10% of the cylinder volume [11].

The results of the pressurizer size are shown in Table 3.

**Table 3. Volume of pressurizer**

| Volume Section                    | Value (m³) |
|----------------------------------|------------|
| Transient volume under static power V1 | 2.37       |
| Transient volume under superheat condition V2 | 11.81     |
| Transient volume under supercooled condition V3 | 11.30     |
| Volume of the heating zone V4    | 9.18       |
| Volume of liquid level measurement error V5 | 4.33      |
| Volume of operating margin V6    | 4.33       |
| Total volume                     | 43.32      |

### 3.2. Simulation

#### 3.2.1. Model description

Based on the preliminary design of CFETR VV PHTS in section 2, a thermal-hydraulic analysis model is established by using AFT fathom code to predict the behaviour of the system. Details of the thermal-hydraulic model are shown in Figure 2. In the Fathom model, most of the key components have a specific model that corresponds to the equipment of the system such as pumps, heat exchangers, isolation valves, check valves and control valves. Control valve in green is set as “flow control” to guarantee that the flow rate of each branch can meet the requirements. Isolation valve in brown has the function to switch between different modes. The flow resistance coefficients of all isolation valves and check valves are based on the industry experience. The total flow rate of 4 parallel primary pumps is 2860 kg/s. In order to prevent evaporation at the primary pump inlet, the pressure value of pressurizer is input according to the requirement of client inlet pressure. Specially, client is modelled as heat exchanger in yellow to absorb thermal load and heat up the temperature of coolant. Moreover, key pressure reference points are marked in the model and each branch is numbered for next analysis.
3.2.2. **Velocity profile.** The analysis model is capable to calculate the velocity of coolant at each point of the VV PHTS through the pipe size input. Figure 3 presents the velocity distribution of the system in plasma operation mode which is regarded as the most crucial condition. The maximum velocity is 9.768 m/s which is lower than the maximum design velocity at 12 m/s. And the velocity of the main supply and main pipes is 9.111 m/s, and the pipe size is DN700, which is also a common dimension in engineering.

3.2.3. **Pressure profile.** Figure 4 shows the inlet pressure of 9 clients in three modes. It is clear that the inlet pressure of each client is within the designed pressure range at 2.8~3.2 MPa. Generally, the inlet pressure of baking mode is higher than other modes and the maximum pressure is 30.79 bar.
which happens in branch D. The minimum pressure is 29.35 bar which is under plasma operation mode.

Figure 5 shows the pressure at each key point including the inlet and outlet of the pump, heat exchanger, baking heater and client for different modes. The black curve represents the pressure drop of the VV PHTS during plasma operation, the red curve and the blue curve show the change of system pressure in baking mode and decay heat removing mode respectively. Obviously, the value of the pressure drop is the largest in plasma operation mode because the high velocity coolant flow through the junctions can cause a huge pressure loss. Therefore, the size of the primary pumps is determined in this condition and shown in Table 4. The pressure of the primary pump outlet is 31.63 bar. The client and the heat exchanger account for the majority of the total pressure loss. And the pressure loss of the pipe from equipment building to tokamak hall is 2.01 bar. The VV PHTS share the same set of pumps and pipes in plasma operation mode and baking mode, while the flow rate in baking mode is only 286 kg/s which causes a very small pressure loss in this mode. Furthermore, the variation of system pressure in decay heat removing mode provides a basis for calculating the size of decay heat pump.

![Figure 4. The inlet pressure of each client](image)

![Figure 5. The pressure drop in three modes](image)

| Vol. Flow (m³/h) | Mass. Flow (kg/s) | Head (m) | Overall Power (kW) | NPSHA (m) |
|------------------|------------------|---------|-------------------|----------|
| Primary Pump     | 2973             | 715     | 192.2             | 1347     | 10.14    |
| DHR Pump         | 1247             | 300     | 88.69             | 260.8    | 22.58    |

4. Discussion
As can be seen from Figure 3, the velocity of each point is in the range of 7 m/s to 10 m/s which is regard as a suitable value according to industry experience. The information of the client inlet pressure illustrates that the maximum pressure is 3.079 MPa which meets the system design requirement. Total pressure drop in plasma operation mode and decay heat removing mode are 1.496 MPa and 0.714 MPa which should be replenished by pump head. The key parameters of pressurizer tank and pump are shown in Table 3 and Table 4. The total volume of pressurizer is 43.43 m³, while the heads of primary pump and DHR pump are 192.2 m and 88.69 m.

5. Conclusion
A concept design of the VV PHTS has been developed to satisfy three main operating modes of the CFETR vacuum vessel. System process is proposed according to the system requirements and functions. The approach of calculating the parameters relevant to the dimension of pipes and
pressurizer is presented. Moreover, thermal-hydraulic analysis of the system is performed by using the AFT Fathom model for the aim of predicting system operational behaviour in different modes. The simulation results show a good agreement with the design value both in velocity, pressure and temperature, which indicates the feasibility of the designed VV PHTS for CFETR. The design methodology presented here can be a good reference for the design of other cooling water system applied in fusion device. Besides, the results of steady-state analysis can be a solid foundation for evaluating transient responses of the system.

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