HEAT TRANSFER ON GROOVED HIGH DENSITY POLY ETHYLENE TUBE FOR SURFACE WATER SOURCE HEAT PUMP

by

Gun Joo JUNG, Geun Sug OH* , Chul Woo CHUNG, and Han Ji KIM

Department of Architectural Engineering, College of Engineering,
Pukyong National University, Nam-Gu, Busan, Korea

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High density polyethylene (HDPE) tube has been successfully utilized in surface water source heat pump (SWSHP) system as a surface water heat exchanger. Since the heat transfer coefficient (U value) of the HDPE tube directly affects performance and energy efficiency of SWSHP, this research aims to increase U value of HDPE tube by grooving external surface of conventional 32A HDPE tube to reduce cross-sectional volume. The final shape of grooved HDPE tube is similar to that of fin. In order to verify the performance of grooved HDPE tube, the U values of grooved and smooth tube were compared experimentally. According to the results, U value of grooved tube showed approximately 21.5% increase with natural convection and 23.5% with forced convection system than U values obtained from smooth tube. The reason for such increase in U value was found to be the reduction in cross-sectional volume of the HDPE tube.

Key words: *surface water source heat pump, surface water heat exchanger, heat transfer coefficient (U value), high density polyethylene tube*

Introduction

Geothermal heat pump (GHP) and surface water source heat pump (SWSHP) provide higher coefficient of performance (COP) than air source heat pump (ASHP) system. Both GHP and SWSHP are known to be energy efficient, and more frequently used as the source of heating and cooling system. Therefore, many researchers have focused on the performance enhancement of GHP and SWSHP systems [1-6].

In surface water heat exchanger (SWHE) system, the use of metallic tube with higher heat conductivity reduces the length of cooling tube and takes advantage in pumping head. However, metallic tube that is immersed in water can potentially causes corrosion and brings durability problem. For this reason, non-corrosive high density polyethylene (HDPE) tube has been utilized in SWHE system. However, HDPE, has low thermal conductivity that is 0.4 W/mK, so thermal efficiency is much less than metallic tube. Therefore, many efforts have been made to increase the heat transfer coefficient (U value) of HDPE tube in SWHE system. There are several ways to increase the U value of HDPE tube. First is to reduce the thickness of the tube which brings the reduction in thermal resistance. Second is so-called fin effect by the increasing of the tube surface

* Corresponding author; e-mail: envioh@pknu.ac.kr

area. It was shown that HDPE tube with low thermal conductivity and high convection heat transfer coefficient (h_o) showed that the fin effect may not occur when the HDPE tube is immersed in water [7].

This research suggests grooved HDPE tube to enhance U value of SWHE system. Conventional 32A HDPE smooth tube was grooved and used for the experiments. Grooved tube has less thickness and higher surface area for better convection. In this research, experimental works were performed and calculations utilizing heat transfer equations were utilized to compare the performance of smooth tube and grooved tube. When better performance is observed with grooved HDPE tube, the reason will be investigated whether the increase in U value is associated with thickness change (reduction in cross-sectional volume) or with the fin effect (increase in surface area).

The purposes of this research are (1) to provide U value of smooth tube (32A HDPE tube) for the design of SWHE system, (2) to investigate the effect of grooved tube on increase in U value, (3) to investigate whether the use of grooved HDPE tube (with fin shape) in SWHE system brings fin effect or not, and (4) to investigate the economic benefits with grooved HDPE tube.

Theories of heat transfer and friction loss in tube

Heat transfer rate of fluid and other typical variables in heat exchanger system can be obtained according to eqs. (1) through (4). In this research, experimentally measured values were used to calculate heat transfer rate [8]. According to the equations, the energy required for cooling when passing through heat exchanger system is in linear relationship with the mass of fluid, specific heat of water, and temperature difference. Also, U value of tube, surface area, and logarithmic mean temperature difference (LMTD) are variables that affect heat transfer rate. When cooling energy is obtained from experimental results, LMTD and total surface area of tube can be used in eq. (3) to obtain U value of the tube:

$$q = mC_p(T_{\text{in}} - T_{\text{out}}) \tag{1}$$

$$q = UA_o\Delta T_m \tag{2}$$

$$U = \frac{q}{A_o\Delta T_m} \tag{3}$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln\dfrac{\Delta T_1}{\Delta T_2}} \tag{4}$$

When the temperature difference of water is not significant, heat transfer through radiation can be ignored. Therefore, when the heat exchange occurs associated with the temperature differences in and outside of tube, the variables that affect heat transfer become conductivity, internal convection, and external convection. In steady-state condition, the heat transfer rate and thermal resistance from inner to outer surface in conductive tube can be obtained using eqs. (5) and (6):

$$q_{\text{r}} = \frac{2\pi kL(t_{\text{i}} - t_{\text{o}})}{\ln\dfrac{r_{\text{o}}}{r_{\text{i}}}} \tag{5}$$

$$R_c = \ln \frac{\frac{r_o}{r_i}}{2\pi k L} \tag{6}$$

In eqs. (5) and (6), as the ratio between r_o and r_i are smaller, in other words, as the thickness of tube becomes smaller (thermal conductivity becomes higher; lower thermal resistance), higher amount of heat passes through.

The inner surface, outer surface, and heat transfer can be classified as components of convection. Surface heat transfer can occur more in turbulent flow than in laminar flow. Surface heat transfer increases as the velocity of liquid increases. When the temperature difference decreases and the diameter of the tube increases, surface heat transfer decreases. Surface heat transfer also increases as the specific heat and the viscosity of water increases. The relationship between such variables and convection heat flow can be explained the Reynolds, Prandtl, Nusselt, and Rayleigh numbers. Equations (7) through (12) allow us to calculate convection heat transfer from inner and outer surface of tube (in SWHE system) in steady-state (natural convection) and forced convection systems. Nusselt, Reynolds and Prandtl numbers can be calculated using eqs. (7) through (9), and in case of laminar flow (Re < 2300) and turbulent flow (Re ≥ 10000), the Nusselt number can be calculated using eq. (10) provided by Sieder and Tate [8], and eq. (11) by Dittus and Boelter [9]:

$$\mathrm{Nu} = \frac{hD_h}{k} = f(\mathrm{Re}, \mathrm{Pr}) \tag{7}$$

$$\mathrm{Re} = \frac{\rho V_{avg} D_h}{\mu} \tag{8}$$

$$\mathrm{Pr} = \frac{C_p \mu}{k} \tag{9}$$

$$\mathrm{Nu} = 1.86[\mathrm{Re\,Pr}/\,L/D]^{1/3}(\mu/\mu_S)^{0.14} \tag{10}$$

$$\mathrm{Nu} = 0.023\mathrm{Re}^{4/5}\mathrm{Pr}^{0.4} \tag{11}$$

If external liquid can be cross flow with tube, the external Nusselt number can be calculated using eq. (12) given by Churchill and Bernstein [10]. Natural convection outside the tube, steady-state Nusselt, Rayleigh, and Grashof numbers can be calculated using eqs. (13) [11], (14), and (15) [7] from Churchill and Chu [11]:

$$\mathrm{Nu} = 0.3 + (0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3})/[1 + (0.4/\mathrm{Pr})^{2/3}]^{1/4}[1 + (\mathrm{Re}/282000)^{5/8}]^{4/5} \tag{12}$$

$$\mathrm{Nu} = \{0.6 + 0.38\mathrm{Ra}^{1/6}/[1 + 0.559/\mathrm{Pr})^{9/16}]^{8/27}\}^2 \tag{13}$$

$$\mathrm{Ra} = \mathrm{GrPr} \tag{14}$$

$$\mathrm{Gr} = g\beta\rho^2\Delta TL^3/\mu^2 \tag{15}$$

To calculate the thermal resistance associated with inner and outer surface convection, eq. (16) can be used. The U value within the tube can be calculated using eqs. (17), (18), and (19) [12].

$$R_h = \frac{1}{hA_t} \tag{16}$$

$$R = R_{hi} + R_c + R_{ho} \tag{17}$$

$$K = \frac{1}{R} \tag{18}$$

$$U = \frac{k}{A} \tag{19}$$

Friction factor can be calculated using eq. (20). Correlation between Reynolds number and the friction factor can be calculated using eqs. (21) and (22) provided by Moody [13]:

$$f = \frac{\Delta P}{\left(\frac{L}{D_\mathrm{i}}\right)\left(\frac{\rho V^2}{2}\right)} \tag{20}$$

$$f = 0.316\mathrm{Re}^{-1/4} \quad \text{for Re} < 20{,}000 \tag{21}$$

$$f = 0.184\mathrm{Re}^{-1/5} \quad \text{for Re} > 20{,}000 \tag{22}$$

Experimental procedure

Preparation of sample

The tube line was constructed using smooth tube HDPE 32A, thickness of 5 mm, length of 24.54 m (submersible). The thermal conductivity of the HDPE 32A was 0.4 W/mK [14]. Similar set-up was also constructed using grooved HDPE 32A tube. Two sets were used for comparison of experimental data. Photographic images of the experimental set-up are shown in fig. 1(a).

The grooving in the HDPE 32A tube was done for 2 mm depth and 3.5 mm width with pitch distance of 5 mm. The width of extruded portion in grooved HDPE tube (fin shape) was 1.5 mm. As the tube was grooved, the reduction in volume of material was 30.3% calculated by eq. (23), but the increase in the surface area of the tube was 69.5% by eq. (24).

$$\text{Volume reduction} = [(\pi D_o^2 / 4)0.0035 - (\pi D_g^2 / 4)0.0035] / \\ / [(\pi D_o^2 / 4)0.005 - (\pi D_i^2 / 4)0.005] \tag{23}$$

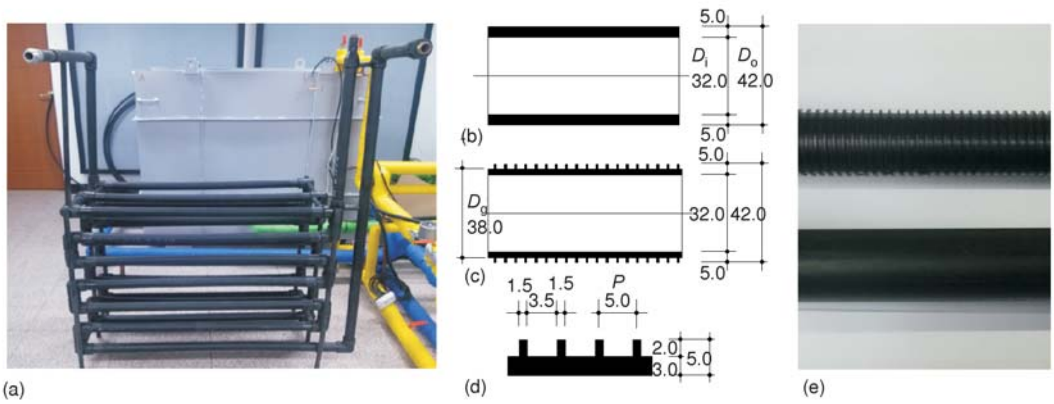

Figure 1. (a) Test tube assembly; (b) Sectional dimension of smooth tube; (c) Sectional dimension of grooved tube; (d) Detail of grooving; (e) Picture of grooved (above) and smooth tube (bottom)

$$\text{Surface increasing} = \{(\pi D_\text{o})0.0015\cdot 200 + (\pi D_g)0.0035\cdot 200 + \\ + [\pi D_o^2/4) - (\pi D_g^2/4)]400\}/(\pi D_o^2) \quad (24)$$

Preparation of testing

The experimental setup consists of two internal cooling water baths (0.8 × 1.2 × 1.2 m), two external circulating water baths, cooling system 2RT, heating system 9 kW, circulating pump with pumping head 10 mAq, maximum flow of 60 lpm, and 25A tube line system. The schematic diagram of experimental set-up is shown in fig. 2. The control for water temperature was done using silicon controlled rectifier that operates with ±1 °C precision. The temperature control in cooling circulating water baths were done using thermo-state. The precision of thermo-state is about ±2 °C. For forced convection within water bath, water pump with 2 m pumping head and 30 lpm flow rate was placed on the bottom of each water chamber.

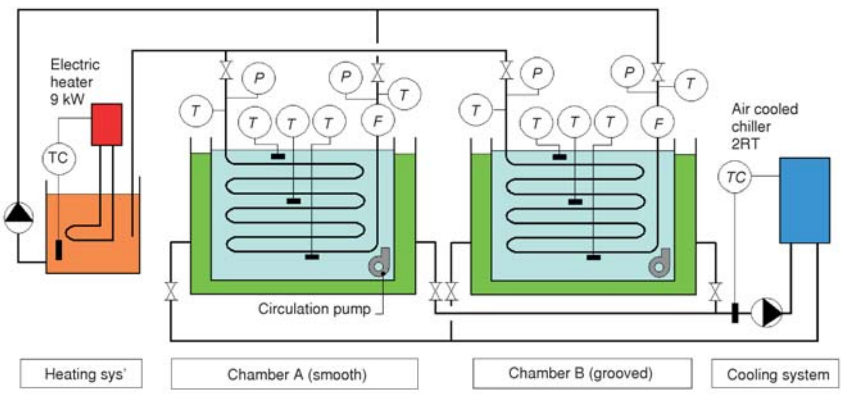

Figure 2. Schematic of experimental system to obtain *U* values and pressure drops of smooth tube and grooved tube

Specification of testing equipment

Electronic flow meter was used. The 28 channels data logger was used to monitor temperature. The speed of external flow of water of tube was measured using propeller type flow meter. Table 1 shows the specification of testing equipment for this experiment.

Table 1. Specification of data loggers and measuring equipment

Measuring object	Type	Points	Range	Sensitivity	Maker
Flow rate of hot water	Electromagnetic flow meter (25A)	2	3 60 lpm	±2%	AICHI TOKEI DENKI Co., Ltd
Temperature	*k*-type MV1204 28chs	28	–220-1370 °C	±0.1 °C	YOKOGAWA
Water velocity of outer tube	Flow watch	2	0.08-42 m/s	±2%	JDC ELECTRONIC
Pressure of inner tube	C-type bourdon tube	4	0-0.01 MPa	±1.5%	MYUNG SUNG INSTRUMENT

Testing

In this study, U values from conventional (smooth tube) and suggested (grooved) HDPE tubes were compared when water was flowing or not. The physical properties of water are presented in tab. 2. The water chamber was in natural convection (when external water flow was terminated) and forced convention mode (external water velocity of 0.1 m/s). U values of smooth tube at natural and forced convection were measured. U values of grooved tube at natural and forced convection were also measured. The experimental variations were set at 4 different conditions (grooved and smooth tube and natural and forced convection). The resulting U values from each condition were compared.

The experiments were conducted at January 2nd 2013 in Architectural Environmental Laboratory located in Pukyong National University, Busan, Republic of Korea. The experiments were continued for 3 weeks. Temperature and relative humidity of the laboratory were set at 22 ± 2 °C and 40 ± 10%, respectively. The temperature of warm water in HDPE tube was set at 41.0 ± 1.0 °C, and the temperature of cooling water chamber was set at 16.0 ± 2.0 °C. LMTD was maintained at approximately 22 °C. The experimental system was operated for 3 hours at a time. Since the actual design of SWSHP uses the flowing rate of cooling water at 0.5-0.7 m/s, the flowing rate of warm water was set at 0.1-0.8 m/s, and varied using step increase of 0.1 m/s. The temperature of warm water at inlet and outlet of the tube was monitored at every 1 second. The temperature of cooling water chamber at lower, middle, and higher portion were also measured at every 1 second. At each flowing rate, the experiments were done for 30 minutes. Before obtaining the meaningful datasets, 10 minutes of thermal equilibrium at testing temperature was provided. The effective flow rate was calculated using the inlet and outlet areas of the tube, doubling these values, and converted them into flow rate per hour.

Table 2. Physical properties of water

Description	Value	Unit	Description	Value	Unit
Conductivity of water (16 °C)	0.596	W/mK	Viscosity of water (16 °C)	$11.258 \cdot 10^{-4}$	N/m^2
Conductivity of water (40 °C)	0.631	W/mK	Viscosity of water (40 °C)	$6.632 \cdot 10^{-4}$	N/m^2
Conductivity of HDPE tube	0.400	W/mK	Kinematic coefficient of viscosity of water (16 °C)	$1.127 \cdot 10^{-6}$	m^2/s
Specific gravity (16 °C)	998.8	kg/m^3	Kinematic coefficient of viscosity of water (40 °C)	$0.668 \cdot 10^{-6}$	m^2/s
Specific gravity (40 °C)	992.3	kg/m^3	Pr number (16 °C)	7.998	–
Coefficient of cubical expansion (16 °C)	0.00015	1/K	Pr number (40 °C)	4.390	–

Results and discussion

Verification of experimental results using the smooth tube

Convective heat transfers of inner and outer surface of smooth tube were verified using Nusselt number with eqs. (19), (18), (17), (16), and (7). Experimental errors of resistances of convections and conduction in tubes were calculated using the difference between calculated U value and experimental U value. Figure 3(a, b) show the Nusselt numbers between calculated and experimental data. The standard deviations (SD) between calculated and experimental Nu

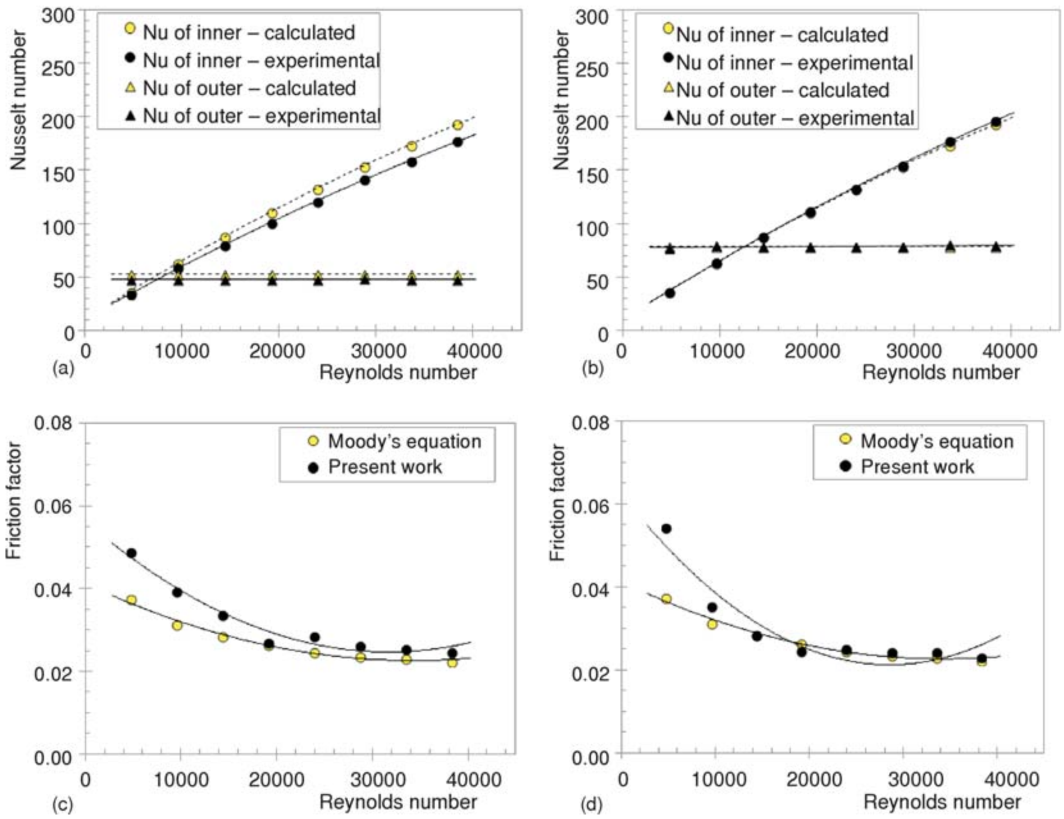

Figure 3. Verification of Nusselt numbers, friction factors of smooth tube; (a) Nusselt numbers at natural convection; (b) Nusselt numbers at forced convection; (c) friction factors at natural convection; (d) friction factors at forced convection

numbers from inner surface of smooth tube were 4.73 at the natural convection in test chamber, and 1.47 at the forced convection, respectively.

In the calculation of experimental friction factors of smooth tube, the equivalent length of fittings was applied as follows: elbow was 2.0 m/each and branch flow tee was 2.7 m/each [15]. In test tube system were 45 elbows and one branch tee, length of connection tube was 1.95 m, and the length of linear test tube was 24.54 m. Therefore, the applied total tube length was calculated as 119.19 m. Experiment results on the friction factors of smooth tube were calculated by eq. (20). The results were verified using eqs. (21) and (22). Figure 3 (c, d) present that the friction factors between calculated and experimental data in smooth tube. SD between calculated and experimental friction factors of inner surface of tube were 0.0004 at the natural convection in test chamber, 0.0006 at the forced convection, respectively. Especially, in the lower Reynolds number range such as 5,000-14,000, friction factor of present work is approximately higher by 0.01 than the friction factor driven by Moody's eqs. (21) and (22).

U value of smooth tube

The *U* value of smooth tube (with outer diameter of 42 mm and tube thickness of 5 mm) was calculated using eq. (5) through (13). According to eq. (5) through eq. (19), at the internal flow velocity of 0.5 m/s, internal thermal resistance R_{hi} of smooth tube at natural convection was 0.00379 K/Wm. External thermal resistance R_{ho} of smooth tube was 0.01311 K/Wm. Thermal resistance due to thickness of the tube R_c was 0.10825 K/Wm. Total thermal resistance R (sum of internal thermal resistance, external thermal resistance, and thermal resistance associated with thickness of the tube) was 0.12515 K/Wm. At the internal flow velocity of 0.5 m/s, the internal thermal resistance R_{hi} of smooth tube at forced convection was 0.00379 K/Wm, external thermal resistance R_{ho} was 0.00696 K/Wm, thermal resistance due to thickness of the tube R_c was 0.10825 K/Wm, and total thermal resistance R was 0.11900 K/Wm. At the internal flow velocity of 0.5 m/s, total thermal resistance in natural convection (0.12515) was 5.2% higher than that in forced convection (0.11900). This procedure was repeated at each velocity level from 0.1 to 0.8 m/s. Table 3 shows the calculated U value of HDPE tube with external diameter of 42 mm and 38 mm in case of natural and forced convection. The calculated U value from smooth tube varied from 56.12 to 61.19 W/m^{2}K with natural convection and with forced convection U value varied from 58.78 to 64.36 W/m^{2}K. Therefore, forced convection showed 5.1% higher U value than natural convection. In general, the calculated U values were very similar to those from experiments. Also, the thermal resistance associated with velocity of water was very low, whereas that associated with thickness of the tube was relatively higher. Table 3 shows the U values of smooth tube and grooved tube.

U value of grooved tube

According to eq. (1) through (4), U value of grooved tube was calculated using LMTD, heat transfer rate of fluid, and the surface area of the tube. Table 3 shows experimental U value of grooved tube at both natural and forced convection.

With varying the water velocity from 0.1 to 0.8 m/s, U value of grooved tube ranged from 62.30 to 70.45 W/m^{2}K in natural convection and from 72.60 to 80.45 W/m^{2}K in forced convection, respectively. Forced convection showed 14.9% higher U value than natural convection. As noted from the case of smooth tube, forced convection always showed higher U value regardless of grooving HDPE tube. U value of grooved tube was 21.5% higher in natural convection, and 23.5% higher in forced convection. The correlation between grooved and smooth tube (associated with the changes in the velocity of the fluid) was calculated. The correlation was 0.9696 in natural convection and 0.9556 in forced convection. Strong correlation was observed between U value and the velocity of water. Table 3 and fig. 3 show that U values of smooth tube and grooved tube in the present work.

Causes of increase in U value of grooved tube

In order to understand the cause of increase in U value, eqs. (1) through (19) were used to obtain U value of smooth tube with outer diameter of 38 mm. For this calculation, the presence of the fin (height 2 mm) was neglected from the grooved tube (outer diameter 42 mm), assuming that it is the smooth tube with the outer diameter of 38 mm and thickness of 3 mm. According to the calculation, the thermal resistance R_c was 0.06838 K/Wm, 36.8% lower than thermal resistance from the case with smooth tube that has the outer diameter of 42 mm and thickness of 5 mm. Thermal resistance of 38 mm smooth tube was decreased by reducing outer diameter by 4 mm and thickness by 2 mm.

Table 3 shows the comparison of U values of 42 mm smooth tube, 38 mm smooth tubes, and 42 mm grooved tube. When the water velocity varied from 0.1 to 0.8 m/s, U value of smooth tube with outer diameter of 38 mm ranged from 88.36 to 100.16 W/m^{2}K at natural convection. With forced convection, U value of smooth tube ranged from 93.87 to 107.29 W/m^{2}K.

Table 3. Experimental, calculated *U* value of 32A HDPE tube

Classification of experiment	Velocity of hot fluid	Re number of inner tube	Rate of flow of hot fluid	Mean temp. difference of inlet and outlet of hot fluid	LMTD	Rate of heat transfer	Experimental *U* value of outer diameter 42 mm	Calculated *U* value of outer diameter 42 mm, thk 5 mm	Calculated *U* value of outer diameter 38 mm, thk 3 mm
	[ms^{-1}]	[–]	[lithr^{-1}]	[°C]	[°C]	[kJhr^{-1}]	[Wm^{-2}K^{-1}]	[Wm^{-2}K^{-1}]	[Wm^{-2}K^{-1}]
Smooth tube, natural convection (Ra number is 38095212)	0.1	4786	286	9.43	17.96	11,178	53.42	56.12	88.36
	0.2	9573	580	5.69	21.23	13,665	55.27	58.68	94.18
	0.3	14359	862	4.01	22.21	14,329	55.40	59.69	96.56
	0.4	19146	1156	3.14	23.11	15,061	55.96	60.25	97.89
	0.5	23932	1446	2.57	23.34	15,410	56.70	60.61	98.75
	0.6	28719	1736	2.20	23.60	15,807	57.52	60.86	99.35
	0.7	33505	2022	1.88	23.68	15,733	57.05	61.05	99.80
	0.8	38292	2314	1.65	23.63	15,805	57.42	61.19	100.16
Smooth tube, forced convection (Re number is 4189)	0.1	4786	292	10.18	18.27	12,317	57.89	58.78	93.87
	0.2	9573	584	6.45	21.48	15,618	62.43	61.58	100.47
	0.3	14359	866	4.61	22.49	16,543	63.15	62.70	103.18
	0.4	19146	1160	3.55	22.92	17,050	63.87	63.32	104.70
	0.5	23932	1446	2.80	22.50	16,781	64.03	63.71	105.68
	0.6	28719	1744	2.45	23.48	17,677	64.63	63.99	106.37
	0.7	33505	2022	2.13	23.28	17,827	65.75	64.20	106.89
	0.8	38292	2322	1.83	23.12	17,633	65.49	64.36	107.29
Grooved tube, natural convection (Ra number is 38095212)	0.1	4786	290	11.99	19.86	14,414	62.30	–	–
	0.2	9573	584	7.02	22.09	16,987	66.03	–	–
	0.3	14359	872	4.91	22.62	17,727	67.28	–	–
	0.4	19146	1162	3.86	23.10	18,568	69.02	–	–
	0.5	23932	1448	3.04	22.60	18,242	69.31	–	–
	0.6	28719	1744	2.60	23.05	18,765	69.89	–	–
	0.7	33505	2030	2.20	22.59	18,536	70.45	–	–
	0.8	38292	2310	1.93	22.37	18,508	71.05	–	–
Grooved tube, forced convection (Re number is 4189)	0.1	4786	288	13.05	18.41	15,569	72.60		
	0.2	9573	572	8.04	21.05	19,049	77.72		
	0.3	14359	872	5.41	21.63	19,555	77.62		
	0.4	19146	1162	4.13	21.84	19,867	78.09		
	0.5	23932	1442	3.41	21.74	20,375	80.48		
	0.6	28719	1734	2.82	21.75	20,298	80.14		
	0.7	33505	2028	2.40	21.63	20,173	80.06		
	0.8	38292	2314	2.10	21.59	20,114	79.99		

With the same range of water velocity, *U* value of smooth tube (outer diameter 42 mm, 5 mm) ranged from 56.12 to 61.19 W/m^{2}K at natural convection and from 58.78 to 64.36 W/m^{2}K at forced convection. When the thickness of smooth tube decreased from 5 mm to 3 mm (30.3% volume reduction of material), U value increased for 63% with natural convection and 65% with forced convection.

According to these calculated results, the reduction in thickness is highly effective to increase U value of HDPE tube.

The experimental U value of grooved tube with outer diameter of 42 mm was 70% and 75% of calculated U value of smooth tube with outer diameter of 38 mm, in natural and forced convection, respectively. If the grooved tube showed fin effect, the experimental U value of 42 mm grooved tube must be higher than calculated U value of 38 mm smooth tube. However, our results showed that U value of 42 mm grooved tube was clearly lower than 38 mm smooth tube. Therefore, the result suggests that the cause of increase in U value with the grooved tube was associated with the reduction in cross-sectional volume of the tube, not associated with the fin effect (fig. 3a and b). Our finding is that HDPE with lower thermal conductivity of material cannot obtain fin effect in SWHE system.

Friction factor and fouling factor of grooved tube

The average friction factors of grooved tube were 22% higher than smooth tube at natural convection, and 35% higher than at forced convection. The changes in friction factors can be assumed to be caused by the fact that the inner surface of grooved tube was very slightly corrugated because it is a soft material and can be deformed during grooving process. Figure 4(a, b) show the correlation between friction factors and Reynolds number from two tubes, at natural and forced convection in the test chamber.

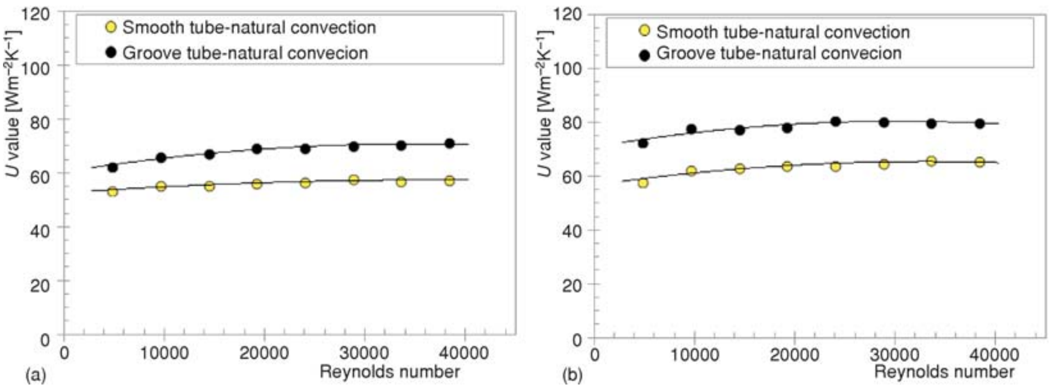

Figure 4. Comparison of U value of smooth tube and grooved tube; (a) at natural convection in the test chamber, (b) at forced convection in the test chamber

Fouling factor can influence on U value of real state SWHE in long term operation. Therefore, calculation of R_{real} [$KW^{-1}m^{-1}$] value at real state tube should consider the fouling factors ($R_{fi} + R_{fo}$). Equation (25) represent the real state resistance of tube can be driven from eq. (17). Fouling factors in the tube and the river were each 0.0001 m^2K/W, respectively [16].

$$R_{real} = R + R_f = R_{hi} + R_c + R_{ho} + \frac{R_{fr}}{A_i} + \frac{R_{fo}}{A_o} \tag{25}$$

In the present work, fouling resistance such as $R_f = (R_{fi}/A_i) + (R_{fo}/A_o)$ is 0.00175 Km/W, it is 1.4% of total resistance $R = 0.12215$ Km/W from clean tube.

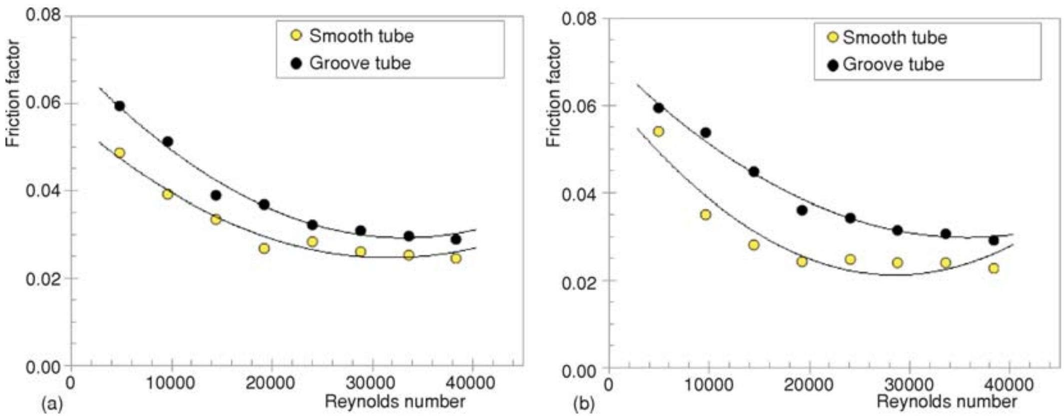

Figure 5. Correlation of friction factors between smooth tube and groove tube; (a) at natural convection in the test chamber, (b) at forced convection in the test chamber

COP comparison between smooth and grooved tube

For the estimation of power consumption with smooth and grooved tube, in this research, our research team established test bed chilling system using 1.0 RT refrigerator (made by LG Co., Ltd.) and the existing tubes. The smooth and grooved tube with 32A HDPE and length of 24.54 m was submerged in water. Calculation of *COP* was done following eq. (26). In calculation of *COP*, the electricity power of cooling water circulation pump was not applied in order to evaluate the thermal performances of test cooling tubes. *COP* experiment was performed only in summer (chilling) condition because the existing experimental system was not designed to support the winter (heating) condition.

$$COP = \text{output Watt with evaporator/input Watt with motor of compressor} \tag{26}$$

In the experiment, the velocity of fluid in the 32A tubes was 0.7 m/s, and temperature of the chamber was maintained approximately at 20 °C. At the condition of natural convection in the chamber, the temperatures of supply cooling water were 39.6 °C in the smooth tube and 36.2 °C in the grooved tube. *COP* of the smooth tube and grooved tube was 1.891 and 2.267, respectively. *COP* of grooved tube was 19.9% higher than that of the smooth tube in the natural convection.

At the condition of forced convection in the chamber, the temperatures of supply cooling water were 37.3 °C in the smooth tube, 35.0 °C in the grooved tube. *COP* of the smooth and grooved tube was 2.129 and 2.491, respectively. *COP* of grooved tube was 17.0% higher than that of the smooth tube in the forced convection.

This increase in *COP* in the grooved tube was because the U value of grooved tube was 21.5-23.5% higher than that of smooth tube, and as a result, the circulation cooling temperature at grooved tube reduced about 2.4-3.1 °C.

Economic evaluation of grooved tube

Cost saving of material with grooved tube in SWHE system was calculated for economic evaluation. Table 4 shows the cost saving with grooved tube in the SWHE system. In determination of length with 32A HDPE tube for 1RT refrigerator system, the lengths of smooth and grooved tube were each found to be 24.32 meters per USRT (assumed same length of both tube systems, see equation in tab. 4). The unit price of the smooth tube was 1.47 US dollars per meter [17], and the unit price of the grooved tube was 1.18 US dollars per meter in Korean market. The manufacturing cost of grooved tube was 10% higher than that of smooth tube. However, the unit volume of grooved tube was 30% lower than that of smooth tube, thus making 20% reduction in total unit price. The material costs of smooth and grooved tube were calculated to be 35.75 and 28.70 US dollars per RT, respectively. Therefore, the material cost of grooved tube was 20% lower than that of smooth tube in SWHE system.

Table 4. Cost saving with grooved tube in test bed SWHE system

Description	Cooling capacity	U value (fluid v. 0.7 m/s, outer of tube was forced convection)	LMTD	Demand of tube length, 32A HDPE	Unit price of 32A HDPE tube, 5thk, in S. Korea	Material cost of cooling tube set	Remark
Calculation eq. and unit	W = 1.2 USRT [WRT^{-1}]	U [Wm^{-2}K^{-1}]	ΔTm [°C]	$L = W/(U\Delta T\text{m}A_o)$ [mRT^{-1}]	Tube unit price [US\$m^{-1}]	Cost = L·Tube unit price [US\$RT^{-1}]	
Smooth	4220	65.75	20.00	24.32	1.47	35.75	(assumed same length)
Grooved	–	–	–	24.32	1.18	28.70	
Saving	–	–	–	–	0.29	7.05	
Ratio	–	–	–	–	20%	20%	

In the calculation of annual power cost savings of circulation pump and compressor with grooved tube system, in this research, our research team used the method of simple estimation, but we did not used the computational simulation. This is the limitation of this research. The assumed factors were applied on running power consumption: *COP* of smooth tube – 2.129, and grooved tube – 2.491 (in forced convection experiment, see previous subsection), LMTD – 20 °C (summer condition), the fluid velocity – 0.7 m/s, the unit pressure drop of smooth tube – 0.018 mAq, the unit pressure drop of grooved tube – 0.023 mAq (from experiment), the annual running time – 1,460 h (6 month per year, 8 h per day operation in the residential), and the average load factor – 0.65 (as known load factor range is 0.3-1.0) [18]. The unit price of electricity power in Korean market was 0.72 US$ per kWh in the residential [19].

As result of calculation, the power of circulation pump with grooved tube was 5% higher than that smooth tube. However, power of compressor with grooved tube was 15% lower than that smooth tube. The annual power consumption with smooth tubes was 1648.12 kWh per RT (running time), and for grooved tubes it was 1424.59 kWh per RT. Annual running power cost with smooth tubes was 1186.65 US$ per RT, and for grooved tubes it was 1025.70 US$ per RT. Therefore, the annual electricity power consumption with grooved tubes is 14% lower than with smooth tubes in test bed system. Detailed information is shown in tab. 5.

Table 5. Power saving of circulation pump and compressor with grooved tube in test bed system

Descrip-tion	Cooling water rate in real state	Pump head in system	Power of circulation pump	Power of compressor (forced convection)	Annual running time	Load factor	Annual electricity power consump-tion	Unit price of electricity in S. Korea	Annual running power cost
Calculation equation and units	$Q = V(\pi D_i^2/4)\,60\cdot 1000\ (V = 0.7\ \mathrm{m/s})$ $[\mathrm{lpmRT^{-1}}]$	H = Condenser loss + tube loss [mAq]	$P_p = (\rho QH/6120\,\eta)$coupling loss $[\mathrm{kWRT^{-1}}]$	P_c = Chilling demand / COP $[\mathrm{kWRT^{-1}}]$	Art = 365(6/12) 8 h $[\mathrm{hyr^{-1}}]$	Lf [–]	Pann = $(P_p + P_c)Lf Art$ $[\mathrm{kWhyr^{-1}RT^{-1}}]$	UPE $[\mathrm{US\$kWh^{-1}}]$	Pann·UPE (US$/yrRT) $[\mathrm{US\$yr^{-1}RT^{-1}}]$
Smooth	33.78	7.03	0.085	1.652	1460	0.65	1648.12	0.72	1186.65
Grooved	33.78	7.40	0.089	1.412	1460	0.65	1424.59	0.72	1025.70
Saving	–	–0.37	–0.004	0.240	–	–	223.53	–	160.95
Ratio	–	–5%	–5%	15%	–	–	14%	–	14%

Calculation process of pumping head and compressor's power
(1) Pumping head of smooth tube; condenser = 5 mAq, tube loss = 0.018 mAq/m (0.7 m/s)·24.32 (m-submersible)·3 (fittings factor) + 0.018 mAq 40.0 (m-connection) = 2.03 mAq
(2) Pumping head of grooved tube; condenser = 5 mAq, tube loss = 0.023 mAq/m (0.7 m/s)·24.32 (m-submersible)·3 (fittings factor) + 0.018 mAq 40.0 (m-connection) = 2.40 mAq
(3) Power of compressor motor with smooth tube; P_c = (3517/2.129) / 1000 = 1.652 kW
(4) Power of compressor motor with grooved tube; P_c = (3517/2.491) / 1000 = 1.412 kW

Conclusions

By varying fluid water velocity in 32A HDPE tube from 0.1 to 0.8 m/s (Reynolds number – approximately 4,000-40,000), at natural convection in test chamber, *U* value of smooth tube ranged from 53.42 to 57.42 W/m^{2}K. At forced convection in test chamber, the outer water velocity of 0.1 m/s (Reynolds number – approximately 4,000), *U* value of smooth tube ranged from 57.89 to 65.49 W/m^{2}K.

U value of grooved tube showed approximately 21.5% increase at natural convection in test chamber and 23.5% increase at forced convection than *U* values from smooth tube.

The result indicates that the increase in *U* value was associated with the reduction in cross-section volume of HDPE tube wall material rather than the contribution by fin effect. In other words, in case of HDPE tube with higher external convective heat transfer rate, with lower thermal conductivity, extruded fin was behaving as a thermal resistor rather than a thermal bridge.

The material cost of grooved tube was 20% lower than smooth tube, the annual electricity power consumption with grooved tube was 14% lower than smooth tube in test bed SWSHP system.

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Nomenclature

A_i – surface area of inner tube, [m^2]
A_o – surface area of outer tube, [m^2]
A_t – surface area of tube, [m^2]
C_p – specific heat, [Jkg^{-1}K^{-1}]
D_i – inside diameter of tube, [m]
D_g – outer diameter of grooved tube, 38 mm [m]
D_h – hraulic diameter, [m]
D_o – outer diameter of smooth tube, 42 mm [m]
f – friction factor, [–]
g – gravitational acceleration, [ms^{-2}]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
K – overall heat transfer coefficient with tube length, [WK^{-1}m^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length, [m]
m – mass rate of flow, [kgs^{-1}]
Nu – Nusselt number, [–]
ΔP – pressure drop across test tube, [Pa]
Pr – Prandtl number, [–]
q – heat transfer rate, [W]
q_r – heat transfer rate with conduction, [W]
R – total thermal resistance, [Wm^{-1}K^{-1}]
Ra – Rayleigh number, [–]
Re – Reynolds number, [–]
R_{real} – thermal resistance with fouling factor, [Wm^{-1}K^{-1}]
R_c – thermal resistance with conduction, [KW^{-1}]
R_{fi} – fouling factor of inner surface, [m^{2}KW^{-1}]
R_{fo} – fouling factor of inner surface, [m^{2}KW^{-1}]
R_h – thermal resistance with convection, [Wm^{-1}K^{-1}]
R_{hi} – thermal resistance with convection at inner tube, [KW^{-1}m^{-1}]
R_{ho} – thermal resistance with convection at outer tube, [KW^{-1}m^{-1}]
r_o, r_i – radius of in and out, [m]
ΔT_m – LMTD, [°C]
T_{in}, T_{out} – temperature of inlet and outlet in tube, [°C]
ΔT_1, ΔT_2 – temperature differences between the fluids at each end of the heat exchanger, [°C]
t – tickness of material, [mm]
t_i, t_o – temperature of in and out, [°C]
U – overall heat transfer coefficient with tube – surface area, [W·m^{-2}K^{-1}]
V_{avg} – avetrage velocity, [ms^{-1}]

Greek symbols

β – coefficient of cubical expansion, [–]
μ – viscosity, [Nsm^{-2}]
μ_S – viscosity with surface temperature, [Nsm^{-2}]
ρ – density, [kgm^{-3}]

Acronyms

ASHP – air source heat pump
COP – coefficient of performance
GHP – geothermal heat pump
HDPE – high density polyethylene
SWHE – surface water heat exchanger
SWSHP – Surface water source heat pump
SCR – silicon controlled rectifier

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