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Research Paper

Design of Cooling Water System (CWS) for CRAFT Lower Hybrid Current Driven (LHCD) System

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Abstract. Lower hybrid current driven (LHCD) system, as the most efficient non-inductive current drive method in the tokamak, is an integral part of the Comprehensive Research Facility for Fusion Technology (CRAFT). The cooling water system (CWS) is necessary to be designed to remove the thermal power generated by the clients of the LHCD system so that system could operate safely. Therefore, a thermal hydraulic model is developed by AFT Fathom aimed at investigating thermal hydraulic behavior of the system under the normal operation. According to the calculation results, the CWS can provide required pressure, temperature and flow rate to address the client requirements of LHCD system. The study could provide a design reference for the construction of cooling water system for LHCD system and other CRAFT subsystems.

Keywords: CRAFT, LHCD, Cooling water system, Thermal hydraulic, Design.

1. Introduction

Comprehensive Research Facility for Fusion Technology (CRAFT) is aimed to demonstrate a comprehensive research platform with the highest parameters and the most complete functions in the field of magnetic confinement. As one of important subsystems of CRAFT, Lower hybrid current driven (LHCD) is the most efficient non-inductive current drive method in the tokamak. During the LHCD system operation, more than 5.67 MW of heat power was generated by the clients of LHCD system, which is need to be removed to the heat rejection system (HRS). Once the heat power is not discharged in time, the clients would be damaged so that tokamak could not operate stably. Therefore, it is necessary to design the cooling water system (CWS) to avoid the external impact factors and meet the limit condition of the pressure and flow requirements.

Bin Guo et al. (2015) applied AFT Fathom code to predict the thermal-hydraulic behavior of ITER component cooling water system loop 2B (CCWS-2B). The main thermal-hydraulic parameters of all clients were obtained to evaluate whether parameters meet operational requirements, such as flow rate, pressure drop and outlet temperature. In addition, the pump size of the CCWS-2B was also obtained. The analysis method proposed by author can provide a reference for the design for the cooling water system.

Bin Guo et al. (2016) carried out coolant distribution system (CDS) design for ITER PF AC/DC converter using a thermal-hydraulic analysis model by AFT Fathom. The analysis results predicted the flow rate, temperature and pressure distribution of the system under the normal operation to evaluate the thermal-hydraulic behavior of the system. Meanwhile, the CDS prototype for the ITER PF converter has been constructed and some experiments have been performed on it. A good agreement of the flow distribution and temperature behavior between the simulated and test results validate the proposed design methodology.

Wei Zhang, Bin Guo, et al. (2017) developed a complicated thermal-hydraulic model to perform the hydraulic analysis on ITER component cooling water system 2B by AFT Fathom 9.0. The work was based on the Indian Domestic Agency (IN DA) fathom model and carried out thermal-hydraulic calculation and evaluation. Three solutions were introduced to improve the lack of fluid at some worst case points by using the current selected pump. Then, pump selected by IN DA cannot meet the client requirement. Finally, the author proposed the revised pump size and showed the pressure profile and flow distribution to verify the correctness.

P.A. Di Maioa, et al. (2018) performed thermal-hydraulic optimization of the DEMO divertor cassette body cooling circuit equipped with a liner by the Finite Volume Method and adopting the commercial Computational Fluid-Dynamic code ANSYS-CFX. The thermal-hydraulic performances of the cassette body were assessed in terms of coolant and structure temperature, coolant overall pressure drop, flow velocity distribution and mass flow rate fed to the Liner cooling circuit. The research results showed some major criticalities, mainly in terms of water coolant vaporization as well as non-symmetric coolant distribution between the two Liner inlets. Therefore, the author provided three potential solutions to ensure the safety operation of the cassette body.

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Table 1. System thermal hydraulic requirement

Client number	Flow rate (kg/s)	Max pressure loss (MPa)	Max heat load (kW)	Conductivity (μs/cm)	Inlet temperature (°C)
Klystron collector	66.67	0.6	44500	< 0.3	≤ 28
Klystron body	16.67	0.8	195	< 0.3	≤ 28
Circulator	5.56	0.8	100	< 0.3	≤ 28
Circulator load	27.78	1	210	< 0.3	≤ 28
Transmission line	22.22	1	500	< 0.3	≤ 28
Antenna	13.89	1.8	220	< 0.3	≤ 28

Francesco Edemetti, et al. (2019) performed the cooling system design of the Water-Cooled Lithium-Lead (WCLL) breeding zone adopting the ANSYS CFX codes. The research was conducted to study and compare four configurations, which rely on different arrangement of the stiffening plates, orientation of the cooling pipes. The author discussed the cooling capacity of the four configurations through thermal hydraulic analysis. The results laid the groundwork for the finalization of the WCLL blanket design, pointing out relevant thermal-hydraulic aspects.

From the review above, most of scholars only care about the thermal-hydraulic performance of components themselves, such as divertor, blanket and first wall, et al. but the external cooling system was ignored. In addition, some researchers of ITER focused on thermal hydraulic analysis of the component cooling water system by AFT Fathom, and Bin Guo only performed the design of coolant distribution system (CDS) rather than total CWS. Hence, the study presents a method based on the AFT Fathom code for the design of LHCD system and develops the thermal-hydraulic model to predict the behavior of the LHCD CWS. According to the calculation results, the CWS can supply cooling water with certain pressure, temperature and flow for the LHCD system, which can meet each client's requirements.

2. System Requirement and Process Design

The LHCD system has 6 clients including klystron collector, klystron body, circulator, circulator load, transmission line and antenna, and cooling requirements of each client are different. According to the requirement of clients, the CWS will provide cooling water with different pressure and temperature for the clients. As shown in table 1, the CWS requirements applied for the simulation are taken from the design basis, mainly based on the latest design parameters provided by the CRAFT organization. During the normal operation, approximately heat power of 5.67MW generated by the clients has to be removed. The inlet temperature of the clients should not exceed 28 °C. And the inlet pressure of antenna is required to be at least 1.8 MPa, which is the largest inlet pressure in all subsystem of CRAFT. Meanwhile, the required flow rate of the clients should be promised. In order to avoid corrosion of the components, the cooling water quality requirement is less than 0.3 μs/cm.

According to client requirement of LHCD system, the CWS is designed as three independent cooling loops because of the client pressure loss characteristic. In detail, klystron collector, klystron body, circulator are three clients in a loop A. Circulator load and transmission line are divided into an independent cooling loop B. Finally, an independent cooling loop C provides cooling water to antenna alone due to the larger inlet pressure demand. Meanwhile, each cooling loop has the same function so that the schematic of all cooling loops can be illustrated in Fig. 1 [5].

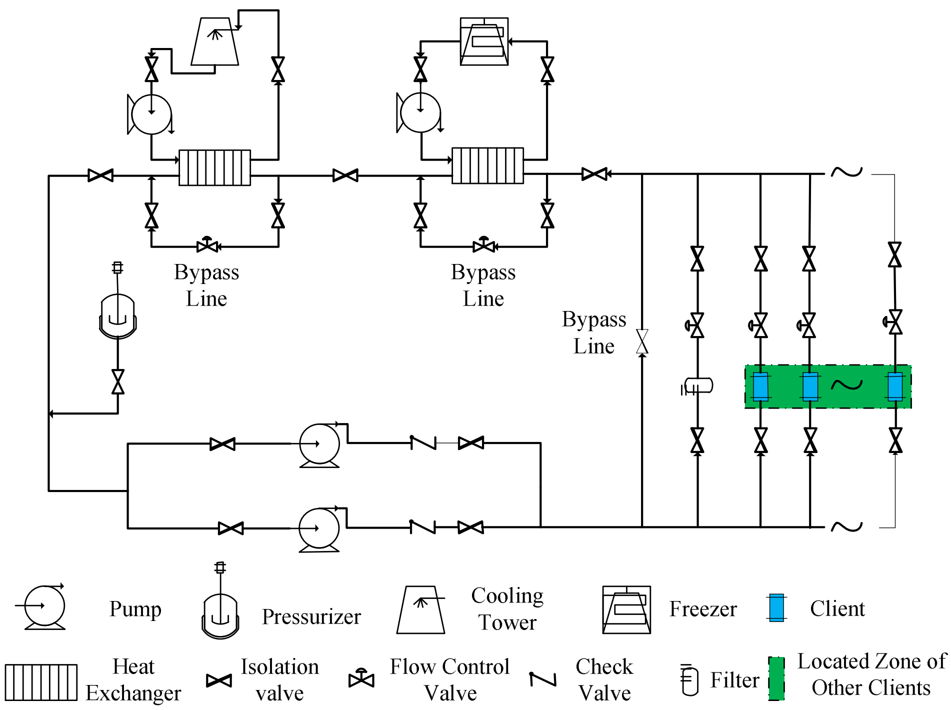

Fig. 1. Design schematic of cooling loop

To satisfy the requirements of different clients, some branches in parallel are designed to feed each client. The clients placed in the green zone can be replaced to other clients. Flow control valves are installed in the downstream of the clients so that the mass flow and inlet pressure of the clients can be controlled at the specific level. Isolation valves are located at the inlet and outlet of each client to isolate the clients when the clients are in maintenance. Bypass line of the clients is designed to balance the flow of clients. Similarly, the bypass line of the heat exchanger has the same function to satisfy the temperature requirement. To avoid lower conductivity than the requirement of clients, filter is placed in parallel of branches. Pressurizer is located at the upstream of the pumps to avoid coolant vaporizing, which will guarantee the system safety. The pumps are located in parallel to increase the reliability of the whole system.

3. Thermal Hydraulic Analysis Theory

To calculate thermal-hydraulic parameters, the Newton-Raphson method is employed in the AFT Fathom code. In the Newton-Raphson method, new values for each unknown are calculated based on the previous value and a correction that uses the first derivative of the function. Meanwhile, the flow rate and head are solved in an inner-outer loop algorithm, where the flow is assumed, the head loss is calculated consistent with that assumption, and the flow is updated according to the new pressure drop information. The Newton-Raphson method is employed to refine each successive solution, resulting in a sparse square matrix that is solved during each solution pass. Therefore, the system flow resistance can be obtained by this method to determine the pump size and fitting size. Then, the pressure distribution of the system is calculated again through the selected pump size and fitting size.

In the following discussion, subscripts denote values at junctions. Thus, P_i represents the pressure at junction i. Double subscripts denote values along pipes connecting two junctions. Thus, m_{ij} represents the mass flow rate in the pipe connecting junctions i and j [6], [7].

The total pressure change between junctions is given by the momentum equation in the form of the Bernoulli Equation [8], referring to (1):

$$P_1 + \frac{1}{2}\rho V_1^{\,2} + \rho g z_1 = P_2 + \frac{1}{2}\rho V_2^{\,2} + \rho g z_2 + \Delta P_f \tag{1}$$

The static and stagnation pressure are related as follows:

$$P_0 = P + \frac{1}{2}\rho V^2 \tag{2}$$

Substituting Equation (2) into (1)

$$P_{0,1} + \rho g z_1 = P_{0,2} + \rho g z_2 + \Delta P_f \tag{3}$$

Solving for the frictional pressure drop for a constant area pipes yields:

$$P_{0,i} - P_{0,j} + \rho g\left(z_i - z_j\right) = \Delta P_f \tag{4}$$

and the basic equation for pipe pressure drop due to friction can be expressed with the Darcy-Weisbach equation:

$$\Delta P_f = f\frac{L}{D}\left(\frac{1}{2}\rho V^2\right) \tag{5}$$

The definition of mass flow rate is:

$$\dot{m} = \rho A V \tag{6}$$

Combining Equations 3 and 6 and substituting for velocity, V, using Equation (6) gives the mass flow for each pipe:

$$\left(\frac{P_{0,i} - P_{0,j} + \rho g\left(z_i - z_j\right)}{R_{ij}^{"}}\right)^{1/2} = \dot{m}_{ij} \tag{7}$$

where $R_{ij}^{"}$ is the effective flow resistance in the pipe and the subscript *ij* refers to the pipe connecting junctions *i* and *j*.

$$R_{ij}^{"} = \left(\frac{f_{ij}L_{ij}}{D_{ij}} + K_{ij}\right)\frac{1}{2\rho A_{ij}^2} \tag{8}$$

Application of the law of mass conservation to each junction yields:

$$\sum_{j=1}^{n}\dot{m}_{ij} = 0 \tag{9}$$

Substituting Equation (7) into Equation (9) results in the equation to be applied to each junction *i*:

$$\sum_{j=1}^{n}\left(\frac{P_{0,i} - P_{0,j} + \rho g\left(z_i - z_j\right)}{R_{ij}^{"}}\right)^{1/2} = 0 \tag{10}$$

where *f* is the pressure loss coefficient, *Z* is the elevation, ρ is the fluid density, *g* is the gravitational acceleration, *L* is the pipe length, *D* is the pipe diameter and *V* is the fluid velocity.

In addition, the heat power generated by the clients was removed by the cooling water. Through the calculation of the AFT Fathom code, the temperature distribution of the cooling water can be obtained. Hence, it is necessary to apply an energy balance to the pipe yields, referring to the formula (11).

$$q_m c_p dT = q \tag{11}$$

where q_m is the mass flow, c_p is the specific heat capacity, dT is the temperature difference of the fluid, q is the heat power.

4. Thermal Hydraulic Analysis and Discussion

4.1 Model Description

The AFT Fathom code as the effective method is used to determine the fitting parameters and predict thermal hydraulic behaviors. Due to the similar structure of the three independent models, only the cooling loop A is given in the article as shown in Fig. 2 for example. And the number marked in the pipe indicates the pressure and temperature reference point. In the model, the clients of each cooling loop are modeled as heat exchange in red to remove heat power by the coolant flowing through it. The pressure drop data of client is entered into Fathom as a point from which Fathom extrapolates a quadratic resistance curve. The heat source value is also inputted in the model for each client. Similar to clients, the pressure loss model is established through quadratic resistance curve, which makes the heat exchange resistance vary with the variations of coolant flow. The outlet temperature of heat exchanger is given to meet the temperature requirements. The control valves in green are modelled as active control to keep the specific flow for clients. The isolation valves in gray are taken reference from the VELAN ball type datasheet. Piping length, bends, and other junctions are defined based on the system arrangement. In addition, in order to ensure the stable operation of the system, each cooling loop is equipped with a backup pump.

4.2 Pressure Profile

In an independent cooling water loop, the smaller the pressure drop value of the downstream control valve, the more critical the closed loop where the control valve is located. Fig.3 shows the pressure drop of the six control valves. It is obvious that the flow path of circulator, transmission and antenna have been calculated as the critical path to decide total pressure drop of cooling loop A, loop B and loop C. Therefore, the control valve at the downstream of circulator, transmission and antenna are set as the pressure drop control valve (PDCV). The other control valves are set as flow control valve (FCV). Table 2 shows the inlet pressure of each client comparing with the requirement pressure. It is obviously that inlet pressure is larger than the requirement pressure of each client, which means that it well meets the system requirement [9].

Fig.4 shows the pressure profile of each client according to the thermal hydraulic calculation results. Obviously, due to the flow loss, the overall pressure distribution shows a downward trend. And there is the maximum pressure of 20.553 bar in the cooling loop C, because the client of antenna requires the larger inlet pressure than the other clients. From the reference point 4 to the reference point 5, the pressure drop is consumed by the clients under its required flow, which give evidence to check whether the client requirement has been satisfied or not. The calculation results show that the designed CWS can well meet the requirement of clients.

4.3 Temperature Distribution

Fig.5, Fig.6 and Fig.7 respectively present the temperature distribution of the cooling loop A, B and C. The main heat exchanger can decrease the coolant temperature to 33 °C, and the CHW heat exchanger has the function of reducing coolant temperature to below 28 °C. As shown in the figures, the inlet temperature of all clients is not higher than 28 °C, which achieves the inlet temperature requirement. And the outlet temperature of the klystron collector achieves 42.98 °C, which is the highest outlet temperature among the all clients. In addition, due to the low heat power of the circulator load and the transmission line, the coolant temperature mixing on the return header is only 30.4 °C. Hence, the control valve in parallel with the main heat exchanger is opened to bypass the main heat exchanger. The CHW heat exchanger is fully capable of reducing the coolant temperature to the client inlet temperature requirement.

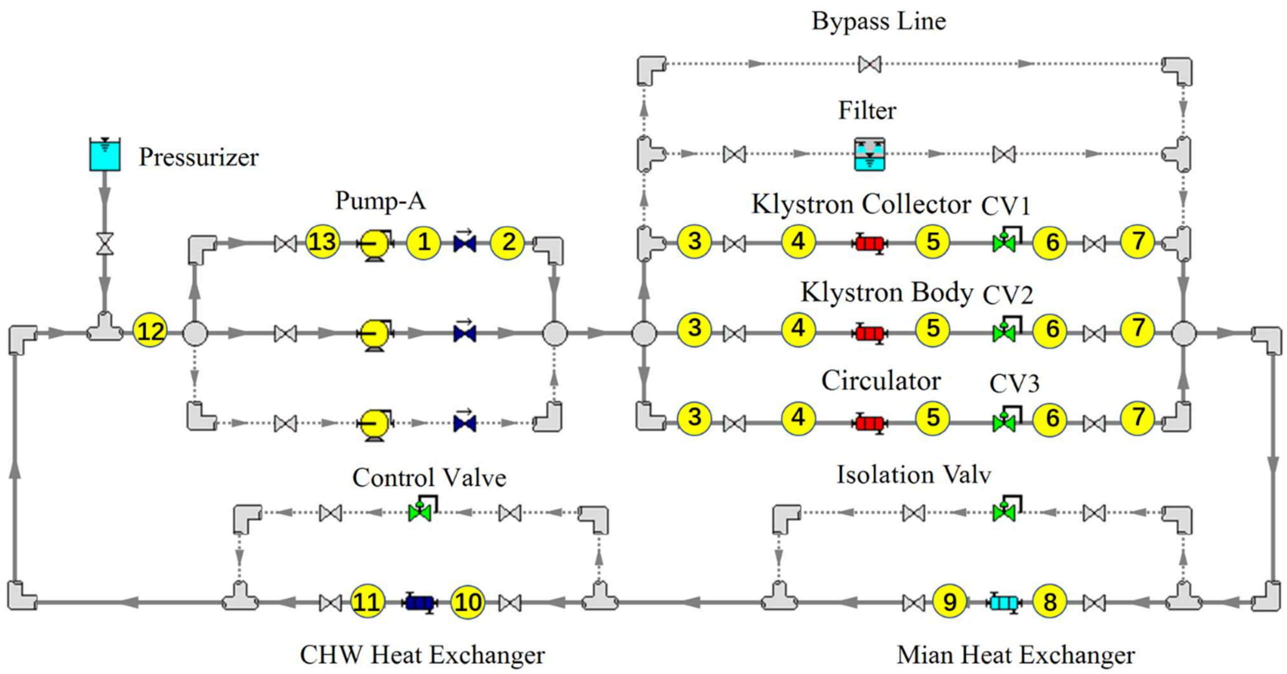

Fig. 2. Thermal hydraulic model of the LHCD cooling loop A

Table 2. Calculated inlet pressure comparing with the pressure requirement

Client	Inlet pressure (bar)	Pressure requirement (bar)
Klystron collector	10.339	6
Klystron body	10.366	8
Circulator	10.359	8
Circulator load	12.863	10
Transmission line	12.665	10
Antenna	20.399	18

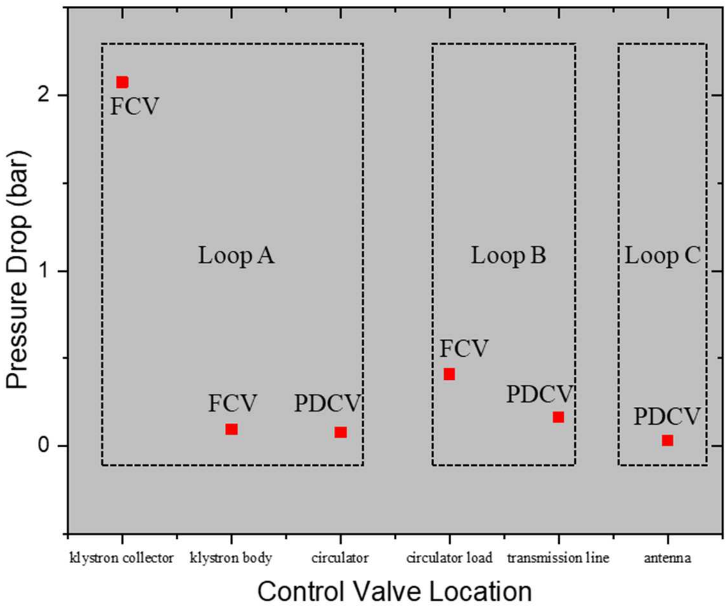

Fig. 3. Control valve pressure drop of the three independent cooling loops

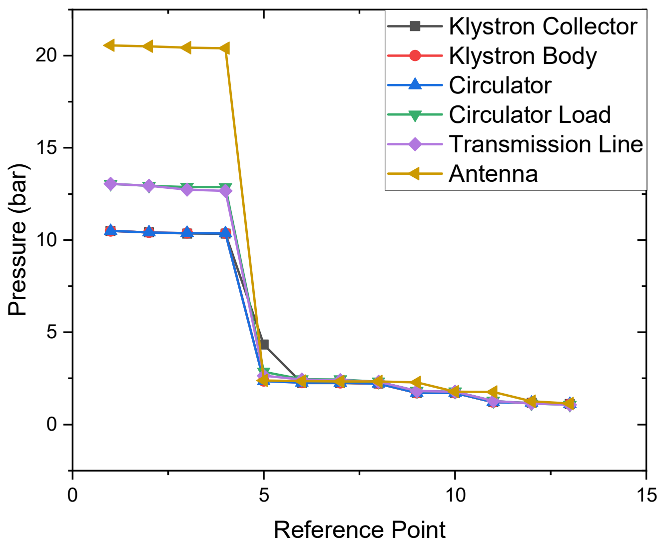

Fig. 4. Clients pressure profile

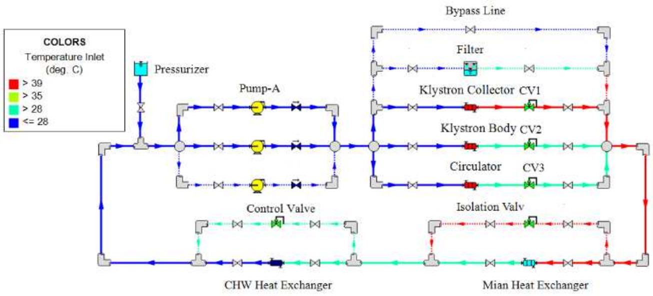

Fig. 5. Inlet temperature distribution of cooling loop A

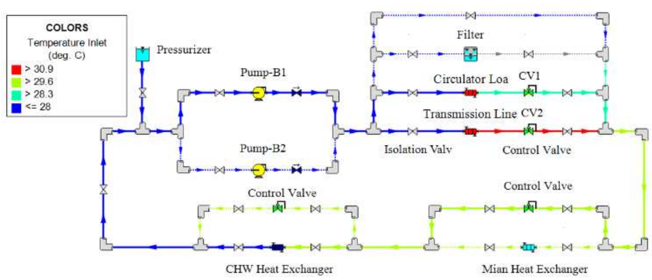

Fig. 6. Inlet temperature distribution of cooling loop B

Table 3. Pump minimum requirement.

Cooling loop	Mass Flow (kg/s)	dH (meter)	Total power (KW)	NPSHa (meter)
A	88.9	99.99	87.1	11.74
B	50.00	119.95	58.78	11.74
C	13.90	193.10	26.28	11.18

Table 4. Pump requirement.

Cooling loop	Mass Flow (kg/s)	dH (meter)	Total power (KW)	NPSHa (meter)
A	88.9	124.99	108.88	11.74
B	50.00	149.94	73.48	11.74
C	13.90	241.38	32.85	11.18

4.4 Fitting Size

Pump size is determined by the pressure loss of the most critical path. According to the calculation results, table 3 presents the minimum pump size of the three independent cooling loops. 20% margin on the minimum required pump head should be considered when specifying these pumps. Therefore, the required pump parameters are provided as table 4 [10].

Each pipe diameter in the system can be calculated according to the flow requirements of each client of the system. The formulation is presented as following. Q_m is the client required flow, where ρ is the density of the cooling water and v is the fluid velocity.

$$D = \sqrt{\frac{4Q_m}{\pi \rho v}} \tag{12}$$

According to relevant design standards, the reference flow rate of the CWS is 1.5-2.5 m/s. Fig.8, Fig.9 and Fig.10 respectively present the velocity distribution of the cooling loop A, B and C, and the diameter of the pipe is marked with purple font. From the figures, it is obviously that the cooling water velocity of three independent cooling loop is basically maintained at 1.5-2.5 m/s. The minimum velocity is 1.45 m/s. In the cooling loop B, the maximum velocity of the supply and return header reaches 2.84 m/s. Once the pipe size is increased, the flow rate is less than 1.5 m/s. Considering the pipe economic, the velocity of 2.84 m/s is acceptable. Meanwhile, the maximum pipe diameter is DN250 applied in the supply and return header of the cooling loop A. Due to the smallest requirement flow of circulator, the pipe size DN65 connecting the circulator is the smallest.

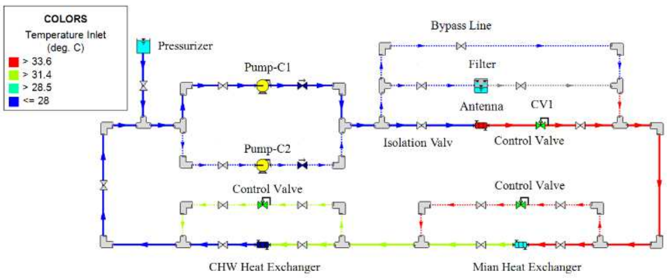

Fig. 7. Inlet temperature distribution of cooling loop C

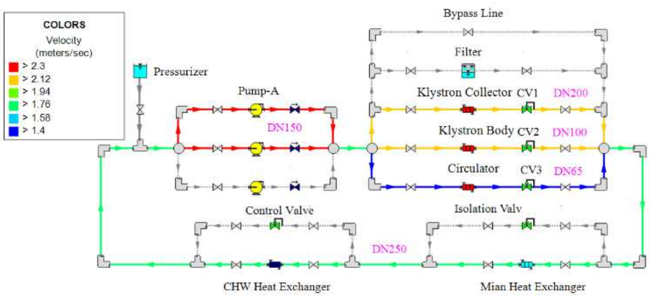

Fig. 8. Velocity distribution of cooling loop A

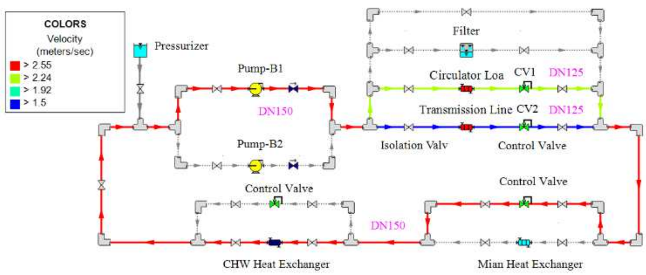

Fig. 9. Velocity distribution of cooling loop B

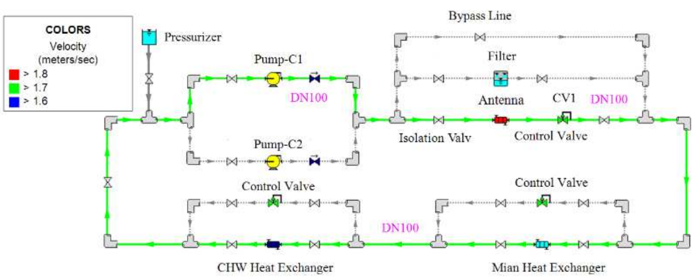

Fig. 10. Velocity distribution of cooling loop C

5. Conclusion

The AFT Fathom as a valid method was used to predict thermal hydraulic behavior of the LHCD cooling water system designed in this paper. The pressure profile, temperature distribution and fitting size were analyzed in detail, some specific results were also achieved. According to the calculation results, the inlet temperature and pressure well met the client requirement of the LHCD system. And the maximum inlet pressure was 20.553 bar occurred in the upstream of the antenna. In addition, the maximum temperature was 42.98 °C happened in the outlet of the klystron collector. Meanwhile, the fitting size was obtained through the calculation, including pump, pipe and control valve. The results obtained in this paper lay the foundation for the stable operation of the LHCD system and provide a good reference for the design of the CWS in the future.

Author Contributions

Weibao Li, Jiansheng Hu, Lei Yang, Bin Guo and Lili Zhu planned the scheme, initiated the project and suggested the research; Weibao Li Jinxuan Zhou, Zhe Liu, Lidong Yao and Qianglin Xu conducted the research and analyzed the results; Weibao Li developed the mathematical modeling and examined the theory validation. The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed and approved the final version of the manuscript.

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Conflict of Interest

Weibao Li, Jiansheng Hu, Lei Yang, Bin Guo, Lili Zhu, Jinxuan Zhou, Zhe Liu, Lidong Yao and Qianglin Xu declared no potential conflicts of interest with respect to the research, authorship and publication of this article.

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Nomenclature

A	Cross-section area [m^2]	t	Temperature [°C]
L	Length of pipe [m]	m_{ij}	Mass flow rate [kg/s]
f	Pressure loss coefficient	Z	The elevation [m]
ρ	Density [m^3/kg]	V	Fluid velocity [m/s]
q_m	Mass flow rate [kg/s]	c_p	Specific heat capacity[J/kg* °C]
q	Thermal power[W]	G	Gravitational acceleration [s^2/m]
P	Pressure [Pa]		

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