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Original Research Article

Development of Heat Exchanger for Waste Heat Recovery from a Household Refrigerator

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ABSTRACT

A household refrigerator rejects heat (superheat and latent heat) of the refrigerant during the condensation process to the room or environment as waste heat. It is necessary to recover and conserve a reasonable amount of this waste heat and use it for other purposes. To this end, a shell and coiled tube heat exchanger was designed and constructed in this research work. It was used to replace the condenser of an existing refrigerator. The area of the heat exchanger was determined to be 0.09 m^2. On performance evaluation with air flowing through the shell side and refrigerant through the coiled tube, the heat exchanger was able to recover waste heat of 142.16 W.

1. INTRODUCTION

Generally, heat from the condenser side of most household refrigerators that use air-cooled condensers is dissipated to room air. If this heat is not utilized, it simply becomes waste heat. Waste heat is generally the energy associated with the stream of gases and liquids that leaves the boundary of the system and enters into environment (Walawade et al., 2013). This waste heat affects the environmental conditions because as heat in the environment increases, it causes global warming (Fande et al., 2015; Kumar et al., 2017; Pathak et al., 2017). Hence it is necessary that a significant and concrete effort should be made to conserve energy through waste heat recovery (Lakshya et al., 2016). By retrofitting a waste heat recovery system, this waste heat can be recovered and utilised for water heating, warming food and preserving snacks etc (Sreejith et al., 2016). However, not all waste heat is practically recoverable. The strategy of how to recover this heat depends on the temperature of the waste heat sources and on the economics involved in the incorporated technology (Walawade et al., 2013; Muzawar and Panvel, 2016).

The need for the development of a system or device for waste heat recovery from the condenser of vapour compression refrigeration system has been identified by many researchers. Hence, Katkar and Dhale (2014) designed, constructed and tested an integrated heat recovery system which consisted a spiral tube heat exchanger in order to enhance the performance of a domestic refrigerator and simultaneously recovers its waste heat. Choudhari and Choudhari (2015) designed and constructed a heat recovery system from the condenser of 145W Godrej refrigerator. The heat recovery system comprised of a heat exchanger with 50 liters water tank using thermos siphon effect stated by Kumbhar (2017). Sreejith et al. (2016) designed, fabricated and experimentally analysed a waste heat recovery system for domestic refrigerator. The waste heat recovery system (WHRS) was a single tube heat exchanger coiled around and over the air-cooled condenser and compressor and having an inlet for the cooling water and an exit for collecting the hot water. Tanaji et al. (2014) carried out fabrication, experimentation and performance evaluation of waste heat recovery system for a refrigerator and refrigerator-cum-water heater. They found that the system operating under full load condition gives a better coefficient of performance (COP) as compared to no load condition. Ankanna and Reddy (2014) asserted that in the present days heat exchangers are the important engineering systems with wide variety of applications including refrigeration and air-conditioning systems, heat recovery systems among others. They remarked that helical coil configuration is very effective for heat exchangers and chemical reactors because they can accommodate a large heat transfer area in a small space, with high heat transfer coefficients. Kumar et al. (2017 developed an auxiliary condenser called hot box, the hot box was placed between the compressor and condenser and fitted after the compressor outlet and before condenser inlet. Romdhane (2009) developed a system that can recover heat from the condenser of the refrigerator. In their work air-cooled conventional condenser was replaced by another heat exchanger to heat water. They also analyzed the economic importance of the waste heat recovery system from the energy saving point of view. Rahman et al. (2007) developed a heat recovery system which can recover heat from a split air conditioning system. On performance evaluation, they found that the heat recovery system improved the compressor efficiency and at the same time continuously supplied warm water for domestic purpose. Sreejith et al. (2014) designed water-cooled heat exchanger and the system was modified by retrofitting it, instead of the conventional air-cooled condenser by making a bypass line and thus the system can be utilized as a waste heat recovery unit.

Waste heat cannot be fully recovered. Notwithstanding, much of the heat can be recovered by developing and adopting different measures or strategies. Hence the purpose of this research work is to develop a shell and coiled tube heat exchanger of a counter flow type to replace the condenser of an existing refrigerator and use air as the fluid for recovery the waste heat. The waste heat recovered directly could be used as enumerated by Reny et al. (2014).

2. MATERIALS AND METHODS

2.1. Materials

The materials and equipment used for the study include LG refrigerator made in China (model: GC-051SA, rated input: 85W, refrigerant: R134a, air cooled condenser). Others include galvanised iron sheet of 1.5 mm thickness, copper coil, K type thermocouple made in China (model: WRN-230, range: 0-1300 ºC) and a 12V DC blower, made in China with rated power input of 200W.

2.2. Methods

2.2.1. Design specifications of the shell and coiled tube heat exchanger

- Outlet temperature of the refrigerant from the coiled tube of the heat exchanger, T_{or} = 38 ºC (measured outlet temperature of refrigerant from the original condenser of the selected LG refrigerator).
- Inlet temperature of air blown by the blower into the shell of the heat exchanger, T_{ia} = 28ºC (taken to be measured ambient temperature)

- Outlet temperature of air through the shell, T_{fa} = 51°C (estimated by taking mean outlet temperature of air that was blown through a cabin that was made to enclose the original condenser of the selected refrigerator in the laboratory in order to obtain the initial design data)
- Shell and coiled tube designed temperature = 60 °C (estimated by taking mean temperature of a cabin that was made to enclose the original condenser of the selected refrigerator in the laboratory in order to obtain the initial design data).
- Evaporator designed temperature = - 10 °C (specified for the LG refrigerator)

2.2.2. Determination of heat given off by the refrigerant in the coiled tube of the heat exchanger

The thermodynamic cycle of operation of the refrigerator fitted with shell and coiled tube heat exchanger is shown in Figure 1. At point 2 in Figure 1, the refrigerant vapour at low temperature and pressure enters the compressor's cylinder and is compressed to point 3, where it is superheated to high temperature. Then it enters the shell and coiled tube heat exchanger. Firstly, in the shell and coiled tube heat exchanger, the superheated vapour cools to a certain temperature (represented by line 3-3') and it condenses at constant temperature along line 3'-4 and this is where it gives up its latent heat to the condensing medium (air in the shell of the shell and coiled tube heat exchanger).

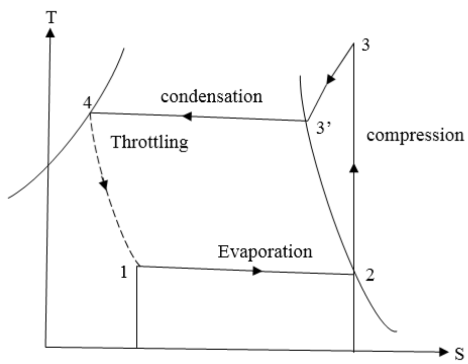

Figure 1: Thermodynamic cycle of operation of the refrigerator fitted with shell and coiled tube heat exchanger

The properties of R134a obtained from thermodynamic tables at the shell and coiled tube heat exchanger and evaporator temperature of 60 °C and -10 °C and at various points in the thermodynamic cycle of operation of refrigerator are (Wark, 1983):

- Enthalpy at point 2, h_2 =241.35kJ/kg
- Enthalpy at point 3', $h_{3'}$ = 275.99 kJ/kg
- Enthalpy at point 1, h_1 = enthalpy at point 4, h_4 = 137.42 kJ/kg
- Entropy at point 2, S_2 = Entropy at point 3, S_3 = 0.9253 kJ/kgK
- Specific heat capacity at constant pressure, C_p = 1.354 kJ/kgK

The entropy at point 3 is expressed as:

$$S_3 = S_{3'} + C_p \text{In}(\left(\frac{T_3}{T_{3\prime}}\right) \quad (1)$$

$$0.9253 = 0.8973 + 1.354 \text{In}\left(\frac{T_3}{333}\right)$$

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$\text{In}\left(\frac{T_3}{333}\right) = 0.02068$

$T_3 = 333e^{0.02068}$

$T_3 = 340\text{K}$

Hence, the inlet temperature of the refrigerant, T_3 or T_{ir} = 340K or 67°C.

The enthalpy at point 3 is expressed as:

$$h_3 = h_{3'} + C_p(T_3 - T_{3'}) \tag{2}$$

$h_3 = h_{3'} + C_p(340-333)$

$h_3 = 275.99 + 1,354(340-333) = 285.468\text{kJ/kg}$

$$\text{Refrigerating effect, } R_n = h_2 - h_1 \tag{3}$$

$R_n = 241.35-137.42 = 103.93\text{kJ/kg}$

Workdone:

$$W = h_3 - h_2 \tag{4}$$

$W = 285.47 - 241.35 = 44.12\text{kJ/kg}$

The coefficient of performance (COP) of the refrigerator is expressed as:

$$\text{COP} = \frac{R_n}{W} \tag{5}$$

$\text{COP} = \frac{103.93}{44.12} = 2.36$

The input power to run the compressor is given as

$$P_c = \dot{m}(h_3 - h_2) \tag{6}$$

Where Pc = 85W and $\dot{m}$ = mass flowrate of the refrigerant

$85 = \dot{m}\,(285.47 - 241.35) \times 10^3$

$\dot{m} = \frac{85}{44120} = 0.0019\text{kg/s}$

The available heat, Q at the shell and coiled tube heat exchanger that would have been wasted or rejected to the environment through the condenser of the refrigerator is given as

$$Q = \dot{m}(h_3 - h_4) \tag{7}$$

$= 0.0019\,(285.47 - 137.42) \times 10^3 = 281.30\text{W}$

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2.2.3. Determination of mass flow rate of air from the blower

From Steam Tables, for T_{ia} = 28^{0}C, with relative humidity of 75%, the partial pressure of the saturated air P_{vs} = 0.0378 bar (Wark, 1983).

Partial pressure of water vapour in the air, $P_v = \Phi P_{vs}$ (8)

Where Φ is the relative humidity of the air (75%)

P_v = 0. 75 x 0.0378 = 0.0284 bar

From Steam Table (Wark, 1983) the dewpoint temperature, t_{dp} of the air corresponding to 0. 0284bar is 24.1^{o}C

Specific humidity:

$$\psi = \frac{0.622 P_v}{P_t - P_v} \quad (9)$$

Where P_t is the total pressure of moist air = 1.0132bar

$$\Psi = \frac{0.622 \times 0.0284}{1.0132} = 0.0179 \text{kg/kg of dry air}$$

According to Rajput (2009), enthalpy of air h_a is given as:

$$h_a = 1.005 t_{db} + \psi(2500 + 1.88 t_{db}) \quad (10)$$

Where t_{db} = Dry bulb temperature = T_{ia} = 28 ^{o}C

The enthalpy of inlet air h_{ai} at t_{db} = 28^{o}C is given as:

h_{ai} = 1.005 x 28 +0.0179(2500 +1.88 x 28) = 73.832 kJ/kg of dry air

For the air at t_{db} = T_{fa} = 51^{o}C, the enthalpy, h_{fa} is given as:

H_{fa} = 1.005 x 51 +0.0179(2500 + 1.88 x 51) = 97.721 kJ/kg of dry air.

The available heat, Q at the shell and coiled tube heat exchanger that would have been wasted or rejected to the environment through the condenser of the refrigerator = Heat gained by the air in the shell of the heat exchanger.

$$Q = \dot{m}_a(h_{fa} - h_{ai}) \quad (11)$$

Where $\dot{m}_a$ is the mass flowrate of air

$$281.30 = \dot{m}_a(97.721-73.832) \times 10^3$$

$$\dot{m}_a = \frac{281.30}{23889} = 0.0118 \text{kg/s}$$

Hence, a DC blower with mass flowrate of 0.0118kg/s was selected to provide airflow for the heat recovery system.

2.2.4. Determination of overall heat transfer coefficient of the shell and coiled tube heat exchanger

The overall heat transfer coefficient, U is given by Vinayak et al. (2016) as:

$$U = \frac{1}{\frac{1}{h_r}+\frac{t_w}{k_w}+\frac{1}{h_a}+f_1+f_2} \tag{12}$$

Where h_r and h_a are the heat transfer coefficients of the refrigerant in the tube and air in the shell of the heat exchanger respectively, t_w is the wall thickness, k_w is the thermal conductivity of the wall material (heat transfer surfaces), f_1 and f_2 are the fouling resistances of both sides. In line with the work of Vinayak et al. (2016), h_r and h_a are evaluated at mean temperature of T_{ia} and T_{or} that is:

Mean temperature:

$$T_m = \frac{T_{ia}+T_{or}}{2} \tag{13}$$

Where T_{or} is the outlet temperature of the refrigerant = 38 ^oC

$$T_m = \frac{28+38}{2} = 33\,^oC$$

From Steam Tables the properties of air at 33^{o}C are (Wark, 1983):

- h_{fg} = 310.24 kJ/kg
- ρ_a = 1.143 kg/m^3
- C_P = 1.005 kJ/kgK,
- k = 0.0268W/m^{o}C
- μ =1.90 x 10^{-5} kg/ms

The properties of refrigerant (R134a) at 33^{o}C are:

- h_{fg} = 164.12 kJ/kg
- ρ_r = 1155.1kg/m^3
- k= 0.0146W/m^{o}C
- μ =2.40 x 10^{-4} kg/ms

The heat transfer coefficient of the refrigerant, h_r is given by Arora (2000) as:

$$h_r = 0.725\left[\frac{k_f{}^3\rho_f{}^2 g h_{fg}}{N D_i \mu_f \Delta T}\right]^{1/4} \tag{14}$$

$$h_r = 0.725\left[\frac{0.0146^3\times1155.1^2\times9.81\times164.12\times10^3}{1\times0.005\times2.40\times10^{-4}\times56}\right]^{1/4}$$

$$h_r = 407.17\,\mathrm{W/m^2K}$$

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According to Rajput (2009) and Arora (2000), Nuselt number is expressed as:

$$Nu = \frac{hD}{K} = 0.023(Re)^{0.8}\,Pr^{0.4} \tag{15}$$

According to Arora (2000), Prandtl number is expressed as:

$$Pr = \frac{C_p\mu}{K} \tag{16}$$

$$Pr = \frac{1005\times1.9\times10^{-5}}{0.0268} = 0.71$$

The Reynold's number:

$$Re = \frac{\rho u D_0}{\mu} = \frac{4\dot{m}}{\mu\pi D_0} \tag{17}$$

$$Re = \frac{4\times0.0118}{1.90\times10^{-5}\times\pi\times0.0065} = 121{,}654$$

The heat transfer coefficient of the air is expressed as:

$$h = h_a = \frac{0.023(Re)^{0.8}\times Pr^{0.4}\times K}{D_0} \tag{18}$$

$$h_a = \frac{0.023(121654)^{0.8}\times0.71^{0.4}\times0.0268}{0.0065} = 967.28 W/m^2K$$

t_w =0.75mm and k for copper = 385W/mK

$$U = \frac{1}{\frac{1}{407.17}+\frac{0.00075}{385}+\frac{1}{967.28}+0.0004+0.0002} = 244.39\ W/m^2K$$

2.2.5. Determination of the total area required for heat recovery

$$Q = UALMTD \tag{19}$$

Where LMTD is the logarithmic mean temperature difference and it is expressed for counter flow as:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{20}$$

Where:

$$\Delta T_1 = T_{ir} - T_{fai} \tag{21}$$

$$\Delta T_1 = 67 - 51 = 16^oC$$

$$\Delta T_2 = T_{or} - T_{ia} \tag{22}$$

$$\Delta T_2 = 38 - 28 = 10^oC$$

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$$\text{LMTD} = \frac{16-10}{In\left(\frac{16}{10}\right)} = 12.77\ ^{o}\text{C}$$

$$281.30 = 244.39 \times A \times 12.77$$

$$A = \frac{281.30}{244.39 \times 12.77} = 0.09\text{m}^2$$

2.2.6. Determination of the length of copper coil for the coiled tube

The area of heat recovery, A is given as:

$$A = \pi D_0 L \tag{23}$$

Where D_0= 0.0065m (external diameter of the copper coil which is the same with the external diameter of the original condenser pipe of the refrigerator)

$$0.09 = \pi \times 0.0065 \times L$$

$$L = \frac{0.09}{\pi \times 0.0065} = 4.4\text{m}$$

2.2.7. Determination of height or length, L_s of the cylindrical shell needed to accommodate the coiled tube

According to Bonafoni and Capata (2015), the length of the cylindrical shell L_s is expressed as:

$$L_s = Np \tag{24}$$

Where p = pitch of the coil = 0.03m (Sagar 2014) and N = number of turns of the coil = 14 (chosen in line with Patil et al. (2018)).

$$L_s = 14 \times 0.03 = 0.42\text{m}$$

2.2.8. Determination of the shell diameter

The shell equivalent diameter, D_e is given by Bonafoni and Capata (2015) as:

$$D_e = \sqrt{\frac{4\dot{m}}{\pi \rho u_a}} \tag{25}$$

Where u_a is the velocity of the air which is 1.4m/s

$$D_e = \sqrt{\frac{4 \times 0.0118}{\pi \times 1.13 \times 1.4}} = 0.097\text{m}$$

The coil volume, V_c is given by Bonafoni and Capata (2015) as:

$$V_c = \frac{\pi d^2{}_c\, l_c}{4} \tag{26}$$

Where d_c = coiled tube internal diameter = 5mm

$$V_c = \frac{\pi \times 0.005^2 \times 4.4}{4} = 0.000864 m^3$$

The volume available, V_a for the air is given by Bonafoni and Capata (2015) as:

$$V_a = \frac{\pi D^2{}_e l_c}{4} \tag{27}$$

$$V_a = \frac{\pi \times 0.097^2 \times 4.4}{4} = 0.0075 m^3$$

The shell volume V_s is given as:

$$V_s = V_c + V_a = \frac{\pi D^2{}_s l_s}{4} \tag{28}$$

$$0.000864 + 0.0325 = \frac{\pi D_s{}^2 \times 0.42}{4}$$

$$D_s = \sqrt{\frac{4 \times 0.033364}{\pi \times 0.42}} = 0.32 m$$

2.3. Construction of the Heat Exchanger

Based on the design calculations, a copper coiled tube of 4.4 metre length with internal and external diameters of 5mm and 6.5mm (the same with the internal and external diameter of the original condenser pipe of the refrigerator) was constructed. It was then enclosed in a stainless-steel cylindrical tank of diameter, 0.32m and length 0.42m. The stainless-steel tank was insulated with fibre glass to reduce heat loss and enclosed in galvanized sheet cylindrical tank. The fibre glass was held in place by galvanized sheet cylindrical tank. The entire assembly constituted the shell and coiled tube heat exchanger. The original condenser of LG refrigerator was removed and the constructed shell and coiled tube heat exchanger was fitted to it. The cross-section of the shell and coiled tube heat exchanger is shown in Figure 2 and the assembly drawing of the refrigerator fitted with the shell and coiled tube heat exchanger is shown in Figure 3.

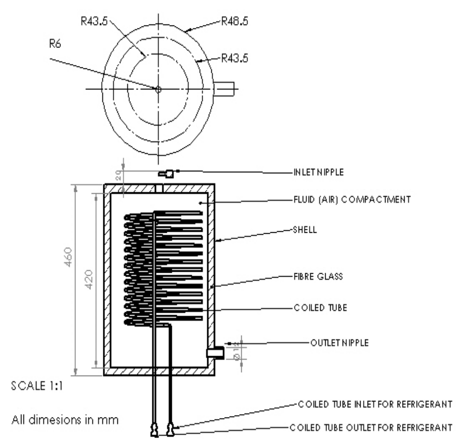

Figure 2: A cross –section of the shell and coiled tube heat exchanger

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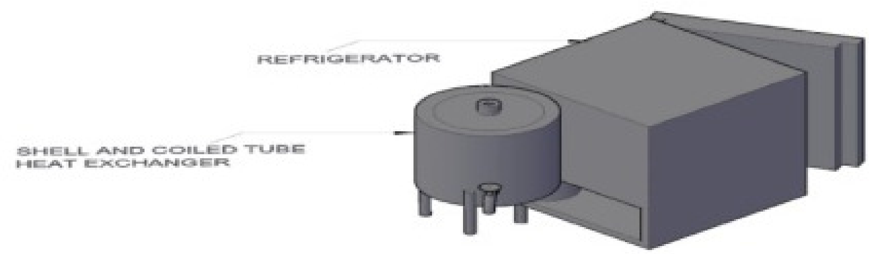

Figure 3: Assembly drawing of fitted shell and coiled tube heat exchanger to the refrigerator

2.4. Performance Evaluation

The refrigerator fitted with the shell and coiled tube heat exchanger was powered and allowed to run for ten minutes for the system to normalize. A DC blower was used to blow air through the inlet nipple into the shell and coiled tube heat exchanger and exited through the outlet nipple. The inlet and outlet air temperatures were measured with the aid of K type thermocouple in an interval of five minutes. A stop watch was used to measure the interval of time for sixty minutes.

2.5. Determination of Recovered Actual Heat

Given the following:

- Air flow rate, $\dot{V}$ = 71.6x$10^{-4}$$m^3$/s (measured)
- Density of air at 29^0C, ρ_a = 1.17Kg/m^3
- Specific heat capacity of air, C_P = 1.005 kJ/kgK
- Initial temperature of air entering the shell and coiled tube heat exchanger, θ_1= 29^0C (measured)
- Final temperature of air leaving the shell and coiled tube heat exchanger after 60 minutes of operation of the refrigerator, θ_2 = 45.9^0C (measured)
- Duration of operation of the refrigerator = 60minutes.

Mass flowrate of air through the shell and coiled tube heat exchanger, $\dot{m}_{a1} = \rho_a\dot{V}$ = 1.17 x 71.6x10^{-4} = 83.7 x 10^{-4} kg/s

The actual heat absorbed by the air = $\dot{m}_{a1}C_P(\theta_2 - \theta_1)$ = 83.7 x 10^{-4}x 1005(45.9-29) = 142.16W

Actual heat absorbed by air = Actual heat recovery achieved.

Actual heat recovery achieved, Q_{ac} = 142.16W.

3. RESULTS AND DISCUSSION

The inlet temperature of air blown by the blower into the shell of the heat exchanger was 29ºC and it was found to be 1 ºC greater than the designed specification of 28 ºC and outlet temperature of air through the shell was 51 ºC and it was found to be 5.1 ºC less than the design specification of 45.9 ºC. These slight variations indicate an effective design of the shell and coiled tube heat exchanger. The actual heat recovery achieved by the shell and coiled tube heat exchanger was 142.16W and the coefficient of performance of the refrigerator was found to be 2.36. The variation of inlet air and outlet air temperatures with time is shown in Figure 4. It can be seen from Figure 4 that the inlet and outlet air temperatures varied with time. The outlet air was the medium that gained the heat given off by the refrigerant in shell and coiled tube heat exchanger which replaced the original condenser of the refrigerator. Maximum outlet air temperature was found to be 52.2^{o}C after twenty five minutes of operation of the refrigerator as evident in Figure 4. This has shown that much heat was given off by the process which implies that the refrigerant released much heat to the air in the shell side.

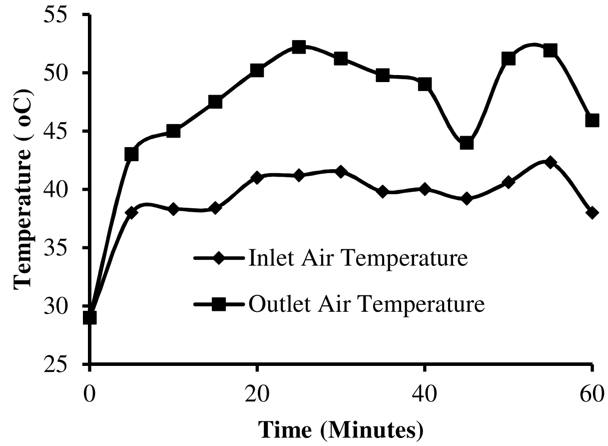

Figure 4: Variation of inlet air and outlet air temperatures with time

4. CONCLUSION

The recovery of heat from the condensation process of a refrigerator can lead to numerous advantages both from an economic and environmental point of view. Heat is recovered by increase in temperature of the inside of shell and coiled tube heat exchanger which resulted in raising the temperature of air that was blown through it. In the present situation of energy crisis, the combination of both the utilities (refrigerator and food and beverage warmer or water heater) can prove to be efficient which will save substantial energy.

5. ACKNOWLEDGMENT

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6. CONFLICT OF INTEREST

There is no conflict of interest associated with this work.

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