Journal of Engineering and Development
www.jead.org
Vol. 05, No. 19, September 2015
ISSN 1813-7822

USING TIO2 NANOFLUID FOR ENHANCING PERFORMANCE OF A WATER CHILLER

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(Received:19/1/2015 ; Accepted:1/4/2015)

Abstract:
In the present work an experimental study is carried out to enhance performance of water chiller system by using nanofluid (water+TiO2) in shell side of chiller with different volume concentration (0.05% , 0.075% and 0.1%) . Experimental work includes two parts; In the first one COP of the system is calculated by using two alternative refrigerants (R-407c and R-134a). In the second part, the optimum COP is fonnd using various concentration of TiO2 nanofluid for system with alternative refrigerant. Experimental results show that the COP of R-134a is better than R407c which will be used in the second part of the present study with nanofluid in shell side of the system. The COP with (0.1%) nanofluid has been found to increase by (8.5%).

Keywords: : *Alternative refrigerant, Nanofluid, COP.*

أستخدام أوكسيد التيتاينوم لتحسين أداء منظومة تثليج الماء

الخلاصة:
تم في هذا البحث دراسة عملية لتحسين اداء منظومة تثليج الماء باستخدام مائع نانوي (ماء+اوكسيد التيتانيوم) خلال غلاف القشرة لمبادل حراري (المبخر) وبتراكيز مختلفة (0.1% and ٠,٠٥،٠,٠٧٥). وتضمن البحث جزئين: في الجزء الاول تم حساب معامل الاداء باستخدام موائع بديلة R134a و R407c . وفي الجزء الثاني تم استخدام المائع النانوي في المبخرلتحسين معامل اداء المنظومة .وقد اوضحت النتائج العملية ان معامل الاداء للمنظومة عند استخدام R134a افضل من R407c ، لذلك استخدم في الجزء الثاني اي مع وجود المائع النانوي، وكذلك اتضح ان معامل الاداء باستخدام (0.1%) من المائع النانوي ازداد بنسبة (8.5%).

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Nomenclature

h : Specific enthalpy(kJ/kg)
k: Thermal conductivity (W/m^{o}C)
m: Mass (kg)
$\dot{m}$: Mass flow rate (kg/s)
Q: Heat transfer (W)
W: Compressor work (W)

Greek symbols

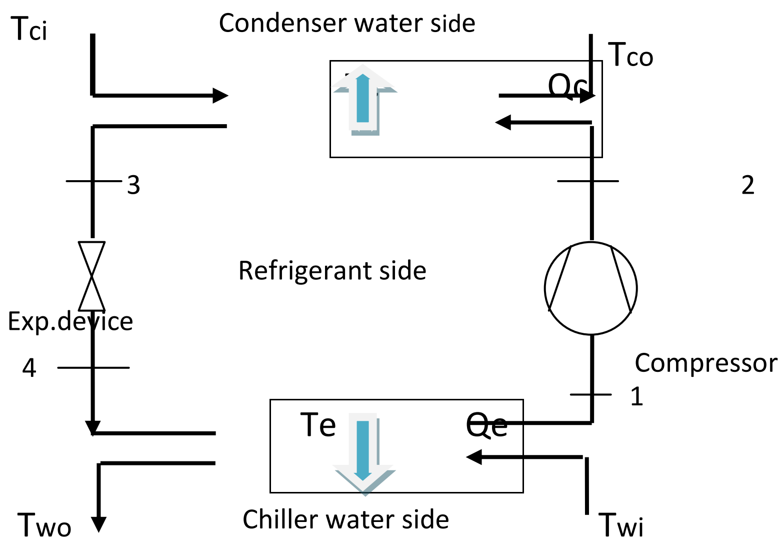
φ_v: Nanoparticle concentration (%vol.)
ρ: Effective density (kg/m^3)

1. Introduction

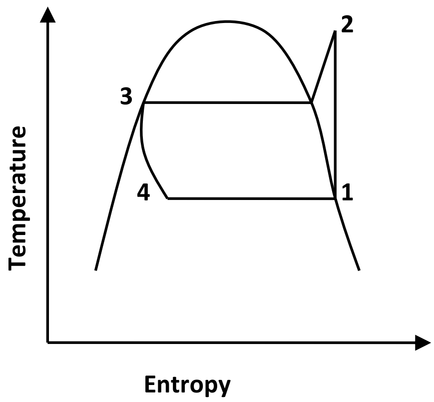
Due to the rapid industrialization which led to exceptional growth in development and technological advancement across the world which in turn led to the emergence of several new concerns thermal systems like water chiller. Air conditioning systems consume large amount of electric power, consequently ways of developing energy efficient refrigeration and air conditioning systems with nature friendly refrigerants need to be explored. Heat transfer fluids have inherently low thermal conductivity that greatly limits the heat exchange efficiency, whereas the effectiveness of extending surfaces and redesigning heat exchange equipment to increase heat transfer rate has reached a limit, so many researchers have started to find new technology.

Vapor-compression water chillers have been widely used to cool water or secondary coolant for air-conditioning and refrigerating applications in both commercial and industrial fields. Figure (1) shows that the main components of a vapor-compression water chiller include compressor and its driver, condenser, throttling device, and evaporator (liquid cooler). The coefficient of performance (COP) for water chillers is defined as the ratio of the evaporator cooling capacity to the compressor input power. Practically, it is more convenient to express chiller performance COP in terms of readily measured water-side data rather than refrigerant-side data. This is especially true for the purposes of performance prediction, evaluation energy-efficient improvements, and fault detection and diagnosis of water chillers [1]. The past decade has seen the rapid development of nanofluid science in many aspects. In recent years, refrigerant-based nanofluids have been introduced as nanorefrigerants due to their significant effects over heat transfer performance. Measurable water-side data include condenser inlet and outlet water temperatures, evaporator inlet and outlet water temperatures, and condenser and evaporator water flow rates. Therefore, developing chiller performance model by using water-side data has been a subject of many studies over the last decade . Chang et al. (1995)[2] studied experimentally the performance of R-134a as alternative to R-22 in the refrigeration systems under various operating conditions. The evaporator water temperature was ranged between (-5 to 10 °C). R-134a offers a (30 to 40 %) decrease in capacity, while the coefficient of performance was close to that of R-22. Greco et al. (1997)[3] studied experimentally the performance of R-407C as a drop in alternative to R-22 in vapor compression plant. The study showed that the coefficient of performance of R-407C was lower than that of R-22 by the range (5 to 17 %). The plant working with R-407C requires higher electric power consumption. Avina (1994)[4] developed a model for a shell and coil heat exchanger (single phase) used in solar domestic hot water system. The model based on the effectiveness NTU method, with a combined heat transfer mode natural and forced convection to determine the heat transfer coefficient over the tubes. The heat exchanger treated as one section, and taking an average fluid temperature at which to find the properties. The helical coil geometry modified to a bundle of tubes in cross flow, each turn assumed to be a straight tube having a length of ($2\pi R$). Mullen et al. (1998)[5] developed modeling of evaporator and condenser finned tube heat exchanger in room air conditioning unit. The condenser was divided into three zones; superheated, two-phase (condensation) and subcooled, each zone was analyzed separately using effectiveness NTU method. The model assumption were uniform air inlet temperature and velocity, the experimental measurement of velocity over the heat exchanger showed, the maldistribution contributes less than a (0.1%) error to the total capacity calculation. Tarrad and Shehhab (2007)[6] developed a model for a finned tube heat exchanger as evaporator for air cooling system. They divided the evaporator in two zones, two-phase (evaporation) and superheated. The model took into account the dehumidification process on the air side, latent load, in addition to the sensible load. The simulated refrigerant was R-22, and the percentage of discrepancy between the experimental and predicted cooling capacity was ranged between (4.6 to 10.2 %) depending on the operating cooling unit conditions.

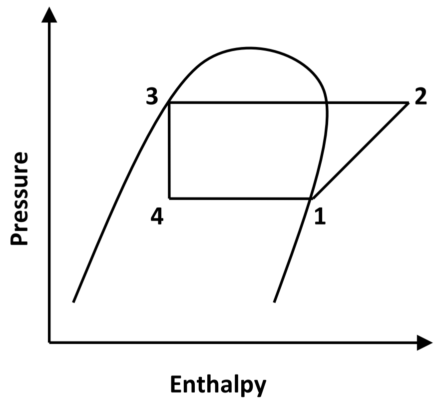
Nanofluids firstly, demonstrated by Choi [7] at Argonne National Laboratory that was defined as suspensions of nanoparticles into base fluid with the typical length scale of particles is 1-100nm. Since 2005 nanorefrigerants have introduced as one kind of nanofluids that can enhance the performance of a refrigeration system. By using nanoparticles in refrigeration system, three main advantages can be obtained; (1) nanoparticle as an additive can increase the solubility between the lubricant and the refrigerant. (2) thermal conductivity and heat transfer characteristics of the refrigerant can be increased. (3) Nanoparticles dispersion into lubricant may decrease the friction coefficient and wear rate. Wang et al. [8], Lee et al. [9] and Das et al. [10] measured the thermal conductivity by addition of nanoparticles containing AL_2O_3 and CuO ,and investigated the effect of the base fluid on thermal conductivity of nanofluids. Xie et al. [11] examined the effect of base fluid on thermal conductivity of AL_2O_3 nanofluid .The target of the present work is to compute the performance of vapor-compression water chiller using two alternative refrigerant (R-407c and R-134a) and to study the effect of using nanofluid (water+ TiO_2) in shell side of water chiller on the performance.



(a) Main component of vapor compression cycle.



(b) T-S diagram of vapor compression cycle.



(c) P-H diagram of vapor compression cycle.

Figure 1: Standard vapor compression cycle of the water chiller.

2. Theoretical formulation

The major evaporator model assumptions are:

1. The water mass flow rate is assumed to be distributed uniformly over the whole length of evaporator coil turn.
2. The water temperature inlet to the evaporator coil turn assumed to be the same for each element of the turn.
3. The average water exit temperature of each turn is considered to be the inlet to the next turn.
4. The enhanced heat transfer due to the swirling motion of water over the helical evaporator coil was neglected.

The measurement is carried out when the water chiller system (test rig) operates at steady state mode. Figure (1) shows a schematic diagram of the vapor- compression water chiller cycle**.**

The evaporator load capacity is estimated for the refrigerant side by the knowledge of the conditions of refrigerant at inlet and exit of the evaporator [12].

$$Q_{evap} = \dot{m}_r \times (h_1 - h_4) \qquad \text{... (1)}$$

Since the refrigerant mass flow rate was not measured during the present experimental work, the energy balance on both sides of the evaporator was used for this purpose.

The condenser heat rejection is estimated from the knowledge of the conditions of refrigerant at inlet and exit sides of the condenser.

$$Q_{cond} = \dot{m}_r \times (h_2 - h_3) \qquad \text{...(2)}$$

The actual compressor work is defined as that work used by refrigerant to change conditions, is estimated by

$$W_{act} = \dot{m}_r \times (h_2 - h_1) \qquad \text{...(3)}$$

Cooling capacity to the actual work of compressor is:

$$COP_{act} = \frac{Q_{evap}}{W_{act}} \qquad \text{...(4)}$$

The equations used to calculate the nanoparticle concentration (%vol.) and the effective density of the nanofluids are [13]:

$$\varphi_v\% = \frac{\frac{m_p}{\rho_p}}{\frac{m_p}{\rho_p} + \frac{m_f}{\rho_f}} \qquad \text{...(5)}$$

$$\rho_{nf} = (1 - \varphi_v)\rho_f + \varphi_v \rho_p \qquad \text{...(6)}$$

The properties of TiO_2 nanoparticles, R-134a and R-407c refrigerants have tabulated in table (1) .Table (2) shows nanoparticle concentration (%vol.) with weight of TiO_2 (g). The thermal conductivity of nanofluid (k_{nf}) can be calculated by [13]:

$$k_{nf} = k_f \frac{\boldsymbol{k_p} + 2\boldsymbol{k_f} - 2\varphi_v(k_f - k_p)}{\boldsymbol{k_p} + 2\boldsymbol{k_f} + \varphi_v(k_f - k_p)} \qquad \text{...(7)}$$

Where k_{nf} ،k_f ،k_p are the thermal conductivity of nanofluid, base fluid and the nanoparticles.

Table (1): The properties of nanofluid , R-134a and R-407c [13] .

	Density (kg/m^3)	*Thermal conductivity (W/m.K)*	*Specific heat (J/kg.K)*	*Viscosity (μPa.s)*
TiO_2(dp=10nm)	4230	8.4	710	
R-134a ($T=8^0C$)	1267	0.0897	1360	260.6
R-407c ($T=8^0C$)	1207.7	0.096	1440.5	190

Tabe(2) :The nanoparticle concentration(%vol.) with weight of TiO_2 .

Nanoparticle volume concentration (φ_v%)	*0.05*	*0.075*	*0.1*
Weight of TiO_2 (g)	85	127.5	170

3. Experimental test rig

Figure (2) shows the experimental test rig. Experimental test rig consists of the basic components required for the refrigeration cycle such as, evaporator (shell and coil), condenser (a finned tube heat exchanger, air cooled which is manufactured by "GB.COIL.INC"), compressor (a reciprocating hermetic compressor has a model "RS43CAE-CAA-201") and expansion device (capillary tube) as shown schematically in figure (٣). The refrigerant side flow arrangement and instrumentation are fixed at selected ports around the rig at inlet and exit sides of the components. The water path through the chiller is shown schematically in figure (4) for which the temperature and flow rate were measured at the entering and leaving sides.

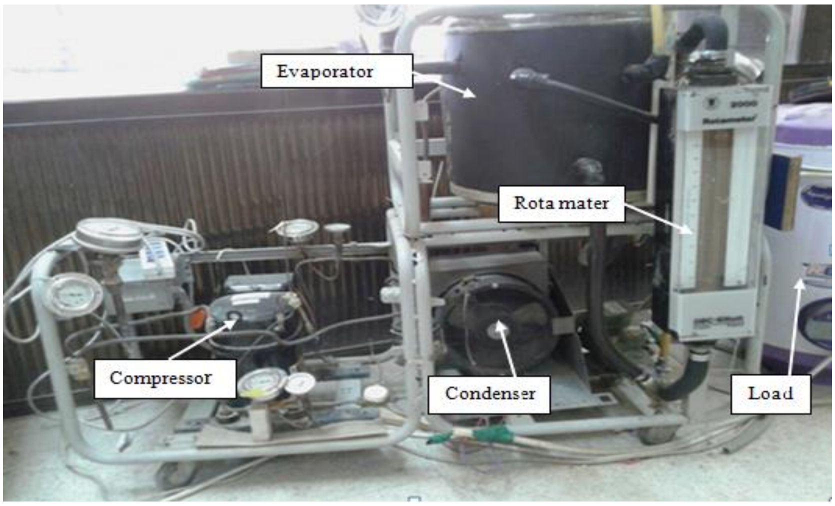


Figure 2: Experimental test rig.

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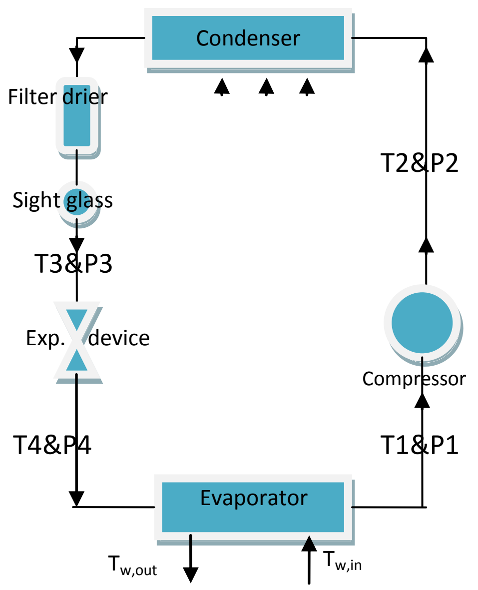


Figure 3: A schematic diagram for the refrigerant side of the chiller.

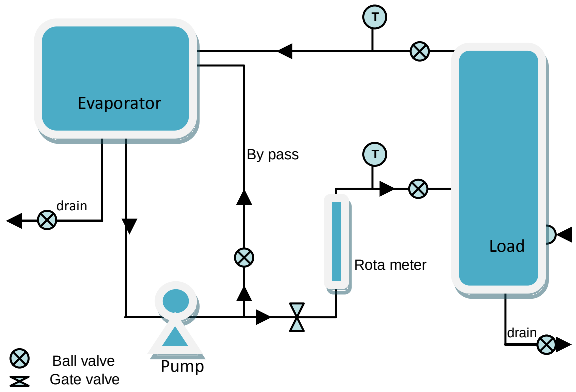


Figure 4: A schematic diagram for the water side path of the test unit.

4. Measuring Instruments

Temperature, pressure and flow rate of water were measured by measuring devices. Temperature measurement of the flowing refrigerant was made by (copper/constantn) thermocouples type's (T) with a temperature range of (-100 to 100°C) at different points of locations, Bourdon gauge was used to measure the pressure of refrigerant at the same locations as the temperature measurement so that the refrigerant thermodynamic properties can be calculated from the measured temperature and pressure at the particular location. Bourdon gauges are classified according to the measuring range to: High pressure gauge (HPG), with range of (0 to 35) kg/cm^2 and low pressure gauge (LPG), with range of (-30 cm Hg to 15 kg/cm^2). These pressure gauges were fixed in the locations and Rota meter 2000 was used to measure the flow rate of the circulating water in (lit/hr). The measurement test is carried out at A/C laboratory in Mech.Eng.Dept. /College of Engineering/AL-Mustansiriya University.

5. Results and Discussion

The present study presents experimental results for the performance of the compression chiller working with R-134a and R-407c in first part. Figure (5) shows the effect of increasing the temperature of the entering water to the evaporator of the chiller system at constant flow rate when using R134a and R407c. Figure shows that COP for the chiller is better with R134a.

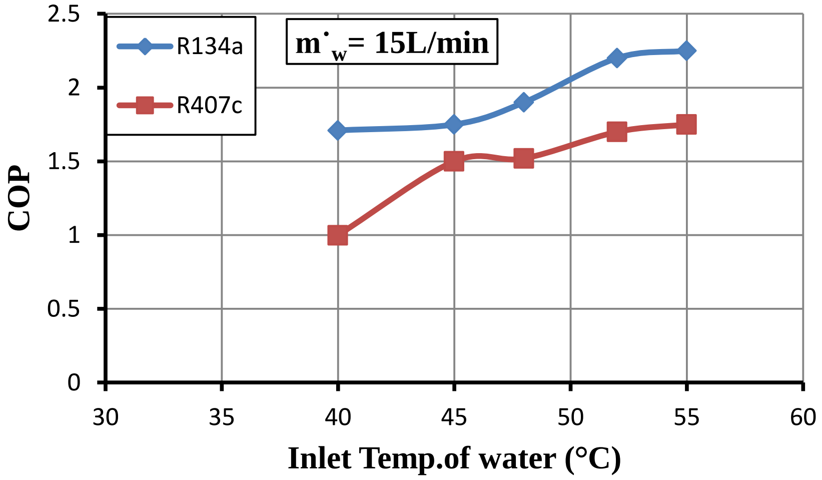


Figure 5: Effect of inlet temperature of water on the COP.

Figure (6) shows the effect of using nanofluid (water+ TiO_2) through the shell side of the evaporator with refrigerant R134a at T_{water} =35°C. The COP increases when using (0.1%) nanofluid by (8.5%) at $\dot{m}$ =30 L/min .This is because of the enhancement in the thermal properties of the water.

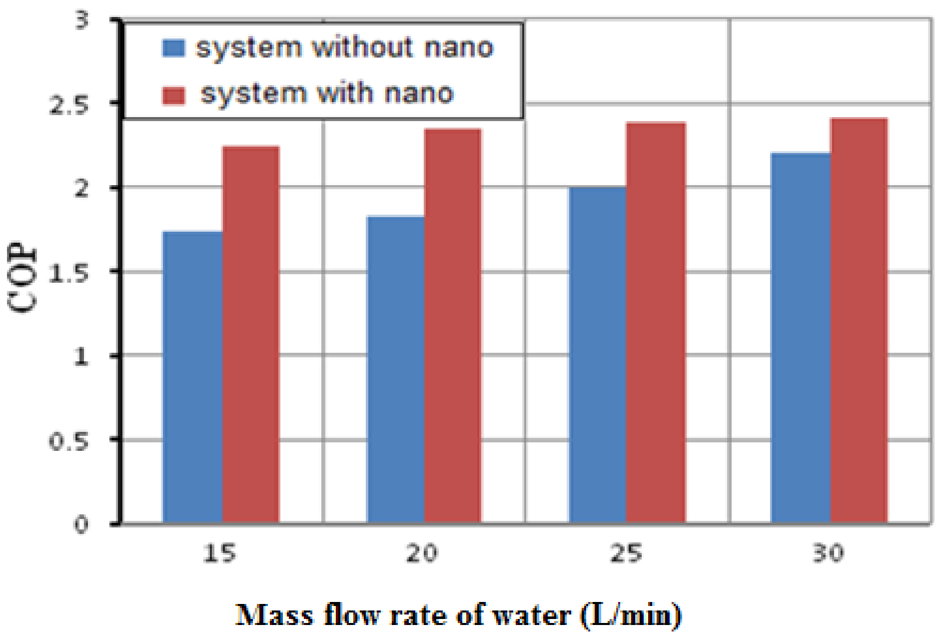


Figure 6: Effect of using nanofluid on the COP (T_{water}=35°C).

Figure (7) shows effect of adding (TiO_2) to chilled water with different volume fraction on the COP of the system. The COP is increases with increasing the concentration of (TiO_2) .This behavior is due to the improvement of the thermodynamic properties of nanofluid. Also COP is increases with increasing of temperature of nanofluid. This behavior is due to heat capacity of nanofluid which is lead to increase the refrigeration effect in evaporator.

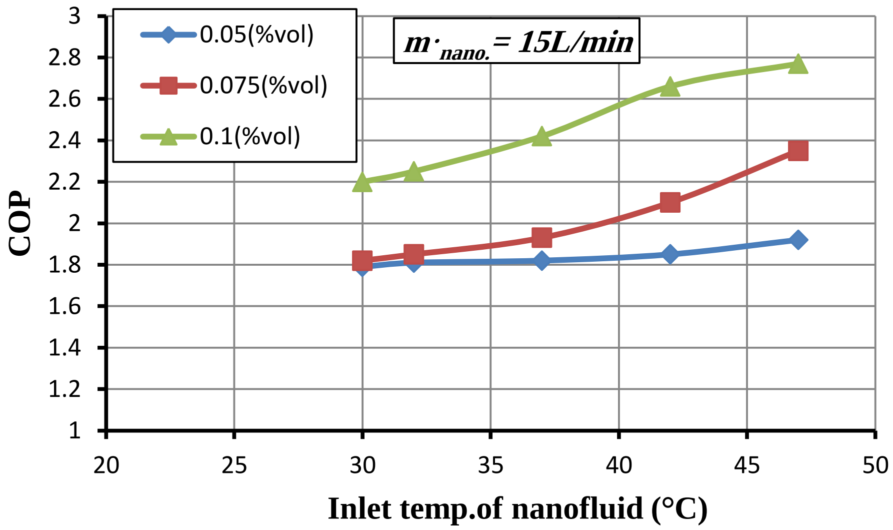


Figure 7: Effect of volume fraction of nanoparticle on the COP ($\dot{m}_{nano}$ =15L/min).

Figures (8 and 9) behave in similar trend, but the COP is increases with increasing of mass flow rate of nanofluid. COP increases because of increasing the heat transfer between refrigerant and nanofluid, which is lead to increases of refrigeration effect.

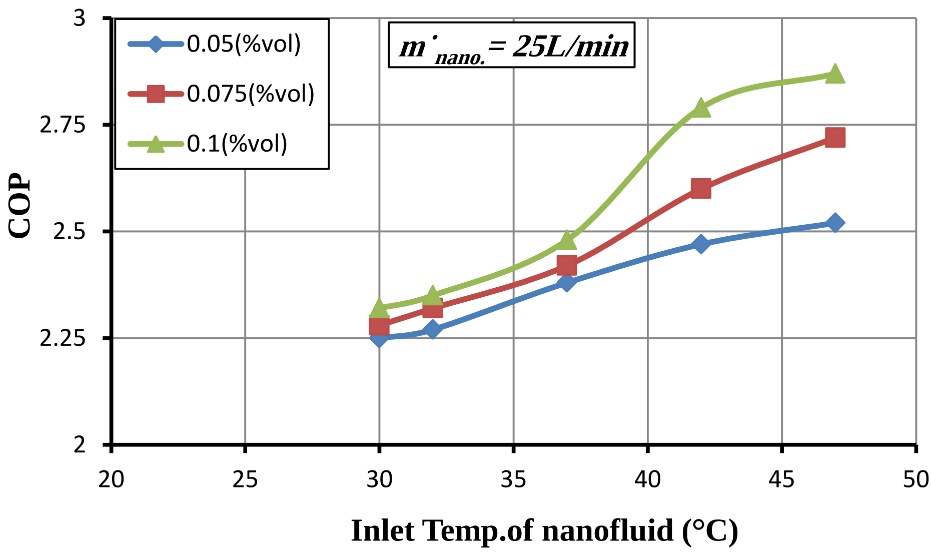
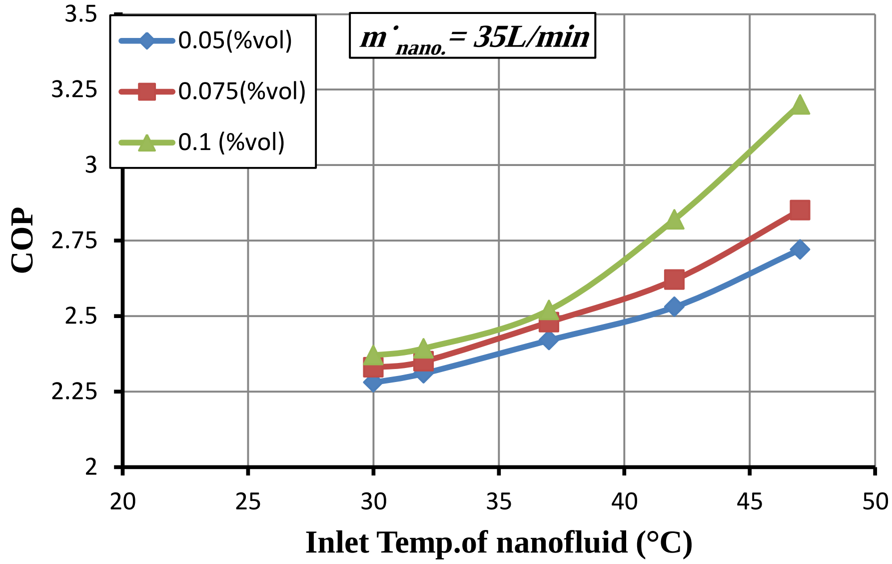


Figure 8: Effect of volume fraction of nanoparticle on the COP ($\dot{m}_{nano}$ =25L/min).



Figures (10 and 11) show that the COP is increases by (25%) when the mass flow rate is increases from (15 to 25) lit/min, when the volume fraction of nanoparticle equal 0.075(%vol) at nanofluid temperature equal 30°C. There is no increment in COP when mass flow rate of nanofluid increases than (25 lit/min).

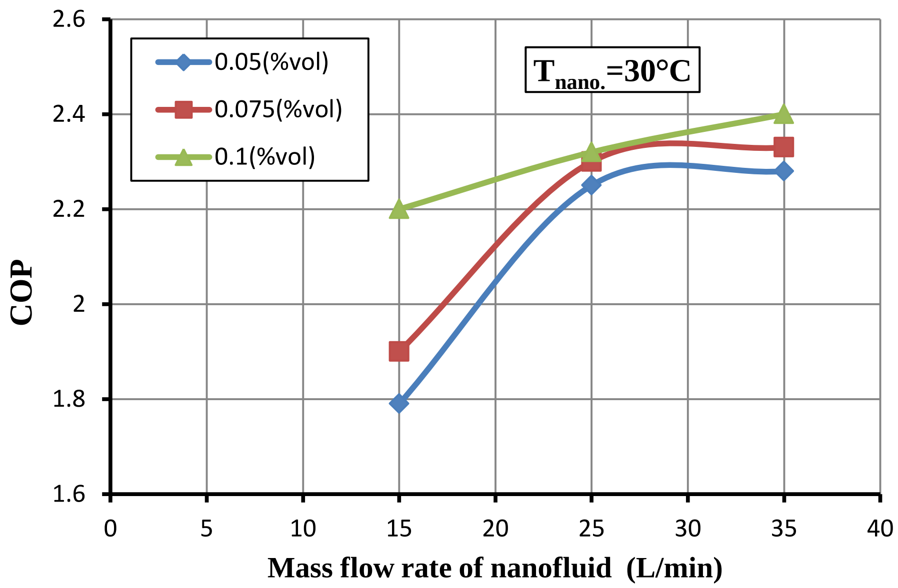


Figure 10: Effect of volume fraction of nanoparticle on the COP ($T_{nano.}$=30°C).

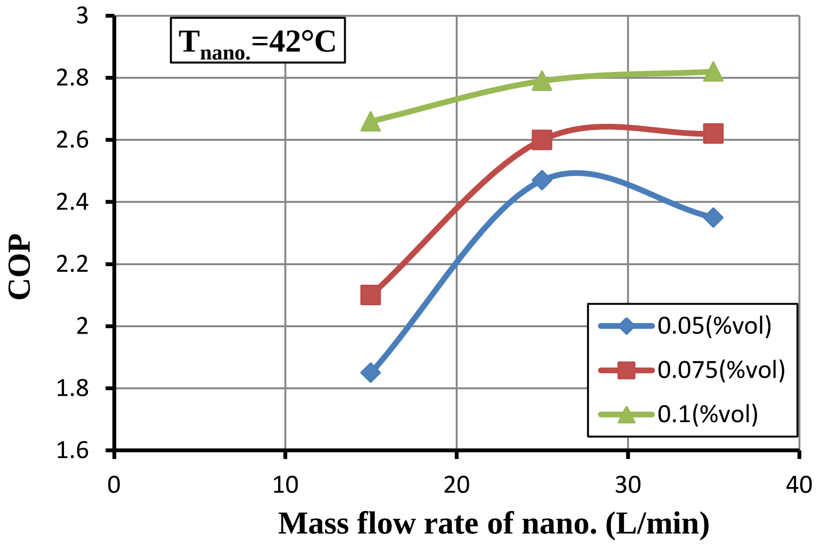


Figure 11: Effect of volume fraction of nanoparticle on the COP ($T_{nano.}$=42°C).

6. Conclusions

The following conclusions can be drawn from the present study:

1. The COP of refrigeration system with R134a is more than COP of the same system with R407c by about (28%).
2. The COP has been found to increase with increasing (%) volume fraction of nanoparticle (TiO_2).
3. The COP increases with increasing of temperature of chilled water.
4. The COP increases by about (25%) when the mass flow rate increase by (66%) at $T_{nano.}$=30°C for nanoparticle volume fraction of 0.075(%vol) .

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