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Design and Analysis of Heat Exchanger Using Sea Water Heat Source for Cooling

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ABSTRACT

Purpose: The temperature in Seoul has risen 3 times more than the average global temperature increase for the past 100 years. Today, summer starts 15 days earlier than the early 20th century and is 32 days longer. This tendency causes rapid increase of cooling energy demand. Following this effect, seawater heat resources are to be used as an countermeasure for global warming. Incheon Port near the Western Sea has the lowest water temperature in the winter in South Korea in which it is suitable to use seawater cold heat resources. **Method:** The cold heat resource is gained from seawater when the water temperature is the lowest in the winter time and saved in a seasonal thermal storage. This can be used as cold heat resource in the summer time. A heat exchanger is essential to gain seawater cold energy. Due to this necessity, sea water heat resource heat exchangers are modeled by heat transfer equations and the fluid characteristics are analyzed. Also, a CFD (computational fluid dynamics) program is used to conduct simulation on the fluid characteristics of heat exchangers. The analyzed data of deducted from this process are comprehensively analyzed and discussed. **Result**: Regarding the performance of the heat exchanger, the heat exchanger was operated following the prediction within the range of heat transfer rate of minimum 3.3KW to maximum 33.6KW per device. In the temperature change analysis of the heat exchanger, fluid analysis by heat transfer equations almost corresponded to the temperature change by CFD simulation. Therefore, it is considered that the results of this study can be used as design data of heat exchangers.

KEYWORD

Sea-water heat source
Heat exchanger
Seasonal Thermal Storage
Computational fluid dynamics

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1. Introduction

1.1. Research Background and Purpose

The average air temperature of the globe has risen 0.74℃ over the last century. According to the record of the Seoul Me-tropolis[1] "Annual average air temperature in Seoul has risen 2.4℃ over the last one hundred years (1908-2007), which is 3 times higher than the average rise of global temperature. The summer[2]came 15 days earlier in 1998-2007 than in 1908-1927 and summer season was 32 days longer (from 92 days to 124 days)." The operation hours and energy consumption of cooling are expected to increase.

Many studies have been conducted to utilize sea water heat source as a countermeasure for global warming. Currently sea water heat source is used for renewable energy power as heat-ing and cooling using heat pump units. Existing heating and cooling system based on heat pump unit instantly consumes acquired thermal source as soon as it is acquired. It is necessa-ry to study on design and analysis of heat exchanger of seawater as cooling source.

This study is aimed to design and analyze heat exchanger of sea water heat source for cooling, and provide the basic data base for heat exchanger of sea-water.

1.2. Research Scope and Method

Design and flow analysis of the heat exchanger are an essential element for cooling energy acquisition from sea water heat source. In this study, the scope of research is set to designing the heat exchanger to acquire sea water heat source and analyze the operational characteristics of the heat exchanger. To take advantage of seawater heat source, it is necessary to analyze the conditions of the west, south and east sea of Korea. The west sea has relatively advantageous conditions for seawater heat source for cooling. The average water temperature of the west sea is 9～6℃ lower than that of the south and east sea during February. The average annual sea water temperature of Incheon Port near the west sea is 1.4℃, which is the lowest among any other west area in Korea.

This study is following as below. First, the studies on seawater heat source, a heat exchanger, and a seasonal thermal storage were reviewed, and the current status and heat transfer theory were examined. Second, a heat exchanger was modeled to use in this study and its operational conditions were designed. Third, this study analyzed the performance of heat exchanger by operating condition using heat transfer formulas. Fourth, to understand the

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1) In meteorology, seasons are divided by temperature change. The beginning day of summer is defined as the first day of 9 consecutive days whose daily temperature rises over 20℃ and remain above it.

2) Seoul Air Quality Information /Climate Change: cleanair.seoul.go.kr/

heat exchange phenomenon of sea water cold thermal source, this study analyzed the flow characteristics of a heat exchanger using Computational Fluid Dynamics (CFD). Fifth, comparison and analysis was made between the results from a theoretical prediction and those of CFD simulations. We reviewed on the flow characteristics and performance of a heat exchanger.

1.3. Literature Review

Looking into research literature of this study, there are about 15 studies examining heating and cooling system using heat pump unit; about 4 studies on heat exchanger and acquisition of seawater heat source; about 10 studies concerning applications and trends. Following are main contents of those studies.

Kim, Gi-cheol (2006) buried synthetic resin pipes at a seawater supply dock in the National Fisheries Research & Development Institute (Pusan) and studied a direct cooling system with cold air supplied. Yoon, Hyeong-gi (2006) summarized a seasonal thermal storage in his study on the technology trend for solar system. Park, Geun-woo (2007) proposed a technology to use heat acquired from ground water for building heating and cooling by heat pump units. And Kim, Myeong-rae et al. (2009) suggested a method of acquiring sea water cold thermal source, storing it during winter, and using it as cold thermal source for summer. Kim, Han-ji (2014) proposed a way to improve the performance of overall heat transmission of an underwater heat exchanger, and measured and analyzed heat transmission in experimental tests.

As known above, most of the domestic studies on seawater heat source cooling system have focused on cooling method of using heat pump units. And that is characterized with simultaneous acquisition and releases of heat. However, a few studies have been carried out on the concept of a seasonal thermal storage that cold heat source from seawater during winter is stored and it is used in next summer. In this study, the heat exchanger for sea water heat source, which is stored in a seasonal thermal storage, and analyzed the flow characteristics of the system.

1.4. Sea water Heat Source and Velocity of Flow

The western sea of Korea is rather shallow, 44m in average depth of sea water, and half closed. The sea water temperature of west is lower than the east and south sea in the winter. Therefore, it is very sensitive to climate changes (Lee, Choong-il et al., 2007). According to the Korea Hydrographic and Oceanographic Administration(Ocean Information, Issue, February, 2015), the average sea water temperature of Inchon Port (2005-2014) is 2.1 1℃ in Feb., which is lower than Pusan (11.07℃ in the southern sea) and Sokcho (6.64℃ in the eastern sea). The sea water temperature of Inchon Port drops to the lowest level in mid February and rise to the highest in mid September, which is cyclical phenomenon. Therefore, the neighborhood of Inchon Port is considered relatively excellent zone for cold seawater thermal source.

The velocity of ocean current affects the performance of a heat exchanger constructed under sea. Inchon Port has considerable sea level change. According to Ocean Information(Feb., 2015), the strongest velocity of flood tide, ebb tide, tidal residual flow in Inchon Port are 80.2cm/s, 74.2cm/s, and 4.7cm/s.[3)]

1.5. Heat Exchanger

Heat transfer takes place only when there is temperature gap, the heat always transfers from high to low temperature. A heat exchanger is a device to exchange heat without mixing two or more fluids with different temperatures.

This study supposed that a heat exchanger is made of circular pipe; the fluid flow is fully developed; and a heat transfer takes place only between two fluids. In addition, this study ignores changes of kinetic energy and potential energy. Under this assumptions, the energy balance between the two fluids of a heat exchanger can be expressed as follows.

$$d\dot{q} = -\dot{m}_h cp_h dT_h = \dot{m}_c cp_c dT_c \qquad (1)$$

Here, $\dot{m}_h$, $\dot{m}_c$: mass flow rate of fluid at high temperature and low temperature

cp_h, cp_c: specific heat of fluid at high temperature and low temperature

Here, heat transfer equation is $d\dot{q} = U(T_h - T_c)dA$

U: the coefficient of overall heat transfer.

In this equation, heat transfer rate $\dot{q}$ and ΔT_{lm} can be expressed as follows.

$$\dot{q} = UA_s \Delta T_{lm} [W] \qquad (2)$$

Here, A_s: the surface area of a heat exchanger pipe

ΔT_{lm} : logarithmic mean temperature difference(LMTD).

$$\Delta T_{lm} = (\Delta T_1 - \Delta T_2)/\ln(T_1/T_2) \qquad (3)$$

$\Delta T_1 (= T_i - T_c)$: temperature gap at the inlet of a heat exchanger

$\Delta T_2 (= T_e - T_c)$ is temperature gap at the outlet.

T_i : the inlet temperature, T_e : the outlet temperature

T_c is temperature of the outer surface.

To know the length of pipe of a heat exchanger, surface area of pipe (of a heat exchanger) $A_s = \dot{q}/U\Delta T_{lm}$ can be calculated in Equation (2) as below.

$$L = A_s/\pi D [m] \qquad (4)$$

3) The strongest flood tide current: the maximum current at rising tide / The strongest ebb tide current: the maximum current at ebb tide. / Tidal residual flow: average velocity of flow of seawater by tidal phenomenon

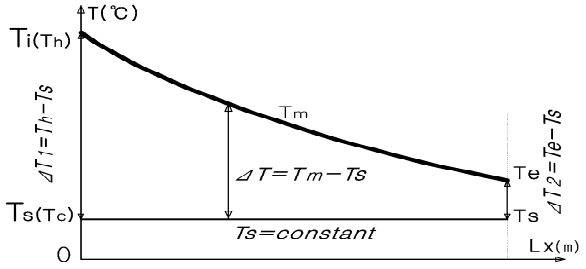

Fig. 1. The Variation of the mean fluid Temperature along the Tube

Fig. 1 shows the operating mechanism inside of a heat exchanger. Supposing that the temperature of its outer surface (T_s:seawater) is constant, it is known from Fig. 1 that the av-erage temperature of fluid T_m decreases in a flow-ing direction due to the heat transfer.

When $dA = Pdx$, $P = \pi D$ and $dT_m = -d(Ts - T_m)$ leads to this equation.

$$\frac{d(T_s - T_m)}{T_s - T_m} = -\frac{UP}{\dot{m}\,cp}dx \qquad (5)$$

When integrating the function from $x = 0(tubeinlet)$ to $x = L(tube\,outlte\,where\,T_m = T_e)$ the following equation (6) comes out.

$$\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{UA_s}{\dot{m}\,cp} \qquad (6)$$

Here, $A_s = PL$ is the surface area of pipe and when the outlet temperature T_e was solved, the average fluid temperature at the outlet can be expressed like below.

$$T_e = T_s - (T_s - T_i)\exp(-UA_s/\dot{m}\,cp) \qquad (7)$$

When changing $A_s = PL$ to Px, the average temperature of fluid $T_{m(x)}$ at the distance x from a heat exchanger inlet can be derived.

$$T_{m(x)} = T_c - (T_c - T_i)\exp(-UPx/\dot{m}\,cp) \qquad (8)$$

Then, we can see the temperature at a fluid point inside a heat exchanger decreases exponentially ($\exp(-UA_s/\dot{m}\,cp)$)as it is longer from the inlet. Here, non-dimensional parameter $UA_s/\dot{m}cp$ is heat transfer coefficient and it is used as an index for the performance of heat transfer system. It is called NTU (Number of Transfer Unit) (Kim, Yu, translated, 2004).

1.6. Seasonal Thermal Storage

A large-scale seasonal thermal storage stores hot water in a thermal storage tank, heated by solar heat during summer and then releases the heat in winter. This idea of seasonal thermal storage began from a plan to push solar heat load ratio from 15% (day thermal storage) up to 80% at seasonal thermal storage. In 1985, a Sweden company Nykvarn constructed a seasonal thermal storage facility of 1,500㎥. There is operating 10 local thermal plants in Germany (Yoon, Hyeong-gi, 2006). The capacity of thermal storage tank is proportionate to its volume and so a bigger thermal storage tank has less heat loss in principle. The operating efficiency of German seasonal thermal storages is about 80-95% and capacity is 2,750㎥～12,000㎥. It can be used by 30 to 40 years (T. Schmidt et al, 2004).

2. Design and Analysis of Heat Exchanger of Sea water Heat Source for Cooling

This study designed a heat exchanger to acquire sea water thermal source for cooling, the amount of heat calculated theoretically, and estimated the temperature.

2.1. Sea water Heat Source

Because sea water heat source is sensible heat of seawater, it is directly affected by seawater temperature. Table 1 shows the monthly average water temperature of 3 cities in Korea during October. The temperature is lowest in February and highest in August and September in a year.

The sea water temperature of Incheon during February was 2.1℃, which was lower than Pusan (10.9℃) and Sokcho (6.9℃). It is known that Incheon is the most suitable area for seawater cold thermal source. Table 2 summarizes 8-day minimum water temperature of Inchon Port over the last 6 years(2010-2015). The lowest average water temperature of 8 days was from February 6th through 13th in 2011, which was 0.4℃ and the highest was 2.5℃ in 2015. 6-year average water temperature analysis shows that the optimum acquisition period of cold thermal source from seawater is February 6th to 13th, in a year.

Table 1. Average Monthly Sea Temperature(℃)
source : Korea Hydrographic & Oceanographic Administration, successive years(2005-2014)

month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Incheon	2.3	2.1	4.8	9.1	14.4	19.3	22.5	24.8	24.1	19.7	13.2	8.2
Busan	11.1	10.9	11.8	13.3	14.9	17.4	19.2	22.4	23.2	20.6	16.9	13.4
Sokcho	8.1	6.9	7.2	9.0	11.5	15.1	18.4	21.6	20.9	18.8	14..9	10.9

Table 2. Sea Water Temperature(℃) of the Successive Years in Incheon source: Korea Hydrographic & Oceanographic Administration

days	6Feb	7Feb	8Feb	9Feb	10Feb	11Feb	12Feb	13Feb	Avg
2015	2.8	2.9	2.7	2.3	2.2	2.3	2.3	2.2	2.5
2014	2.2	2.2	2.3	2.3	2.3	2.3	2.6	2.6	2.3

2013	1.7	1.6	1.2	0.9	0.9	0.7	0.7	0.8	1.1
2012	0.9	0.9	0.6	0.6	0.6	0.7	0.8	1.0	0.8
2011	0	0	0.3	0.5	0.6	0.6	0.5	0.5	0.4
2010	1.2	1.2	1.3	1.4	1.5	1.6	1.6	1.7	1.4
Avg	1.5	1.5	1.4	1.3	1.4	1.4	1.4	1.5	1.4

2.2. Heat Exchanger System

1) Heat Exchanger

For the material of a heat exchanger pipe, this study selected a High Density Polyethylene(HDPE), which is durable (so that it can work under seawater properly), cheap, and easily available in market. Its length was 100m and 32mm in diameter. A heat exchanger of seawater heat source for cooling was designed using heat transfer formulas (Equation 1 to 8)

The heat transfer rate of a heat exchanger had to be determined depending on a seasonal thermal storage capacity. The operating hours of a heat exchanger was according to the acquisition of seawater heat source. The heat exchanger with seawater heat source was set to 8 days as shown in Table 2. In this study, 16 heat exchangers separately. Heat transfer rate of each heat exchanger is about 8.5RT. Equation (4) was used to calculate the proper pipe length of a heat exchanger for target heat transfer rate. The length was estimated to be 361.4m (pipe). This pipe was designed to fit underwater evironment: 2.2m in diameter, 0.08m at pitch interval, 52.3 times of coiling in spiral pattern, and 4.2m in height. The coils of a seawater heat exchanger were fixed to a rigid stainless frame so that it can not cause problem during transportation, installation and construction.

2) System Composition

The seasonal thermal storage capacity was 2,500㎥ and structured with reinforced concrete (0.3m in thickness). It is similar to the smallest German thermal storage tank (2,750㎥～12,000㎥). The inside of the thermal storage tank was finished with stainless plates and its exterior envelope was made of thermal insulation boards (0.3m in thickness) treated with waterproof finish and buried.

Fig. 2 shows the schematic diagram of the system developed in this study. This system consists of a thermal storage tank installed on the ground, and 16 heat exchangers installed under seawater, connection pipes, a circulating pump necessary for circulation, and control system. The heat exchangers were installed combining with buoys and anchors so that their tops can submerge 1 meter under water.

The heat exchangers, water filled in the thermal storage tank was used. The beginning temperature of seasonal thermal storage tank

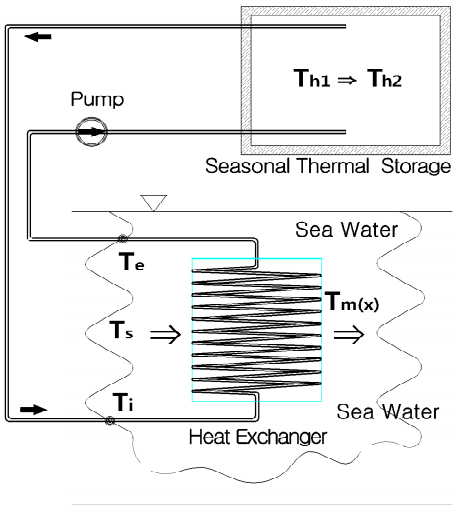

Fig. 2. Schematic Diagram of Seawater Thermal Heat Exchanger and Seasonal Thermal Storage System

temperature(T_{h1}) and heat exchange was 25℃. That was the temperature of water in a thermal storage tank heated after having been used as cool thermal source previous summer. The seawater average temperature(T_s) for 8 days (the period of the lowest average water temperature in a year over last 6 years 6 as seen in Table 2: February 6th to 13th) was 0.4℃～2.5℃. The target temperature of final thermal storage(T_{h2})was set to 5℃, which is higher than the average water temperature of 2015 (2.5℃).

2.3. Analysis of Heat Exchanger

(1) Input Conditions

There were several options of pipe thickness (2mm, 3mm, 5mm). The pipe material of the designed heat exchanger is HDPE and its properties are shown in Table 3. In this study, the thickness of HDPE pipe was 3mm chosen in consideration of tolerable pressure and heat transmission coefficient.

Table 3. Specification of Heat Exchanger's Pipe

Diameter mm	Inner Diam mm	Thickness mm	Weight kg/m	Thermal Conductivity k : W/m℃	Note
32	32	3	0.275	0.4	KSM-3408-2

To analyze the operating characteristics of the heat exchanger, the velocity of flow, fluid temperature, and overall heat transfer coefficient are like in Table 4. Ti is the temperature of water flowing into a heat exchanger and Te is the temperature of cold water flowing out a heat exchanger (see Figure 2).

The inlet temperature (T_i) of a heat exchanger is equal (T_i=T_{h1})to that of the thermal storage tank (T_{h1}). The outlet temperature

(T_e)of a heat exchanger is equal (T_e=T_{h2}) to thermal storage tank temperature (T_{h2}), which is the target temperature at the final stage of heat exchange. Here, coefficient f overall heat transfer system (111 W/m^2·℃) was quoted from the study by Kim, Han-ji (2014, pp. 38). The velocity of the flow was 0.5m/s.

Table 4. Condition of Heat Exchanger

Velocity of Fluid v (m/s)	In Flow's Temp. T_i (℃)	Exit Temp. T_e (℃)	Sea Temp. T_s (℃)	Overall Heat Transfer Coefficient U (W/m^2·℃)
0.5	25	5	3	111

(2) Method for Analysis

Fig. 3 was the thermal storage temperature graph of the designed heat exchanger system of seawater heat source. As seen here, under constant seawater temperature(T_s), the thermal storage tank temperature decreases from T_{h1} to T_{h2} along the exponential line. As the inlet temperature decreases, heat transfer rate($\dot{q}$)also decreases exponentially. The temperature of the thermal storage tank keeps dropping down to heat exchanger outlet temperature(T_e). Because the inflow temperature (T_i) also lowers exponentially, this study divided into 3 sections as seen in Fig. 3. The analysis of heat exchanger was conducted on three points(T_i =25℃, 16℃, 7℃) of temperature, T_i ,which are equivalent to the beginning and middle and end period of sea water cold thermal source. Among 16 heat exchangers, this analysis was carried out on one heat exchanger.

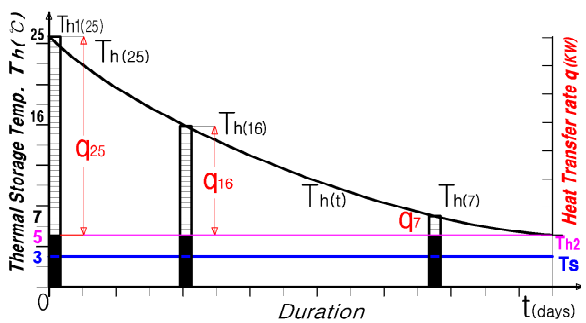

Fig. 3. Variable Heat Rate(q) with Thermal Storage Temperature(T_h) of the Heat Exchanger

2.4. Analysis of Flow Characteristics

As for the analysis conditions of a heat exchanger, assumption was made that fluid running through the pipe is fully developed. The flow analysis was followed Equation (2), (3), and (4). The estimation was made by substituting proposed values in Table 3 and Table 4.

(1) Inside Velocity of Heat Exchanger

To decide proper velocity of flow for the performance of the heat exchanger, each heat transfer rate($\dot{q}$) was calculated for the velocity of flow, 0.25m/s, 0.5m/s, and 1m/s. The surface area of a heat transfer, pipe length, logarithmic mean temperature difference (LMTD), and number of transfer unit (NTU) were calculated. The result are in Table 5.

Table 5. Various Parameters of Heat Exchanger

Velocity of Fluid V (m/s)	Heat Transfer Rate q (W)	Area of Heat Transfer As (m^2)	Length of Pipe L (m)	LMTD ΔT_{lm}(℃)	NTU
0.25	-16,809	18.2	180.9	-8.34	2.4
0.5	-33,618	36.3	361.4	-8.34	2.4
1.0	-67,237	72.6	722.7	-8.34	2.4

Comparing the result, it is known that heat transfer rate, surface area of a heat transfer, and pipe length proportionally increased twice as velocity of flow increased. This proportionate increase of heat transfer rate and pipe length to velocity of flow leads to disadvantageous, because the energy consumption of circulating pump is proportional to the square of velocity of flow. Therefore, when the velocity of flow of a heat exchanger is 0.5m/s, it is more advantageous than 1,0m/s in terms of operational power cost and price of a heat exchanger. When velocity of flow is 0.25m/s, liquid flow is too slow in general, so heat transfer rate($\dot{q}$) gets smaller and operating hours get longer.

In Table 5, when velocity of flow is 0.5m/s, heat transfer rate($\dot{q}$) is 33,618W and it is equal to 8.5RT in terms of refrigeration tonnage. The reason why the pipe length of a heat exchanger prolongs to 361.4m is that the heat conductivity (K) of HDPE pipe is lower than that of metal.

Table 6. Performance of Heat Exchanger

In Flow's Temp. T_i (℃)	Heat Transfer Rate q (W)	Area of Heat Transfer As (m^2)	Length of Pipe L (m)	LMTD ΔT_{lm}(℃)	NTU
25	33,618	36.3	361.4	-8.34	2.4
16	18,537	24.8	282.8	-5.88	1.87
7	3,380	10.5	105.0	-2.89	0.69

(2) Heat Transfer Rate of Heat Exchanger

Table 6 shows the performance of the designed heat exchanger when inflow temperature T_i of the heat exchanger is 25℃, 16℃, and 7℃. When a heat exchanger keeps operating, the temperature of a thermal storage tank (T_h) decreases exponentially. Along with the temperature of a thermal storage tank (T_h), the inlet temperature (T_i) of a heat exchanger also decreases, narrowing the gap (T_i-T_s) between the temperature of a heat exchanger (T_i) and that of

seawater (T_s). As inflow temperature decreases, the heat transfer rate of a heat exchanger sharply deceases, too. As seen in Table 6, at T_i=25℃, heat transfer rate was 33,618W, 18,537W at T_i=16℃, and 3,380W at T_i=7℃. The results of this analysis showed that the operating characteristics of a heat exchanger are much affected by the gap(T_i-T_s) of inlet temperature and outlet temperature.

Using Equation (8), the average temperature $T_{m(x)}$ at the point L(x) in a heat exchanger pipe was calculated under the different conditions of inlet temperature T_i =25℃(Case 1), 16℃(Case 2), and 7℃(Case 3). And the changed temperature gap △T was analyzed and summarized in Table 7.

Temperature gap (△T) between the average temperature ($T_{m(x)}$) at 180˚, 360˚, and 720˚ coil was increased. It turned out that the temperature gap (△T) at the random point L(x): 3.45m(180˚), L(x): 6.91m(360˚), L(x): 13.82m(720˚) was increased in proportion to the length of the pipe. This phenomenon was supported the explanation by Kim, Cheo-loo, translated (2008) that "As for turbulent flow, fluid becomes fully developed at the first middle point of a coiled pipe and inlet area is moistly ignored in engineering calculation."

Table 7. Estimate Mixing Temperature at the Heat Exchanger Pipe Section

Case 1 T_i =25℃	L(x) 3.45m 180˚	L(x) 6.91m 360˚	L(x) 13.82m 720˚
$T_{m(x)}$ ℃	24.05	24.01	23.07
△T ℃	0.50	0.99	1.93
Case 2 T_i =16℃	L(x) 3.45m 180˚	L(x) 6.91m 360˚	L(x) 13.82m 720˚
$T_{m(x)}$ ℃	15.71	15.42	14.86
△T ℃	0.29	0.58	1.14
Case 3 T_i =7℃	L(x) 3.45m 180˚	L(x) 6.91m 360˚	L(x) 13.82m 720˚
$T_{m(x)}$ ℃	6.91	6.82	6.51
△T ℃	0.09	0.178	0.349

3. CFD Simulation Analysis

3.1. Conditions for simulation

To confirm the temperature change by heat transmission on the surface of heat exchanger pipe, 3D CFD simulation analysis was carried out. Commercial code STAR-CCM+[11] was used to secure the reliability of numerical analysis. The Inner flow of a heat exchanger is turbulent flow. To minimize the impact by the inlet, fully developed laminar flow was used as inlet flow. It is in Equation (9).

$$v_z = 2\,V_{avg}\left(1-\frac{r^2}{R^2}\right) \qquad (9)$$

Here, v_z : velocity to circular pipe

V_{avg} : average velocity of flow,

R: radius of a heat exchanger

r : means location.

For boundary condition, inlet was set as velocity profile like in Fig. 4, while outlet condition was set as constant pressure. Constant heat transmission(111 W/m2·℃) was applied to the surface of a heat exchanger like in Table 4.

The length of one heat exchanger coil was 361.4m. If 3D thermal flow analysis was conducted on the whole coil of one heat exchanger, it could consume computing resource excessively. This study modeled only a certain length from the inlet and did calculation it.

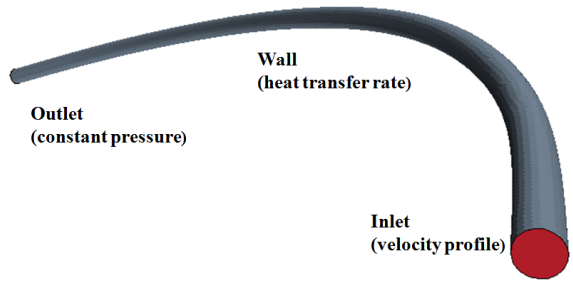

Fig. 4. 90° of Heat Exchanger

3.2. Reliability Test for CFD of Heat Exchanger

In general for CFD, the number of cell increases, accuracy of result increases. But it requires drastic consumption of computing resources on memory. It is very important to make a right division on the number of cell. Cell-dependent simulation enabled a right decision on grid number.

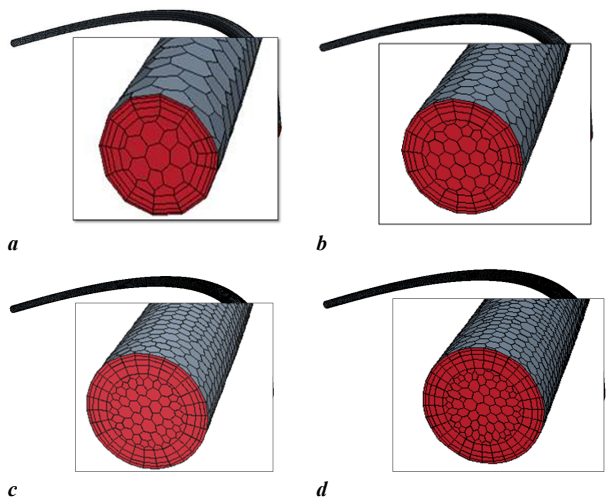

Fig. 5. Meshes of (a)Coarse, (b)Base, (c)Refine 1 and (d)Refine 2

First, as seen in Fig. 4, thermal flow simulation under 4

conditions. was conducted on the length of a heat exchanger at 90˚ while doubling cell number. Fig. 5 shows magnified 4 types with different number of cells. The results are as shown in Table 8.

As the cell number increases, the pressure drop per unit length (CP/L) decreases. Here, $CP=(p_i-p_{m(x)})/(0.5\rho V_{avg}^2)$ is pressure coefficient, which is the ratio of dynamic pressure to pressure gap between the inlet and the outlet.

Comparing the relative error based on pressure drop per unit length (CP/L) for the 4 cases in Table 8. It is known that relative error is less than 1% when more than 78,100 grids are used. Therefore, this study used all the cases whose cell size is 78,100 for calculation. To determine the length of coil, this study selected the heat exchanger from 90˚ to 720˚ by 90˚ (2 rotations). In total, 8 cases of thermal flow were simulated.

Table 8. Results of Mesh Dependency Test CFD

No. of Cells	V_{avg} m/s	CP/L	ΔT (℃) $T_i-T_{m(x)}$	CP/L Relative Error(%)	Comp. Time [min]
15,000	0.5209	0.64154	0.27	0.29	15
39,000	0.5098	0.62525	0.27	2.26	30
78,100	0.5044	0.63598	0.27	0.58	55
112,000	0.5033	0.63968	0.27	0.0	75

Fig. 6 shows the cases of simulated forms of a heat exchanger. The conditions for simulation are the same as used in Fig. 4 and Fig. 5.

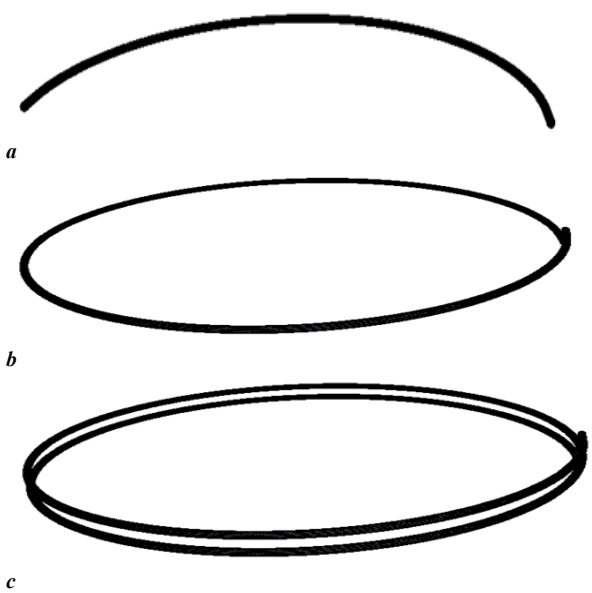

Fig. 6. Computational Domains: (a) 180°, (b) 360°, and (c) 720°

Fig. 7 shows that as angle increases and the length of coil prolongs, the impact by the inlet condition decreased. Eventually, converges on a certain value. At 720˚, no change happens. Therefore, it is known that if this length is used, the performance of overall heat exchanger can be estimated. Therefore, following evaluation of the performance of a heat exchanger was conducted at the point of 720˚(2 rotations) and it was applied to the whole length.

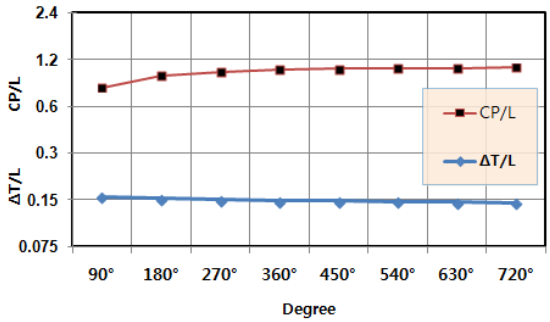

Fig. 7. Pressure and Temperature Drop According to Degree

As seen in Table 9, the number of grid linearly increases as the angle of a heat exchanger increases and so does the temperature of the inlet and the outlet. Temperature drop per unit length ($\Delta T/L$) converges on a certain value as length increases. Table 9 is the results of simulation implemented at 8 points under case 1 conditions (standard velocity of flow is 0.5m/s; inflow temperature(T_i) is 25℃ while rotating a heat exchanger coil from 90˚ to 720˚by 90˚.

Table 9. Results of Variations According to Degree(Length) with CFD Simulations

Deg.	V_{avg} m/s	CP/L	ΔT (℃) $T_i-T_{m(x)}$	$\Delta T/L$ ℃/m	No. of Cells	Comp. Time[min]
90˚	0.5044	0.63598	0.27	0.1563	81,000	55
180˚	0.5043	0.80030	0.53	0.1534	162,000	100
270˚	0.5043	0.85483	0.78	0.1505	256,000	150
360˚	0.5043	0.88256	1.02	0.1476	328,000	200
450˚	0.5047	0.89840	1.27	0.1470	410,000	250
540˚	0.5043	0.90982	1.51	0.1456	496,000	300
630˚	0.5046	0.91763	1.75	0.1447	579,000	350
720˚	0.5048	0.92328	1.98	0.1432	661,000	400

3.3. Simulation by Temperature

To confirm the performance of the heat exchanger, this study conducted 3D thermal flow analysis under the conditions as seen in Table 7. For inflow temperature, simulation was run at T_i=25℃, 16℃, 7℃ as defined in 3.3 (Conditions and Method for Analysis of Heat Exchanger). Here, the average velocity of flow was set to 0.5m/s, but since the velocity of flow was set to the center of the

grid, a slight error (0.6%) existed in the average velocity of flow. Table 10 demonstrated that temperature gaps (ΔT) of a heat exchanger at 720 ˚ are quite similar to the designed estimations at 3 points in Table 7.

Table 10. Temperature and Pressure Drop According to Inlet Temperature with CFD Simulations

T_i (℃)	$T_{m(x)}$ (℃)	V_{avg} m/s	Δp N/m²	ΔT (℃) $T_i - T_{m(x)}$	$\Delta T/L$ ℃/m
25	23.11	0.4973	1462.6	1.89	0.1367
16	14.88	0.4973	1467.2	1.12	0.0810
7	6.656	0.4973	1471.9	0.344	0.0248

Necessary power for a heat exchanger can be gained from total pressure calculated in CFD simulation. The power force ($W = \Delta p(\dot{m}/\rho)$ of a pump, which is necessary to overcome flow resistance related to pressure drop of a heat exchanger ($W = \Delta p(\dot{m}/\rho)$ can be derived from substituting Δp in Table 10: like 588W(at 7℃), 586W(at 16℃), and 584W(at 25℃). Using them, the pump power for the overall heat exchanger system can be computed.

4. General Discussion

It was found that Inchon Port in the western sea has the lowest water temperature in South Korea and thus was considered as the most suitable area for seawater cold thermal source. Analysing average water temperature in February was found as optimum period for acquiring seawater cold thermal source. This study designed heat exchangers to store seawater cold thermal source, which is acquired for 8 days (February 7th to 13th), during which the seawater temperature of Inchon Port is lowest, in a seasonal thermal storage and analyzed its operational performance. Using prediction equation, this study confirmed that it is possible to design a heat exchanger for acquiring seawater heat source with the analysis data of flow characteristics of heat exchanger. And then CFD simulation was conducted and the results showed that two analyses brought forth almost similar operating characteristics of a heat exchanger, which evidences the designed system in this study is very reliable.

4.1. Seawater Cold Heat Exchanger

(1) Design Dimensions of Heat Exchanger

A total of 16 heat exchangers were installed for this study and each size is about 8.5RT. The maximum heat transfer rate of each heat exchanger was 33.6KW when inflow temperature (T_i) was 25℃, so the total heat transfer rate was 538KW for 16 heat exchangers. The characteristics of heat exchange fluid were analyzed for 3 different velocity of flow: 0.25m/s, 0.5m/s, and 1.0m/s. The results showed that operation at 0.5m/s was the most advantageous, so 0.5m/s was set as standard velocity of flow. The days taken for a heat exchanger to cool the intial temperature (25℃) of a seasonal thermal storage (25℃) down to the target temperature 5℃ is 8 days like the period of annual minimum water temperature.

(2) Discussion over the Performance of Heat Exchanger

The length of the designed heat exchanger is 361.4m. The inflow temperature (T_i) of the heat exchanger was 25℃ and the outlet temperature was 5℃ at the end (361.4m). While the heat exchangers keep operating, the inflow temperature will drop below 25℃. As a result, the heat transfer rate will decrease exponentially as heat exchange proceeds because the gap ($T_i - T_c$) between the inflow temperature (T_i) of the heat exchangers and seawater temperature (T_c) narrows down. This will cause the decrease of heat exchange efficiency of the whole system and rise of power cost due to increasing operating hours. In this case, it is necessary to control velocity of flow, which depends on inflow temperature (T_i) to optimize the performance of the heat exchangers.

(3) Discussion of Material for Heat Exchanger

The heat conductivity of HDPE pipe is 1/800 of metal pipe. Because of this characteristic, the coils of a heat exchanger has to be long more than necessary. Therefore, it is suggested that advanced HDPE pipe with high heat conductivity be developed for acquisition of seawater heat source or ground heat.

Therefore, this study chose a pipe (32mm in thickness) that is mainly used for geothermal heat piping. The pipe has diverse diameters (D16, D20, and D25mm). When a pile with D20mm is chosen, it has 1.6 times greater 'surface ratio for mass flow' than a pipe of D32mm. Therefore, a pipe with smaller diameter is more advantageous in terms of heat transfer rate. Thus, a pipe with smaller diameter will be more considered for choice on field.

4.2. CFD Analysis for Heat Exchanger

In this study, CFD analysis was widely applied to a heat exchanger, from its partial flow characteristics to a whole system. As shown in Figure 7, CFD analysis was conducted on the coil of the heat exchanger increasing from 90˚ to 720˚ by 90˚(2 rotations) to decide the partial length, it showed that as it rotated, the impact of the inlet got weaker, converging to a certain value, so it reached 720˚, it didn't make any change. Therefore, CFD analysis up to 720˚ can be a whole analysis of fluid characteristics.

Table 11 shows the temperature gap (ΔT) between temperature ($T_{m(x)}$) and inlet temperature (T_i) at the point where heat exchanger

coil rotated twice (720° from th coil inlet: converted to length of 13.82m) by inlet temperature. And the estimated values (Table 7) and CFD simulation values (Table 10) were compared. The gap between the estimated values by transfer equation and CFD simulation values was 0.04℃ when T_i=16℃, and 0.005℃ when T_i =7℃. Here, the biggest temperature gap was 0.04℃ when T_i =25℃. Therefore, it is known that two types of values almost match with each other and thus that the estimated values based on the design system are reliable.

Table 11. Temperature Difference Comparison Between Estimate and CFD simulation value

Comparison Temp.(at 720°)	$T_i = 25℃$	$T_i = 16℃$	$T_i = 7℃$
Design Estimate ΔT (℃)	1.93	1.14	0.349
CFD simulation ΔT (℃)	1.89	1.12	0.344
Temp. difference (℃)	0.04	0.02	0.005
Relative error (%)	2.1	1.7	1.4

Therefore, CFD simulation on a heat exchanger was conducted at the point where belongs to the second of total 52.3 heat exchanger coils. And because the result turned out valid, it can be believed generalizable to the whole coils.

5. Conclusion

This study designed a heat exchanger to acquire seawater cold thermal source acquisition. And it theoretically analyzed the fluid characteristics of the heat exchangers to store the acquired cold thermal source in the seasonal thermal storage and ran CFD simulations. The findings are as follows.

(1) Judging from the analysis of seawater temperature in Korea, Incheon area was proved most suitable as seawater cold thermal source. Inchon Port has minimum water temperature for 8 days (February 7th to 13th) a year. The annual average water temperature of 8 days over last 6 years is very low, 1.4℃, so the period was determined to be the fittest for acquiring seawater cold thermal source. The average water temperature of the same period (February 7th to 13th) in 2015 was 2.5℃. The design outer temperature of a heat exchanger was set to 3℃, considering safety factor.

(2) The heat exchanger was designed with HDPE pipe (D32mm), which is durable under seawater and coil-typed (2.2m). it was set for the heat exchanger that when inlet temperature(T_i) is 25℃, outlet temperature(T_e) is 5℃ and seawater temperature (T_c) is 3℃. To optimize the performance of a heat exchanger, velocity of flow was analyzed for 0.25m/s, 0.5m/s, and 1.0m/s, respectively. Operation at 0.5m/s was turned out the most appropriate. With the temperature conditions above considered, it was estimated that the optimum length of pipe per heat exchanger is 361.4m at operating velocity (of flow) of 0.5m/s, using the equations.

(3) Under the same condition as (2), heat transfer rate $(\dot{q})$ was calculated to be 33.6kW by heat transfer formulas. As the heat exchangers operate, the temperature of the thermal storage tank lowers. When inflow temperature (T_i) drops below 25℃, heat transfer rate decreases exponential. And when inflow temperature T_i=7℃, heat transfer rate sharply drops to 3.3kW. In this case, it is necessary to consider controlling velocity of flow to improve the performance of heat exchange.

(4) This study confirmed that it is possible to design a heat exchanger for the purpose of acquiring seawater heat source by using prediction equation. In this study, the length of the pipe was longer than it should be because synthetic resin whose thermal conductivity is far lower than metal was used as a material for the pipe. Therefore, it is deemed necessary to develop HDPE pipe with high heat conductivity for the purpose of acquiring seawater heat source or using geothermal heat.

(5) Using 3D CFD, this study ran simulation on the temperatures of the heat exchangers. Instead of calculating the entire coils, this study defined 720° (2 rotations) as scope of analysis. It was known possible to exclude the impact of inlet area and estimate the performance of the entire heat exchangers. In addition, it was confirmed that the designed temperature gaps are almost the same as actually measured gaps.

This study demonstrated that it is reliable to use prediction equations in designing a heat exchanger, and the findings of this study can be used as the basic data for design of a heat exchanger for seawater heat source.

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