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Heat Transfer Efficiency of Different Composite Insulators

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Abstract

This research aims to investigate the thermal performance of different thermal composite insulators, wrapped around a closed-loop copper pipe (CLP). To achieve this aim a system was designed and manufactured. It is consisted of closed water tank insulated by Rock Wool, and supplied with two electric heaters, two thermostat, a flow meter, a water pump, digital temperature scales, and four series of (CLP).

Six insulators were prepared namely; composites of Impregnated Fiberglass with Elastoclad and foaming Rubber (FER), Impregnated Fiberglass with Elastoclad resin and Polymeric Membrane (FEM), Impregnated Fiberglass with Polyurethane thermoset resin and Foaming Rubber (FUR), Impregnated Fiberglass with Polyurethane thermoset resin and Polymeric Membrane (FUM), Fiberglass woven tape (F) , and foaming rubber tape (R). Thermal conductivities of all composite specimens were measured by Lee's Disc device and their thermal performances were evaluated by measuring inlet and outlet temperature ΔT_w at different flow rates. It was found from all test results that ΔT_w decreased as flow rate increased.

The optimum result was obtained for the (FER) insulator at flow rate 8 L/min where $\Delta T_w = 0.8\ ^oC$ (efficiency $\eta = 99\ \%$).

Thermal efficiency of the prepared insulators was according to the following sequence:

FER > FEM > FUR > FUM >R > F

Keywords: Composite insulators; tube insulation; heat losses; Thermal conductivity

Introduction

Energy consumption is an essential element in development and its considered as a major key in the generation of wealth. With the increase in the cost of the energy and the high energy consumed by some areas such as industrial, building, transportation and agriculture, these sectors especially building have recently received considerable attention on energy consumption because of heat losses, Kaynakli [1]. While increased energy use clearly has many benefits, we are also becoming increasingly aware of the negative environments impacts of energy use. However, more efficient use of energy could reduce the negative impacts of energy consumption, while still allowing the same economic development, Mustafa Omer [2].

Energy efficiency is understood to mean the utilization of energy in the most cost effective manner to carry out a manufacturing process or provide a service, whereby energy waste is minimized and the overall consumption of primary energy resources is reduced. There are many benefits of increased energy efficiency. These can broadly be categorized into financial/economic, environmental and social benefits, Trianni, Cagno, Worrell, Pugliese, and Bohm [3,4].

Insulation materials are extensively used to reduce the heat losses (or gains) from thermal systems like buildings, tubes and ducts, components of HVAC installations, etc. In these systems, the insulation layers account for most of the thermal resistance between the hot (or cold) element/s and the environment, Domínguez-Muñoz, Bond, Abdou, and Broin [5-8].Insulation materials fall into two broad categories: organic and inorganic materials. The organic materials include polystyrene, polyurethane, phenolic foam, and polyethylene foam etc. The inorganic materials include mineral wool, calcium silicate, cellular glass, microporous silica, magnesia, ceramic fibre, vermiculite and perlite, Persson, Lingbin, Wei, Al-Ajlan, and Wei [9-13]. Since, the main function of insulation is to reduce the heat transfer, the insulation material must have the appropriate characteristic to retard the transport of heat occurred by conduction, convection and radiation. Conduction loss in insulation is negligible, Keçebas [14].

Various thermal insulation systems taking advantages of different types of thermal insulation materials on both organic (expanded plastics, wood, wool, cork, straw, and technical hemp) and inorganic basis (foamed glass, glass and mineral fibers) are being designed and tested, Pavlik [15]. Besides, most of the available studies focus on insulating buildings, Uyguno_glu, Kaur, Al-Turki, and Tenpierik [16-19] and cold stores, Soylemez, and Kecebas [20, 21] because of the large potential for energy savings. These studies consider the flat plate or slab as the geometric configuration, presenting the large areas of roofs and façades. On the other hand, studies to improve thermal insulation for cylindrical geometry are few in spite of the extensive use of tube lines and cylindrical heat exchangers in refineries, chemical industry, district heating/cooling, and power plants, Zaki, Wechsatol, Wechsatol, and Kalyon [22-25].

Therefore, the objective of the present work is to study the thermal performance of different insulators wrapped on a closed-loop copper pipe (CLP) at different flow rates using water as a heat carrier. Electrical heaters are used to supply the (CLP) with hot water. Six insulators were used namely; composites of Impregnated Fiberglass with Elastoclad and foaming Rubber (FER), Impregnated Fiberglass with Elastoclad and Polymeric Membrane(FEM), Impregnated Fiberglass with Polyurethane thermoset resin and Foaming Rubber (FUR), Impregnated Fiberglass with Polyurethane thermoset resin and Polymeric Membrane(FUM), Fiberglass woven tape (F), and foaming rubber tape (R). The results of this study will provide (i) a basis for comparison between the used insulators (ii) environmental impacts caused by heat losses that occur during use of the heat distribution system.

Modeling and Analysis

1. The Structure of the Piping System

In our experiment heat transfer is carried out by the use of hot water flowing through a closed loop system where the heat is convected to the ambient air and then the hot water flows to a storage tank where it's reheated again. Heat loss and temperature change of transfer tube lines are significantly influenced by (i) insulation, (ii) surrounding environment (ambient air) for tube, and (iii) tube structure. The hot water piping system considered in this study is shown in Fig. 1 for a unit length of a long straight conduit segment, installed in a constant environmental temperature and constant thermodynamic properties at an appropriate mean temperature. The hot water for this system is pumped through the tube with a constant velocity under steady-state steady-flow control volume conditions. Pressure drops due to the liquid flow friction is neglected in this study.

2. The heat loss calculation for the piping system

Heat losses occurred from hot water through piping system can be calculated by the following equation.

$$Q = UA\,(T_{AV} - T_A) = UA\,\Delta T = \Delta T/\Sigma R_{thermal} \qquad \ldots (1)$$

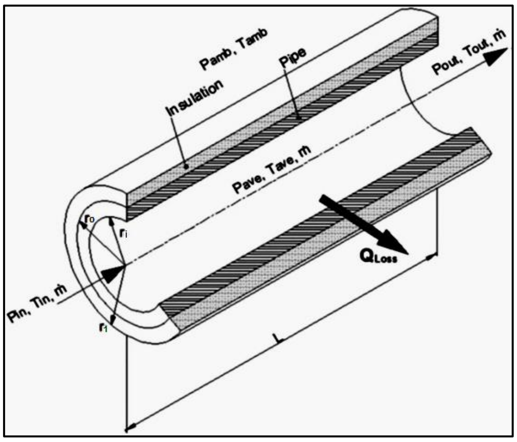


Fig. 1 The hot water tube line with insulation

Where A is the total surface area of a tube, T_A is the temperature of outside air, T_{AV} is the average design temperature of inside fluid, and U is the overall heat transfer coefficient. The total internal resistance of any piping system, $\Sigma R_{thermal}$, is equal to the summation of the surface resistances of convective heat transfer over the inside and outside surfaces of the tube and the total internal resistance of all layers of piping system is given as in Holman, and Cengel [26-27].

$$\Sigma R_{thermal} = R_{water\ convection} + R_{copper\ conduction} + R_{1\text{-}insulation\ conduction} + \ldots R_{n\text{-}insulation} + R_{air\ convection}$$
$$= 1/h_i A_i + \ln(r_O/r_i)/2\pi kL + \ln(r_1/r_O)/2\pi k_1L + \ldots \ln(r_n/r_{n-1})/2\pi k_nL + 1/h_OA_n \qquad \ldots (2)$$

Where r_1, r_2, etc. are insulation layers of piping system radii. The length of the tube is L. The inside surface area of the tube is $A_i = 2\pi Lr_i$ while the outside surface area of the tube is $A_O = 2\pi Lr_O$ and the surface area of the last layer of piping system is $A_n = 2\pi Lr_n$:

In this study, the total internal resistance of un-insulated piping system is

$$\Sigma R_{un\text{-}insulated\ pipe} = R_{water\ convection} + R_{copper\ conduction} + R_{air\ convection}$$
$$= 1/h_i A_i + \ln(r_O/r_i)/2\pi kL + 1/h_OA_O \qquad \ldots (3)$$

And the total internal resistance of insulated piping system is the following form:

$$\Sigma R_{insulated\ pipe} = R_{water\ convection} + R_{copper\ conduction} + R_{1\text{-}insulation\ conduction} + R_{air\ convection}$$
$$= 1/h_i A_i + \ln(r_O/r_i)/2\pi kL + \ln(r_1/r_O)/2\pi k_1L + 1/h_OA_1 \qquad \ldots (4)$$

The convection heat transfer coefficients for the inside and outside surfaces of piping system hi and ho are calculated as [26-28]:

$$Nu = 0.023\,Re^{0.8}\,Pr^{0.3} \qquad \ldots (5)$$

$$h_O = 1.32\,(\Delta T_S/d_O)^{1/4} \qquad \ldots (6)$$

The difference between the overall heat transfer coefficients of un-insulated and insulated piping systems can be written as

$$\Delta U = U_{un\text{-}insu.\ pipe} - U_{insu.\ pipe.} \quad \ldots (7)$$

For outer radii greater than the critical value any increase in insulation thickness will cause a decrease in a heat transfer [26]. So for any layer of insulation the outer radius must be larger than the critical radius which is defined as

$$r_c = k_{insulation} / h_O \quad \ldots (8)$$

Where r_c is the critical radius of insulation.

$$Q = (T_{AV} - T_A) / \Sigma R_{thermal} \quad \ldots (9)$$

$$Q_{un\text{-}insu.\ pipe} = (T_{AV} - T_A) / \Sigma R_{un\text{-}insu.\ pipe} \quad \ldots (10)$$

$$Q_{insu.\ pipe} = (T_{AV} - T_A) / \Sigma R_{insu.\ pipe} \quad \ldots (11)$$

$$Q_{save} = Q_{un\text{-}insu.\ pipe} - Q_{insu.\ pipe} \quad \ldots (12)$$

$$\text{Heat loss} = (Q_{save}/Q_{un\text{-}insu.\ pipe}) * 100\% \quad \ldots (13)$$

Where T_A is the temperature of the ambient air, T_{AV} is the average temperature of the fluid inside the tube ($T_{AV} = (T_{in} + T_{out})/2$)

Experimental

1. Materials

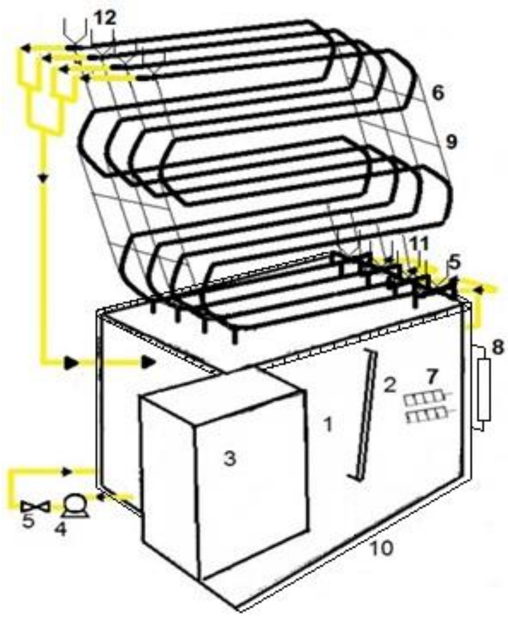
The materials which are used in the experiment as insulators are: E-type fiberglass, bitumen membrane, rubber insulation foam tape, thermoset polyurethane (PU) resin, elastoclad, and rock wool. All the specifications are listed in the Table 1:

Table 1: Specification of the materials

Materials	Specifications
E-type fiberglass	• Woven fabric of 0.25 mm thickness • Aerial weight 270 g/m^2 • density $2.5 gm/cm^2$
Elastoclad resin	• 100% acrylic copolymer • Heat reflective • Volume solids Approx. 60%, • Thermal Conductivity 0.19 W/(m.K) • Specific gravity 0.86 ± 0.05 • Cure time 2-4 hours
Rock wool	• Not burn and bear high temperature up to 750°C. • Coefficient of thermal conductivity is low 0.035 W/m .K
Self-adhesive rubber foam tape	• Density (ρ) $60\sim100 kg/m^3$ • Temperature Range (-24°c -+170°C)
polyurethane (PU)	• unfilled thermoset resin • Specific Gravity1.03-1.5 • Water Absorption (% weight increase)0.2 - 1.5 • Thermal Conductivity (W/m-K) 0.209

2. Experimental Set Up and Procedure

The experimental setup as shown in Fig. 2 consisted of galvanized water tank of dimensions (1*0.5*0.6) m insulated by rock wool. Four 5m long copper tubes (99% purity) are used in the experiment. The tubes are bended by a tube processing machine (TUBOMAT), tub 642 type, manufactured by TRACTO-TECHNIK GmbH & Co.KG and made in Germany. The internal diameter of the tube is 0.017m and the outer diameter is 0.019m. The thermal conductivity of the copper tube (k) is (385W/m.K). Other equipments like two electric heaters of (3000 Watt each.) as heat sources, two thermostats were used in the experiment. A flow meter (blue white, California, USA) of (1-4 Gpm, and 2-16 Lpm) to measure water flow rate in the copper tubes. Electrical water Pump, Marques, made in china, model MKP 60-1, (voltage 220, 50 Hz, power 0.5Hp, 370 watt, maximum 40 Lpm) is used to pump water from the tank.



1. galvanized tank of hot water
2. indicator of the level of water
3. temperature control board
4. water centrifugal pump
5. gate valve
6. four copper tubes
7. electric heater
8. water flow meter
9. connections between the tubes
10. tank rock wool insulator
11. input screw type thermocouple sensor
12. output thermocouple sensor

Fig. 2 Schematic diagram of the apparatus

Digital temperature scales (Redlion, USA) with eight thermocouple sensors (Autonics sensor, Korea) are used to measure water temperature at the inlet and outlet of the copper tubes. Five gate valves are used to control the flow of water.

3. Preparations of Composite Insulators

Different composite insulators were prepared and used for insulating the tubes and also machined to desire final dimension for thermal conductivity test as mentioned in 3.4.

3.1 Preparation of FER

A suspension of Elastoclad- in water was prepared and stirred well for 10 min. Fiber glass woven tape was impregnated with the above suspension. The suspension was used to impregnate glass fiber tape on a weight basis (60 /40). The suspensions were applied with a brush and the solvent was allowed to evaporate for 5-6 hours under ambient conditions. The dried impregnated tape so prepared was stacked over rubber foam tape, and placed between Stainless steel plates.

3.2 Preparation of FEM

Fiber glass woven tape was impregnated and prepared as described in 3.1 .The bitumen membrane was shredded into tapes of 6 cm and 1500 cm length. The dried impregnated tape so prepared was stacked over the bituminous membrane tape – using a flame to melt the polyethylene layer to achieve good adhesion - and placed between stainless steel plates.

3.3 Preparation of FUR

Fiber glass woven tape was impregnated with polyurethane thermoset resin on a weight basis 60/40 and the resin was treated as in 3.1.

3.4 Preparation of FUM

Fiber glass woven tape was impregnated with polyurethane thermoset resin, dried and stacked over the shredded bituminous membrane tapes of 6 cm and 1500 cm length – using a flame to melt the polyethylene layer to achieve good adhesion. The dried impregnated tape so prepared was stacked over rubber foam tape, and placed between stainless steel plates.

4. Measurement of Thermal Conductivity

Thermal conductivity of all specimens was measured by using Lee's Disc device (Griffin & George/England) in University of Technology. Three copper discs (A, B, C) of 40 mm diameter x 12.25 mm thickness each are used in Lee's Device as shown in Fig. 3. The insulator samples were machined to diameter 40 mm and thickness 3mm.The value of (e) and (k) were calculated according to the following equations. The different insulator composites; their thermal conductivity data and designation are shown in Table 2.

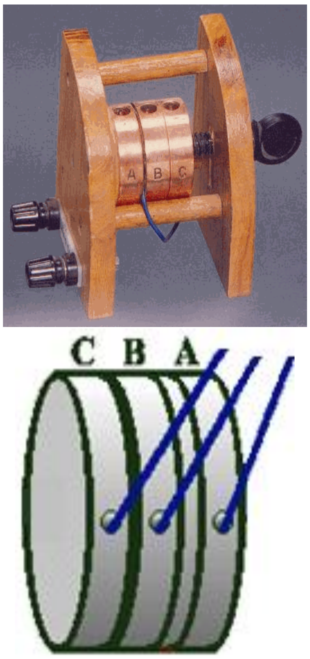


Fig. 3: Three copper discs (A, B, C) in Lee's Disc Device

Table 2: Insulators types, their designation and thermal conductivity values

No	Insulators Types	Designation
1	Impregnated Fiberglass with Elastoclad and foaming Rubber	0.01353
2	Impregnated Fiberglass with Elastoclad and Polymeric Membrane	0.0635
3	Impregnated Fiberglass with Polyurethane thermoset resin and Foaming Rubber	0.1383
4	Impregnated Fiberglass with Polyurethane thermoset resin and Polymeric Membrane	0.1662
5	Fiberglass Roven tape	0.0396

$$Q = IV = \pi r^2 e(T_a + T_B) + 2\pi r e[d_A T_a + d_S \tfrac{1}{2}(T_a + T_B) + d_B T_B + d_C T_C] \quad \dots (14)$$

$$k\left(\frac{T_B - T_a}{d_S}\right) = e[T_a + \frac{2}{r}\left(d_A + \frac{1}{4} d_S\right) T_a + \frac{1}{2r} d_S T_B] \quad \dots (15)$$

Where

Q: Amount of heat transfer (W).

(T_a,T_B,T_C):Temperature of discs (A,B,C) respectively (°C).

(d_A, d_B, d_C): Thickness of discs (A, B, C) respectively (mm).

d_S: Thickness of sample (mm).

r: Radius of disc (mm).

I: Current (Amp).

V: Supply Voltage (volt).

e: Convection heat transfer coefficient ($W/m^2.°C$).

5. Measurement of Heat Loss

The pump turned on and left for 15 minutes to make sure that all four tubes were filled in water. The water heated up to 80 °C and several runs were performed. In each run the inlet and outlet temperature of hot water at different flow rates for all four tubes were taken, by adjusting the flow rate valve and by using RTD (resistance temperature Detector) sensor.

Results and Discussion

Different resins were used as a matrix polymer for the impregnation of the fiber glass:

1- Elastoclad is a 100% acrylic waterproofing and heat reflective coating which is formulated with hollow core ceramic microspheres to dissipate and reflect heat and also provides a long lasting elastomeric waterproof resin.

This resin as indicated by the supplier has many advantages:-

Deadens sound, resists hail damage, and is fungus resistant.

- Resistant to chemicals and deterioration from hot weather conditions.
- Reduces radiant heat transmission from external surfaces by up to 92%.
- Protects metals from rust corrosion
- The cured waterproof membrane is elastomeric and can tolerate substrate movement without cracking or flaking.

Polyurethane thermoset resin is also well known for its low thermal conductivity

In the experiment, several runs were done and many data points were considered to draw different curves that illustrate the relations between the used parameters. The calculations were done as in the appendix.

From Fig. 4, it can be noticed that the least value of temperature difference between the inlet and outlet water flow is achieved when using FER insulator (impregnated fiberglass with Elastoclad and foaming rubber).

Table 3 and Fig. 5 show the temperature difference of inlet and outlet water of different insulated tubes at different flow rates of hot waters in tubes. For each run, the best sequence of insulators is FER, FEM, FUR, FUM, R and F. It's also seen that as the flow rate increases the temperature difference decreases for the same insulators.

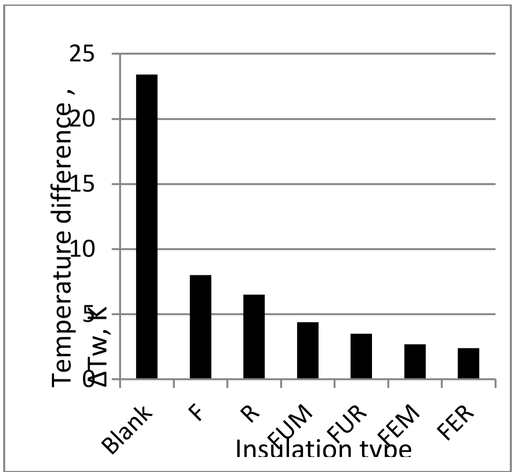


Fig. 4: Temperature difference of different insulators

Table (3) Temperature difference of inlet and outlet water in tubes of different insulators at different flow rate, K

V_R L/min	ΔT_w (K)						
	Blank	F	R	FUM	FUR	FEM	FER
0.5	29.6	12	8	6.1	5.2	5.1	5.0
2.0	27.4	9.5	6.6	5.3	4.5	4.1	3.3
4.0	23.4	8	6.5	4.4	3.5	2.7	2.4
6.0	18.5	7.1	6.4	4.3	2.7	2.3	1.6
8.0	17.9	6.5	6.2	3.7	2.2	1.5	0.8

Fig. 6 shows and compares between the temperature difference values of the blank tube (without insulation) and the tube with FER insulation which has the smallest temperature difference. Fig. 7 gives the relation between the temperature difference of the blank tube and the tube of FER insulation.

Table 4 shows the thermal conductivity, thermal resistance, the heat loss through tubes, the heat saved compared with an un-insulated tube (blank one), and the percent reduction which is calculated for the six insulators used. Its seen that FER insulated tube has 77.7 % reduction in heat loss compared with the blank tube, so it is considered the best between the others.

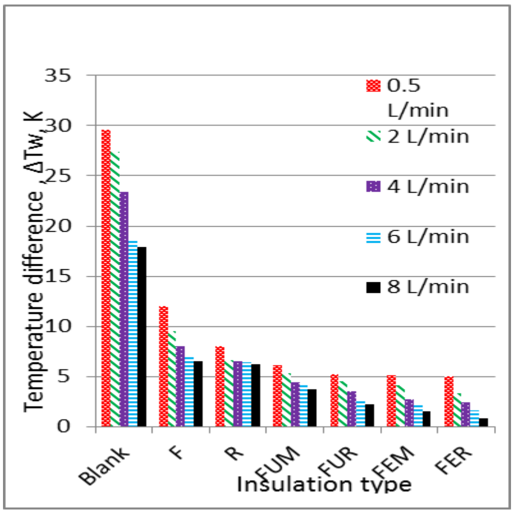


Fig. 5: Temperature difference of different insulators at different flow rate

$$\Delta T_w \text{ (blank, without insulation)} = 3.0855\ \Delta T_w \text{ (FER)} + 15.276 \quad \ldots(16)$$

Where $\Delta T_w = T_{in} - T_{out}$

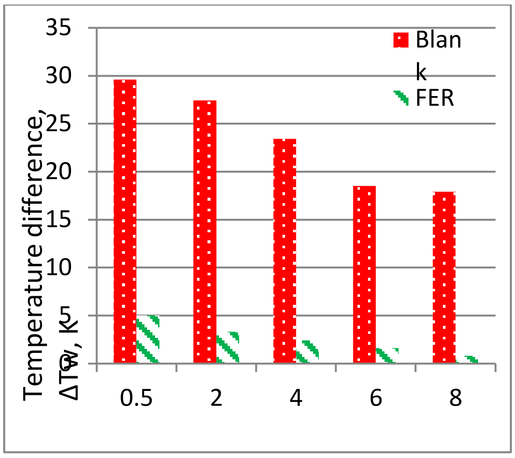


Fig. 6: Comparison of temperature difference between the blank and FER tube

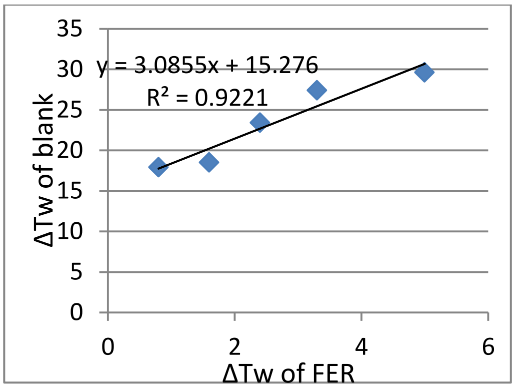


Fig. 7: A plot of ΔT (blank tube) versus ΔT (FER insulated tube)

Table 4: Saved heat and percent reduction for different insulators at 1 L/min

Insulation type	k, (W/m.K)	Resistance R, **(K/W)**	Heat loss (W**)**	Heat save	% reduction
FER	0.013	1.4936	26.5	91.8	77.7
FEM	0.02	0.97	37.7	80.6	68.12
FUR	0.08	0.24	100.92	17.57	14.8
FUM	0.085	0.228	103.9	14.5	12.2
R	0.09	0.21	108.39	10.05	8.49
F	0.095	0.204	109.8	8.6	7.2
Blank	385	0.37	118.45	0.0	0.0

Fig. 8 shows the relation between the saved heat and the percent reduction with respect to the types of insulators used. The best insulator is FER type where the heat saved and the percent reduction is 91.8 W and 77.7 % respectively.

Fig. 9 shows the heat loss for all types of insulators beside the blank tube without insulation. The minimum heat loss is 26.5 W which obtained when FER insulator is used while the maximum heat loss is 118.45 W for the un-insulated tube. It's noticed from the figure that the two composite insulators (FER and FEM) are good insulators because of using the elastoclad suspension for impregnation.

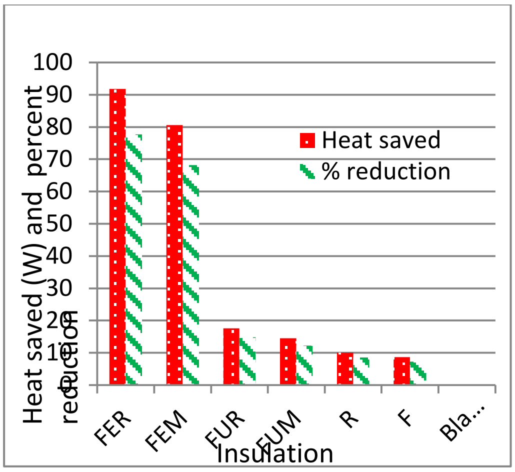


Fig. 8 Heat saved and percent reduction for several insulators

At different flow rate (0.5, 2.0, 4.0, 6.0, 8.0 L/min) and for different insulators (FER, FEM, FUR, FUM, R, M), the efficiency of each one is calculated according to the percent of the outlet temperature to the inlet temperature and it's found that FER efficiency (99 %) is the best at 8 L/min flow rate as shown in table (5) and Fig. (10).

$$\eta = (\text{outlet temp.} / \text{inlet temp.}) * 100\% \quad ...(17)$$

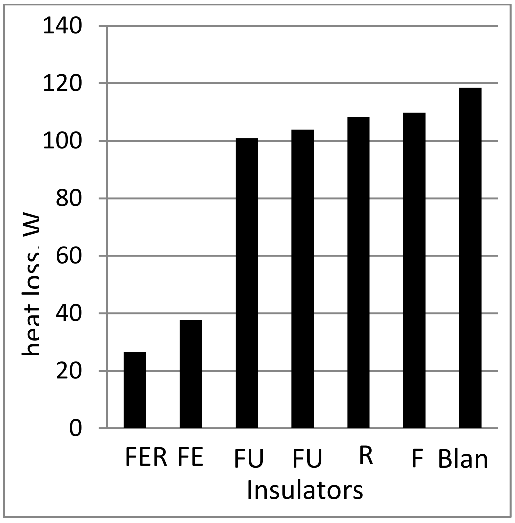


Fig. 9 Gradual heat loss of all insulators and the blank tube

Table 5: The efficiency η of different insulators at different flow rate, (%)

V_R L/min	Blank η	F η	R η	FUM η	FUR η	FEM η	FER η
0.5	63	85	90	92.37	93.5	94.37	93.75
2.0	65.75	88.12	91.75	93.37	94.37	94.87	95.87
4.0	70	90	91.87	94.5	95.62	96.62	97
6.0	76.87	91.12	92	94.62	96.62	97.12	98
8.0	77.62	92.87	92.25	95.37	97.25	98.12	99

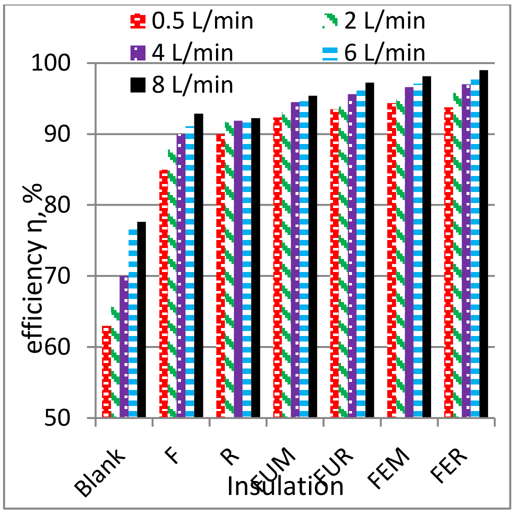


Fig. 10: The efficiency of different insulators at different flow rate

Conclusions

The appropriate insulation must be selected on the basis of temperature, thermal conductivity and other limiting factors that might limit application. The appropriate thickness must be determined for the particular application.

From the results achieved it was concluded that:

1- Thermal insulators are necessary materials for minimizing heat loss and saving energy.
2- Temperature difference decreases as the flow rate of hot water increases.
3- The efficiency and percent heat loss reduction of insulation for the prepared insulators was according to the following sequence:
   FER > FEM > FUR > FUM > R > F
4- There are three cases of heat transfer in tubes in the experiment
   i. Forced convection heat transfer in hot water inside the tube.
   ii. Conduction heat transfer through the wall of the copper tube and through the insulation layer.
   iii. And finally the free convection heat transfer from the outer surface of the tube to the ambient cold air.
5- The emissivity of the copper material is very small (0.018), so the radiation heat loss from the outside surface of the tubes was neglected.
6- The insulators FER and FEM show high efficiency of insulation which explain the effect of using Elastoclad.

Nomenclature Symbols

A	total surface area of tube (m^2)
A_C	cross sectional area (m^2)
A_i	inside surface area of the tube (m^2)
A_O	outer surface area of the tube (m^2)
A_n	outside surface area of the last layer of piping system (m^2)
d_A, d_B d_C	Thickness of discs (A, B, C) respectively (mm)
d_S	Thickness of sample (mm)
e	Convection heat transfer coefficient in Lee's disc device (W/m^2. °C).
F	Fiberglass Roven tape
g	gravitational acceleration (m/s^2)
Gr	grashof number
h_i, h_O	heat transfer coefficients of water inside the tube and ambient air (W/m^2.K)
I	Current (Amp)
k	thermal conductivity of the copper tube (W/m.K)
k_1, k_2	thermal conductivities of insulation layers of piping system (W/m.K)
k_i	thermal conductivity of the fluid inside the tube (W/m.K)
k_O	air conductivity (W/m.°K)
Nu	Nusselt number
Pr	Prandtl number
Q	Amount of heat transfer (W)
Q_R	radiation rate of heat loss (W)

R	Foaming Rubber tape
$R_{thermal}$	Thermal resistance (°C/W)
r	Radius of disc (mm)
r_1, r_2	radii of insulation layers (m)
Ra	Rayleigh number
Re	Reynold number
Rp	total internal resistance of any piping system (K/W)
T_a,T_B, T_C	Temperature of discs (A, B, C) respectively (°C)
T_A	temperature of the surrounding ambient air (K)
T_{AV}	average temperature of inside fluid (K)
T_f	film temperature (K)
T_S	temperature of the surface (K)
U	overall heat transfer coefficient ($W/m^2.K$)
u	velocity of water (m/s)
V	Supply Voltage (volt)
V_R	volumetric flow rate (L/min)

Greek symbols

β	volumetric expansion (K^{-1})
Δ	difference between two values
ΔT_S	the temperature difference between the tube surface and the ambient air (K).
μ	dynamic viscosity (kg/m.s)
ν	kinematic viscosity (m^2/s)
ρ	water density (kg/m^3)
Σ	summation
σ	Stefan Boltzman Constant ($W/m^2.K^4$)
Є	emissivity of the copper
η	efficiency of insulator

Subscripts

1	first insulation layer
2	Second insulation layer
c	cross section
i	inside the tube
n	insulation number n
o	outside the tube

Superscript

n	the power value of Prandtl number

Abbreviations

CLP	closed-loop pipe
FEM	Impregnated Fiberglass with Elastoclad and foaming Rubber
FUM	Impregnated Fiberglass with Polyurethane thermoset resin and Polymeric Membrane
FUR	Impregnated Fiberglass with Polyurethane thermoset resin and Foaming Rubber
HVAC	heating, ventilation, and air conditioning

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Appendix

Case Study

Heat loss calculation for a copper tube is:

Thermal conductivity, k = 385 W/m.K

Average surface temperature of the tube, T_S = 65°C (338 K)

Ambient air temperature, T_A = 25°C (298 K)

Average water temperature in the tube, T_{AV} = 70 °C

$\Delta T = T_{AV} - T_A$ = 45 °C

$\Delta T_S = T_S - T_A$ = 40 °C

Inner area of the tube, $A_i = \pi d_i L = \pi (0.017) *5 = 0.267\ m^2$

Outer area of the tube, $A_O = \pi d_O L = \pi (0.019) *5 = 0.298\ m^2$

Outer area of the first insulator layer, $A_1 = \pi d_1 L = \pi (0.035) * (5) = 0.546$ m

tube length = 5m; inner diameter = 0.017 m and outer diameter = 0.019 m

Number of tubes = 4

Radiation heat transfer $Q_R = \sigma \epsilon A_O (T_S^4 - T_A^4)$

$= 5.669 * 10^{-8} * 0.018 * 0.298 (338^4 - 298^4) = 1.57$ W which is neglected because it's a small value

Where

Q_R = radiation rate of heat loss, W

A_O = outer surface area of the tube, m^2

T_S = temperature of the surface, K

T_A = temperature of the surrounding ambient air, K

ϵ = emissivity of the copper surface, 0.018

σ = Stefan Boltzman Constant = $5.669 *10^{-8}\ W/m^2.K^4$

Total heat loss = $Q = UA (T_{AV} - T_A) = UA\Delta T = \frac{\Delta T}{\sum Rth}$

Heat transfer coefficient of hot water inside the tube:

$h_i = 0.023 * Re^{0.8} * Pr^{n}\ k_i/d_i$

n = 0.4 for heating and 0.3 for cooling

For water flow inside the tube at T_{AV} = 70°C, the water properties [26]

$\mu = 4.0678 *10^{-4}$ kg/m.s

$\rho = 977.89\ kg/m^3$

k_i = 0.6638 W/m.K

Pr = 2.5699

V_R (volumetric flow rate) = 1L/min

$u = V_R/A_C = 1 * 10^{-3}/60\ ((\pi/4) (0.017)^2) = 0.073$ m/s

$Re = \rho\ u\ d_i/\mu = 977.89 * (0.073) (0.017)/ 4.0678 *10^{-4} = 3006.58$

Since Re is greater than the critical Re of tubes (2300), the flow is turbulent

$Nu = h_i\ d_i/k_i = 0.023 * Re^{0.8} * Pr^{n} = 0.023 * 3006.58^{0.8} * 2.5699^{0.3} = 18.499$

$h_i = (k_i/d_i) * 0.023 * Re^{0.8} * Pr^{n} = (0.6638/ 0.017) * 0.023 * 3006.58^{0.8} * 2.5699^{0.3}$

$= 722.35\ W/m^2.K$

To calculate h_O for the ambient air with the outside surface area, air properties are found at T_f as [26, 27]:

Film temperature, $T_f = (T_S + T_A)/2 = (65+25)/2 = 45$ °C = 318 °K

Volumetric expansion, $\beta = 1/T_f = 1/318 = 3.144 * 10^{-3}\ K^{-1}$
Gravitational acceleration, $g = 9.81\ m/s^2$
Kinematic viscosity, $\nu = 17.5152 * 10^{-6}\ m^2/s$
Prandtl number, $Pr = 0.704$
The outer insulator surface diameter, $d_1 = 0.035$ m
Air conductivity, $k_O = 0.031$ W/m. K
Grashof number, $Gr = g\ \beta\ (T_S - T_A)\ d_O^3/\nu^2$
$= 9.81 * 3.144 * 10^{-3} * (65 - 25) * 0.019^3/(17.5152 * 10^{-6})^2 = 27583$
Rayleigh number, $Ra = Gr * Pr = 27583 * 0.704 = 19418.46$
Convection heat transfer coefficient at the ambient air outside the tube is calculated according to the following equation:
$h_O = 1.32\ (\Delta T_S/d)^{1/4}$ for $10^4 < Ra < 10^9$
$h_O = 1.32 * (40/\ 0.019)^{1/4} = 8.94\ W/m^2.K$
$Q = (T_{AV} - T_A) / \Sigma R_{thermal} = \Delta T/$ sum of thermal resistances
$\Sigma R_{thermal}$ (un-insulated tube) $= R_{water\ convection} + R_{copper\ conduction} + R_{air\ convection}$
$\Sigma R_{thermal}$ (un-insulated tube) $= 1/h_i A_i + \ln (d_O/d_i) / 2\pi kL + 1/h_O A_O$
$= 1/(722.35 * 0.267) + \ln (0.019/0.017) /2\pi * 385 *5 + 1/(8.94 *0.298) = 0.3799$ °C/W
Total heat loss of un-insulated tube, $Q_{un\text{-}insulated} = 45/ 0.3799 = 118.425$ W
For insulated tube with FER insulation ($k = 0.01353$)
ΣR (insulated tube) $= R_{water\ convection} + R_{copper\ conduction} + R_{1\text{-}insulation\ conduction} + R_{air\ convection}$
$= 1/h_i A_i + \ln (d_O/d_i) / 2\pi\ kL + \ln (d_1/d_O) / 2\pi\ k_1 L + 1/h_O A_1$
$= 1/(722.35 * 0.267) + \ln (0.019/0.017) /2\pi * 385 *5 + \ln (0.035/0.019) / (2\pi *0.01353 *5) + 1/(8.94* \pi * 0.035* 5)$
$= 0.005185 + 0.000009196 + 0.2744 + 0.2$
$= 1.7$ °C/W
Total heat loss of insulated tube, $Q_{insulated} = 45/1.7= 26.43$ W

$Q_{save} = Q_{un\text{-}insulated\ pipe} - Q_{insulated\ pipe} = 118.425 - 26.43 = 91.989$ W
Reduction in heat loss % $= [(Q_{un\text{-}insulated\ pipe} - Q_{insulated\ pipe}) / Q_{un\text{-}insulated\ pipe}] * 100$ %
$= (91.989 / 118.425) *$