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Research Article

An enhancement of double pipe heat exchanger performance at a constant wall temperature using a nanofluid of iron oxide and refrigerant vapor

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ABSTRACT

This study reports on experimentally enhancing the performance of a concentric double pipe heat exchanger using nanofluid and refrigerant vapor under constant wall temperature conditions. Ferro-nanoparticles with diameters of 80 nm are distributed in distilled water with volume concentrations of 0.1-0.7 % (nanofluid), which is used as hot fluid flowing turbulently inside the inner tube with Reynolds numbers ranging from 3900 to 11800, while refrigerant vapor produced from the refrigeration unit is used as cold fluid with counterflow through the annular tube. The results show that the convection heat transfer coefficient and Nusselt number in the inner tube increase proportionally with a rise in the mass flow rate of nanofluid and the ratio of nanoparticles in the fluid (concentration). Under Reynolds number 11900, the maximum enhancement for convection heat transfer coefficient and Nusselt number in the inner tube was 13.4% and 10.7%, respectively, when using the iron oxide nanofluid with volume concentration of 0.7% compared to pure water. The results of the test were also compared with an almost similar study that used water in the annular tube, and it was found that the use of refrigerant vapor in the annular tube gives better performance compared to water.

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INTRODUCTION

The working principle of heat exchangers is the transfer or conversion of thermal energy. Heat exchangers have many types that are widely used for industrial and domestic purposes, including air conditioning [1, 2], refrigeration [3, 4], power generation [5, 6] and food processing [7]. The importance of heat exchangers in many applications led to an increase in researchers' interests by enhancing its efficiency (the increased heat transfer rate). In general, There are two primary approaches to enhance heat exchanger efficiency: the passive and active methods [8]. In the passive technique, no external force is required for enhancement of heat exchanger performance, including additives for fluids to increase heat transfer [9, 10], or an increase in heat transfer surface, such as a surface shape change and also

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the addition of fins, blocks, ribs, twisted tapes, and coiled wires [11-15], to create turbulence in the fluid flow, which increases the boost mixing of fluid and thermal boundary layer, and therefore increases heat transfer rate [16]. In the active method, an external power source is required to improve heat exchanger performance like stirring the fluid, and vibrating the surface [17, 18].

Nanoparticles are additives for fluids to form nanofluids. Because nanofluid has superior thermal properties that improve the performance of thermal applications [19, 20], it is commonly used in heat exchangers instead of traditional fluids like water. It is worth noting that the enhancement ratio varies according to the type of application, the type and diameter of the nanoparticles (metal, metal oxides, carbon components, etc.), the concentration of nanoparticles in the base fluid, and the stability of nanofluids [9].

The double-pipe heat exchanger transfers The transfer of thermal energy between two fluids of varying temperatures. It consists of one or more pipes positioned concentrically within a larger pipe with fittings to guide the flow. While one fluid flows through the inner pipe (tube side), another fluid flows across the annular gap (annular tube) [21]. Many researchers have tended to enhance the performance of this heat exchanger, whether by passive or active methods, or both [22-26]. Kumar et al. [27] presented an experimental study to estimate the efficiency of a two-pipe heat exchanger with a return bend for the inner tube, and they used different volume concentrations (0.005%, 0.01%, 0.03%, and 0.06%) of the Fe_3O_4-water nanofluid. The hot water and nanofluid flow through the inner tube with a Reynolds number ranging from 15000 to 30000, and an annular tube transports cold water. by the counter flow. The results showed an increase in Nusselt number reached 14.7% and an increase in effectiveness reached 2.4% for volume concentration of 0.06% compared to water at a Reynolds number of equal 30000. Baba et al. [28] presented an experimental study of the formed Fe_3O_4-water nanofluid, inner tube cold working fluid flows using nanoparticles with an average diameter of 75 nm mixed with water at varying amounts (0.02, 0.04, 0.06, 0.08, and 0.1%). The findings indicated that the convection heat transfer coefficient and heat transfer rate increased with increasing the enhancement of the heat transfer rate is about 70-80% with the use of the finned tube instead of the smooth tube with a volume concentration of 0.1%. Sundar et al. [29] presented an experimental study to evaluate the performance of the In a double-pipe U-bend heat exchanger, varied pitch ratios of wire coil with core-rod (WCCR) inserts are used. and the use of water and Fe_3O_4-water nanofluid with varying volume concentrations (0.005%, 0.01%, 0.03%, and 0.06%). The hot fluid flows through the inner tube at a Reynolds number that ranges from 16000 to 29000, in addition, the cold water runs via an annular tube by the counter flow. The results showed an increase in Nusselt number reached 14.7% for a volume concentration of 0.06% compared to water when Reynolds number was 28954. Also, the results showed that with the use of a coil of wire with a core-rod insert of a certain pitch ratio = 1, an increase in Nusselt number reached 37.9% for a volume concentration of 0.06% compared to water when Reynolds number was 28954. Dhiaa et al. [30] presented an experimental study to improve the performance of a vertical double-pipe heat exchanger with counter flow using MgO-water nanofluid at a concentration of 0.1%. The nanofluid passes into the inner tube as a hot fluid, while the water passes through the annular tube as a cold fluid. The results showed an improvement in the performance, as the Nusselt number increased to 17% when using the nanofluid compared to distilled water. Alsaffawi et al. [31] presented a numerical study using ANSYS FLUENT R-14.5) to find out the effect of Al_2O_3- The effects of water nanofluid and CuO-water nanofluid on the performance of a two-pipe heat exchanger in turbulent flow regulation, and they used different volume concentrations (1%, 3%, and 5%) of both nanofluids. The results showed an increase in Nusselt number and effectiveness when using nanofluids instead of water and as follows (5% > 3% > 1% > 0% of volume concentrations). The results also showed that the effectiveness decreases with the increase in the Reynolds number and that Cu-water nanofluid has a Nusselt number and effectiveness greater than Al_2O_3-water nanofluid. Jalili et al. [32] presented an experimental study to evaluate the performance of a countercurrent flow double pipe heat exchanger in turbulent flow with the smooth inner tube and various inner tubes, and they used water, Al_2O_3-water nanofluid, and TiO_2-water nanofluid with different volume concentrations (0.4%, 2%, 4%, and 6%) water serves as the cold fluid in the inner tube and as the heated fluid in the annular tube. The results showed that the convection heat transfer coefficient is as follows: (Al_2O_3-water nanofluid > TiO_2-water nanofluid > water) and increasing the volume concentration to 6% leads to an increase in the convection heat transfer coefficient by 12%. The studies also show that heat exchangers with a rectangular or curved fin are 81% and 85% more efficient than those without a fin. Kassim et al [33] presented an experimental and numerical study to investigate the performance of a two-pipe heat exchanger, and they used varying volume concentrations (0.1%, 1%, and 3% of the SiO_2-water nanofluid with a nanoparticle size of (15–20) nm. The hot nanofluid flows with a high Reynolds number through the inner tube. The number ranges from 3019.43 to 4824.22 in addition, the cold water runs via an annular tube. by the counter flow. The results showed an increase in Nusselt number reached 15.72% and an increase in performance factor reached 11.57% when volume concentration was 3% and Reynolds number was 4824. Poongavanam et al. [34] presented an experimental study of analyzing the heat transfer, Nusselt number, and pressure drop characteristics in a double pipe heat exchanger, in counterflow with the smooth and shot peened inner tube, and they used a

Multi-walled carbon nanotube nanofluid (MWCNT-Solar glycol) at various volume concentrations (0.2%, 0.4%, and 0.6%). The findings revealed that the shot was successful peened tube increased the performance of the heat exchanger compared to the smooth tube. The heat transfer rate and Nusselt number increased when using MWCNT-Solar glycol nanofluid compared to the basic fluid, and the biggest increase was with a volume concentration of 0.6%. Singh and Sarkar [35] conducted an experiment to determine the effectiveness of the double-pipe heat exchanger. when inserting a tapered wire coil and using an (Al_2O_3+MgO/water) hybrid nanofluid flowing in the inner tube under turbulent conditions. The results showed that the hybrid nanofluid and tapered wire coil combination improved the double pipe heat exchanger hydrothermal characteristics.

Hamza and Aljabair [36] presented an experimental and numerical study to investigate the enhancement of a horizontal double-pipe heat exchanger performance using hybrid nanofluid (CuO+Al_2O_3/water) at volume concentrations of 0.6, 1.22, and 1.8%. The nanofluid passes in the inner tube as a hot fluid with a Reynolds number range of 3560-8320 and uniform heat flux (13217.5 W/m^2) while the water passes through the annular tube as a cold fluid, and a twisted tape (twist ratios = 9.2) into the inner tube was inserted. Results show a high increase in the Nusselt number at a concentration of 1.8% and twisted tape, as well as an enhancement in the heat transfer of about 6.70% to water.

After conducting a literature survey, it was found that all nanofluids improve the performance of the double-pipe heat exchanger. The improvement quality varies according to the size of the nanoparticles, the type of nanofluid, the method of its preparation, the operating conditions, the size and design of the heat exchanger, as well as the presence or absence of additions such as fins and twist tubes, etc. Most previous studies also relied on water acting as an auxiliary fluid in the other tube, exchanging heat with the nanofluid according to a constant heat flux as a boundary condition. Constant heat flux generates a magnetic field that negatively affects the performance of the thermocouples, resulting in unstable readings.

This study describes how to enhance the performance of a double pipe heat exchanger by using an Fe_3O_4-water nanofluid flowing through the inner tube and refrigerant vapor flowing through the annular tube. The refrigerant vapor is used for the first time in this field, and the aim of its use is to improve the performance and to achieve a constant wall temperature as a boundary condition.

EXPERIMENTAL SETUP

The experimental test of nanofluid heat transfer properties was implemented in the experimental apparatus as shown in Figures 1 and 2. It chiefly consists of a refrigeration unit, a test section (double pipe heat exchanger), a pump to circulate the fluid from a receiving tank to the test section tube, thermocouples to read temperatures, a turbine flowmeter to observe the nanofluid rate of flow, and the rest of the constituents. The operating fluid rate of flow will be controlled by employing a bypass branch with a control valve once the pump discharges. The test section is 0.45 m long, with hot pure water and nanofluid applied inside the inner tube of a horizontal double-pipe

Table 1. Uncertainty and operation range of the measuring devices

The device	Operation range	Uncertainty
Thermostat	0-100°C	± 1%
Thermocouples	0-180°C	± 1%
Flowmeter	0-30 L/min	± 0.08%

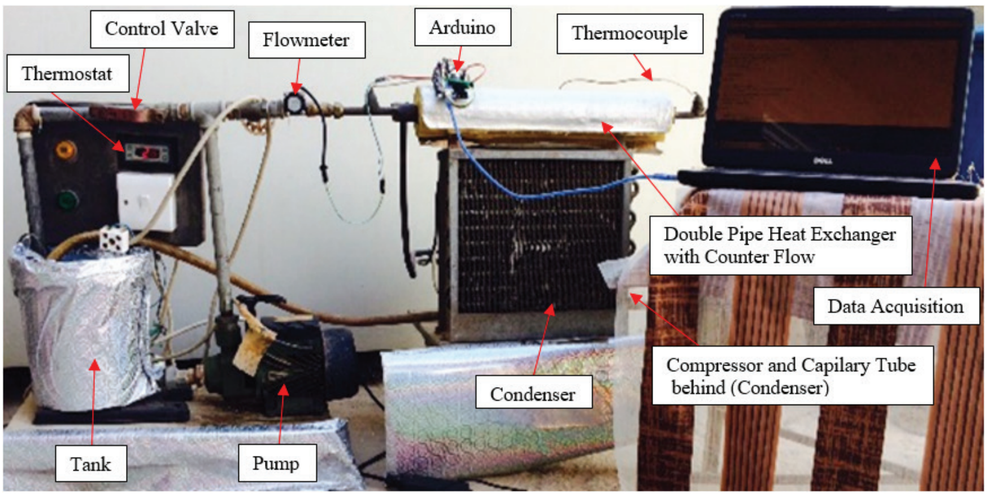

Figure 1. The test rig.

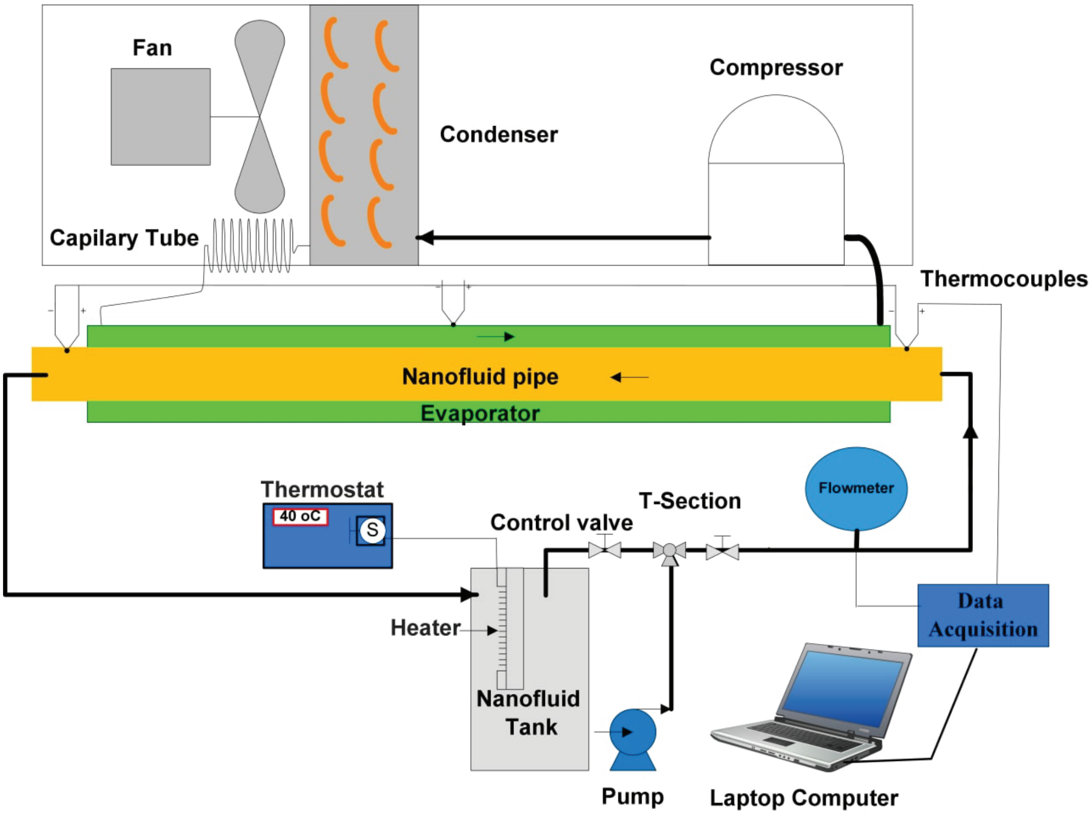

Figure 2. Diagram of the test rig.

heat exchanger with a counterflow path; the annular tube contains the evaporator of the refrigeration unit acting on R134a with a capability of 0.88 kw. The inner and outer tubes are made of copper and measure 12.5 mm and 28 mm in diameter, respectively, to measure the inlet and outlet temperatures of the nanofluid, two K-type thermocouples were used, and a third thermocouple was also used to measure the temperature of the refrigerant inside the evaporator. Grooves have been made on the surface of the tube to fix the thermocouples. To maintain the temperature of the nanofluid, an electrical heater with a thermostat consisting of two thermocouples was installed within the nanofluid tank. The test measurement uncertainty is also recorded, as shown in Table 1.

PREPARATION OF FE_3O_4 NANOFLUID

The iron oxide nanoparticles (Fe_3O_4) used in the experiment have 99.0% purity with a median particle size of 80 nm (see Table 2). The base operating fluid is pure water, and the nanoparticles are magnetic Fe_3O_4. By dispersing Fe_3O_4 nanoparticles in water, Fe_3O_4-water nanofluids were produced. In a bulk amount of five liters, completely different particle concentrations were ready. It is necessary to achieve homogeneous nanoparticle dispersion within the base fluid (distilled water). The expression in Table 3 can be used to calculate the density, specific heat, viscosity, thermal physical phenomenon, and particle variety of the Fe_3O_4 particles and nanofluid. The nanofluid properties are often predicted by the correlations listed in Table 3. Using a mixer, the nanoparticles and water are combined immediately and stirred for 15-30 minutes before every experiment. Nanofluid samples are ready for various concentrations by dispersing pre-weighed quantities of nanoparticles in water. The concentrations utilised in the experiments are (φ = 0.1, 0.3, 0.5 and 0.7 available volume). The volume concentration (φ) is evaluated from the subsequent relation in percentage [37]:

$$\varphi = \frac{\text{vol. of nanopartical}}{\text{vol. of nanopartical} + \text{vol. of water}} \text{ x } 100 \qquad (1)$$

Table 2. Material properties

Substance	Fe_3O_4	Water
Mean diameter (nm)	80	-
Density (Kg/m^3)	5180	997
Thermal conductivity (W/m.K)	80.4	0.607
Specific heat (J/kg.K)	670	4180
Viscosity (N.s/m^2)	-	$8.91\text{x}10^{-4}$

Table 3. The correlations used to calculate nanofluid Property

Property	Correlation
ρ_{nf}	$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_{np}$ [38]
C_{nf}	$C_{nf} = (1-\varphi)C_{bf} + \varphi C_{np}$ [39]
μ_{nf}	$\mu_{nf} = (\frac{\mu_{bf}}{(1+\varphi)^{2.5}})$ [40]
K_{nf}	$K_{nf} = K_{bf}\,[\frac{2+K_{pf}+2\varphi(K_{pf}-1)}{2+K_{pf}-\varphi(K_{pf}-1)}]$
	where: $K_{pf} = \frac{K_{np}}{Kb_f}$ [41]

where: ρ_{nf}, ρ_{bf} and ρ_{np}: density of nanofluid, base fliud and nanoparticles (Kg/m^3), C_{nf}, C_{bf} and C_{np} specific heat of nanofluid, base fliud and nanoparticles (J/kg.°C), μ_{nf} and μ_{bf} viscosity of nanofluid and base fliud (N.s/m^2), K_{nf}, K_{bf} and K_{np} thermal conductivity of nanofluid, base fliud and nanoparticles (W/m.°C).

EXPERIMENTAL CALCULATIONS

Using the experimental data, the convective heat transfer coefficient and Nusselt number of nanofluids were calculated with varied concentrations of particle volume. The heat transfer rate of the hot fluid (water and nanofluid Fe_3O_4) flowing in the inner tube will be expressed as [28]:

$$Q_h = \dot{m}_h C_h (T_{in)h} - T_{out)h}) \tag{2}$$

where: Q_h: hot fluid heat transfer rate (W), $\dot{m}_h$: rate of mass flow of the heated fluid (kg/s), C_h: specific heat of a warm fluid, $T_{in)h}$ & $T_{out)h}$: inlet and outlet temperatures of the hot fluid (°C).

While heat transmission of the refrigerant R134a (cold fluid) and the mass flow rate for the outer tube are correlated, they are not identical are [42]:

$$Q_c = \dot{m}_c (F_1 - F_4) \tag{3}$$

$$\dot{m}_c = \frac{IV cos\emptyset}{\eta_{is}\eta_m (F_2 - F_1)} \tag{4}$$

where: Q_c: heat transfer rate of the cold fluid (W), $\dot{m}_c$: mass flow rate of the cold fluid (kg/s), F_1, F_2 & F_4: refrigerant vapor enthalpies (cold fluid) (J/kg), it can be found by Figure 3. I: compressor current (A), V: compressor voltage (V), cos∅: power factor, η_{is} & η_m: mechanical and isotropic efficiency.

The cold and hot sides are used to forecast the average heat transfer rate (Q_{av}) employed in the computation as shown [28]:

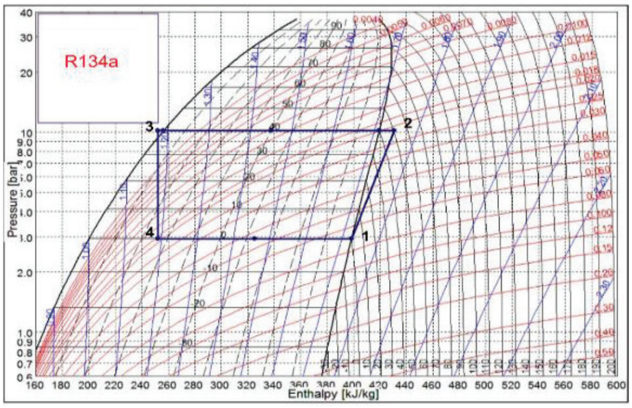

Figure 3. Ideal refrigeration cycle of cold tube [43].

$$Q_{av} = \frac{(Q_c + Q_h)}{2} \tag{5}$$

Overall heat transfer coefficient for hot fluid flows in a concentric heat exchanger tube, the calculation is based on [44]:

$$U = \frac{Q_{av}}{A_i \Delta T_{LM}} \tag{6}$$

where: U: overall heat transfer coefficient (W/m^2.°C), A_i: inner tube area (m^2), ΔT_{LM}: logarithmic mean temperature difference (°C), it can be determined by Eq. (9) [45]:

$$\Delta T_{LM} = \frac{\Delta T_a - \Delta T_b}{ln\,(\Delta T_a/\Delta T_b)} \tag{7}$$

where: $\Delta T_a = T_{out)h} - T_c$, $\Delta T_b = T_{in)h} - T_c$ and T_c: temperature of the cold fluid (°C).

The outer convective heat transfer coefficient is calculated from Eq. (8) [46]:

$$h_o = \frac{Nu_c\, k_c}{D_{hyd}} \tag{8}$$

$$k_c = \Delta x \frac{F_{fg}}{L} \tag{9}$$

$$Nu_c = 0.0082({Re_c}^2 k_c)^{0.5} \tag{10}$$

$$Re_c = \frac{\rho_c v_c\, D_{hyd}}{\mu_c} \tag{11}$$

where: h_0: outer convective heat transfer coefficient (W/m^2.°C), Nu_c: Nusselt number of the cold fluid, k_c: thermal conductivity of the cold fluid (W/m.°C), D_{hyd}: hydraulic diameter (m), which represents the difference between the outer and inner diameter. $\Delta x \cong 0.75$: difference in dryness fraction, F_{fg}: refrigerant vapor enthalpy at evaporater pressure (J/kg), L: tube length (m), Re_c: Reynolds number of the cold fluid, v_c: velocity flow of refrigerant vapor (m/s), ρ_c: density of the cold fluid (Kg/m^3), μ_c: viscosity of the cold fluid (N.s/m^2).

The inner convective heat transfer coefficient, can be found from the following Eq. (12) [47]:

$$U = \frac{1}{\frac{1}{h_i} + \frac{D_i ln\,(D_o/D_i)}{2k} + \frac{D_i}{D_o}\frac{1}{h_o}} \tag{12}$$

where: h_i: inner convective heat transfer coefficient (W/m^2.°C), k: thermal conductivity of the tube wall (W/m.°C), D_i&D_0: inner and outer diameter of tube (m).

Now Nusselt number (Nu_h) and Reynolds number (Re_h) of the hot fluid can be calculated using Eq. (13)&(14) [48]:

$$Nu_h = \frac{h_i * D_i}{k_h} \tag{13}$$

$$Re_h = \frac{\rho_h v_h D_i}{\mu_h} \tag{14}$$

where: k_h: thermal conductivity of the hot fluid (W/m.°C), v_h: velocity flow of the hot fliud (m/s), ρ_h: density of the hot fluid (Kg/m^3), μ_h: viscosity of the hot fluid (N.s/m^2).

The losses of heat are seeming to be very low and the difference between Q_h and Q_c not more than 10% in the heat exchanger, which indicates that the experimental work is well adjusted and is completely ready to start to proceed with the experiments.

RESULTS AND DISCUSSION

The results were obtained at laboratory conditions of 35°C, the humidity and wind speed are moderate. Water and nanofluid were tested at four volume concentrations (0.1, 0.3, 0.5, and 0.7 %) with mass flow rates ranging from (0.0332 kg/s) to (0.0997 kg/s). The results were found based on readings taken after the test rig reached a thermally stable state. The effect of thermophoretic forces on nanoparticles was not taken into account.

Figure 4. depicts the variation of the heat transfer coefficient in an inner tube with the Reynolds variety for various concentrations of nanofluids as compared to that of base fluid (pure water). It will be determined that the heat transfer coefficient of nanofluid will increase with an increase in particle concentrations and Reynolds range. The improvement in the heat transfer coefficient for nanofluid is caused by the Brownian movement of the nanoparticles. The largest percentage improvement in the heat transfer coefficient reached almost 13.4% at a Reynolds number of 11900 and a particle concentration of 0.7%.

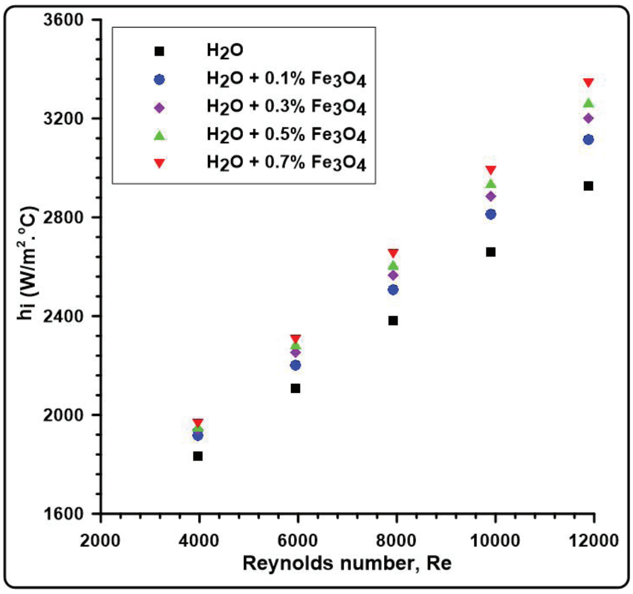

Figure 4. Heat transfer coefficient of the hot fluid at different concentrations and Reynolds number.

The inner tube Nusselt number of the water and the various volume concentrations of nanofluid with Reynolds variety are shown in Figure 5. It's clear that the identical trend of heat transfer coefficient will be found in this figure, in spite of the fact that the most significant part of Nusselt's number is that the convection heat transfer addition to tube diameter and thermal conductivity that is additionally increased in a nanofluid. Underneath the identical Reynolds number, compared to water, the Nusselt number of nanofluid is rather high. This is possible with nanoparticle dispersion. At

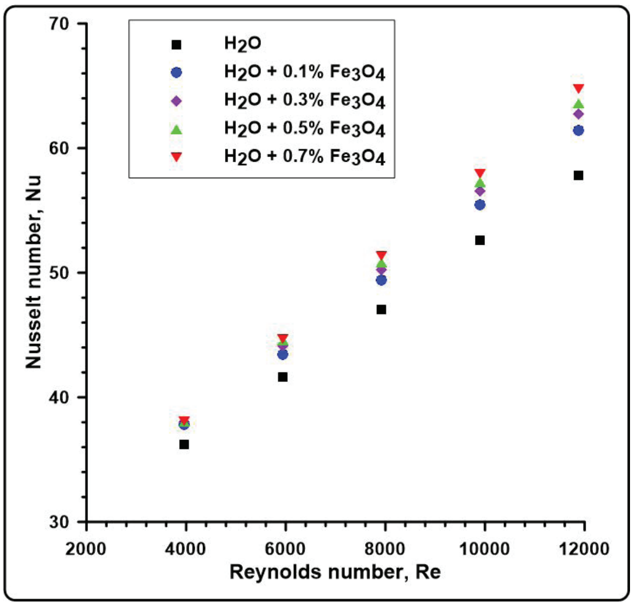

Figure 5. Nusselt number of the hot fluid at different concentrations and Reynolds number.

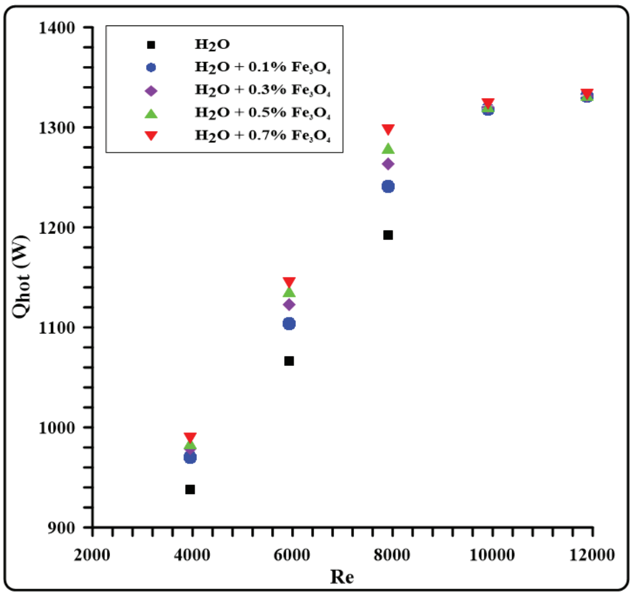

Figure 6. Heat transfer rate of the hot fluid at different concentrations and Reynolds number.

a particle concentration of 0.1%, the improvement in Nusselt number at a Reynolds number of 11,800 is worrisome at 5.8% compared to pure water. Similarly, when the concentration of particles is 0.3%, the improvement in the Nusselt number at Reynolds number of 11,800 is around 7.8% compared to the water data, but when the particle concentration is 0.7 percent and the Reynolds range is 11,800, the Nusselt number increases to 10.7% at the same Reynolds number compared to the pure water result.

Figure 6 shows the heat transfer rate of the water and the various volume concentrations of nanofluid with the Reynolds variety. The heat transfer rate increases as the Reynolds number increases due to the increase in fluid mass flow rate. Increasing the particle concentration also leads to an increase in the heat transfer rate compared to pure water due to the increase in the specific heat. According to the study's findings, the heat exchanger's overall heat transfer area may be smaller since nanofluid improves heat transfer there.

To employ nanofluids in practical applications, it is required to analyse their flow characteristics in addition to their heat transfer performance. In general, nanofluids need a higher pumping force than their base fluid. An increase in the volumetric concentration of nanoparticles in the working fluid results in an increase in the fluid's density relative to pure water, which increases energy consumption. Additionally, when the working conditions improve, the mass flow of the refrigerant required for the heat exchanger's cooling decreases, resulting in a decrease in pumping consumption. It can be may disregard this extra power by restricting the rise in volume concentration such that the advantage of improved heat transfer is higher than the power needed for pumping. Figures (7, 8) show the heat transfer coefficient and Nusselt number variety with respect to volume concentration for various Reynolds numbers. Rates of enhancement can be seen in the two parameters (h_i and Nu_h) with an increased volume concentration in all ranges of Reynolds numbers, This increase decreases with the increase in concentration, due to the increase in the viscosity of the nanofluid, so the proportions of nanoparticles in base fluid must be controlled because excessive proportions lead to negative results.

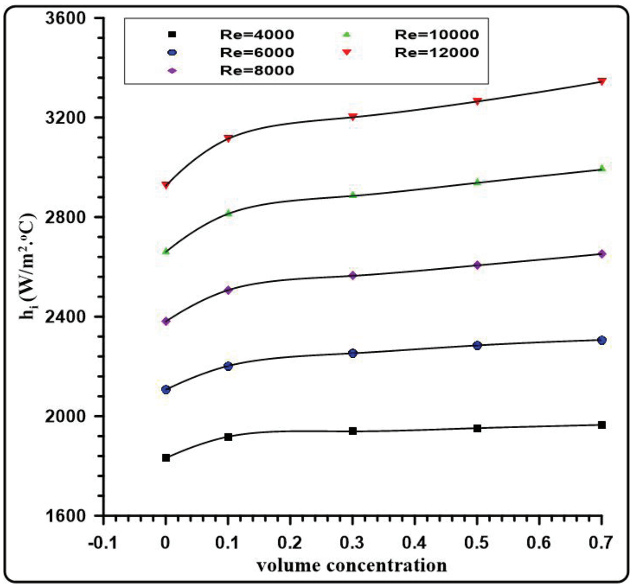

Figure 7. The effect of volume concentration on heat transfer coefficient in inner tube for different Reynolds.

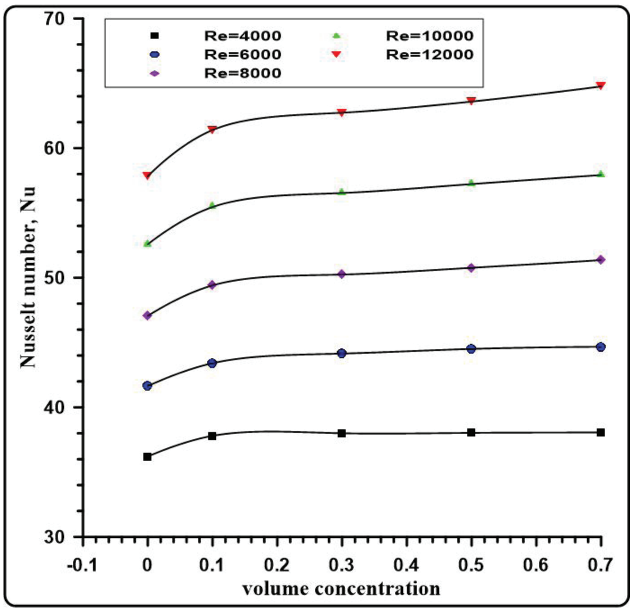

Figure 8. The effect of volume concentration on Nusselt number in inner tube for different Reynolds.

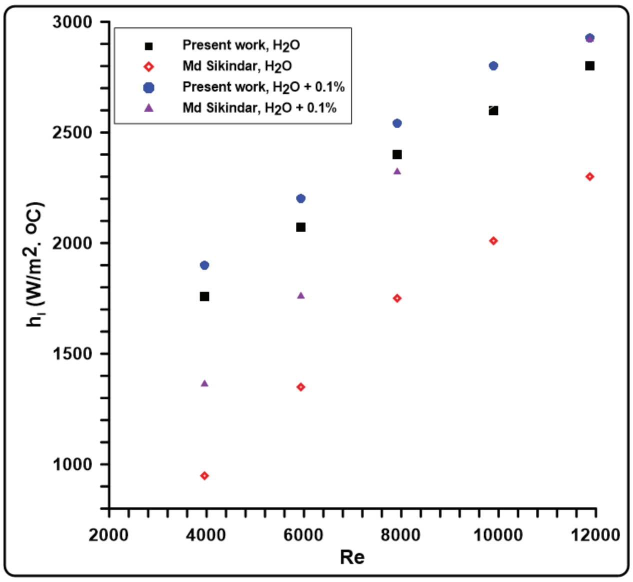

Figure 9. Heat transfer coefficient of the hot fluid is compared with Md Sikindar Baba et.al. [22].

Figure (9) supplies a good comparison of the heat transfer coefficient in the inner tube between this work and Md Sikindar Baba et. al. [22]. This comparison proves two facts: that the heat transfer coefficient increases with the increase in Reynolds number, and that the use of nanofluid within the calculated concentrations instead of water leads to an increase in the heat transfer coefficient.

In [22], water was used as a cold fluid in the annular tube, so it can be compared with this study that used refrigerant vapor in the annular tube. Despite the difference in the test section between the two studies, the comparison gives an acceptable perception due to the similarity of the Reynolds number, the type of nanoparticles, and their concentration in water. Figure 9 shows that the heat transfer coefficient values in the inner tube are higher in the current study, whether it is water or nanofluid, and this is due to the low and almost constant refrigerant vapor temperature (5°C) which enhances the heat exchange process. Figure 9 shows also that the difference between the heat transfer coefficient of the inner tube for the water is larger than the nanofluid due to the mean diameter of the nanoparticles used in the [22] equals 75 nm, which is smaller than it was in the current study and which leads to better thermal properties for the same concentration [49].

CONCLUSIONS

The main target of this work is to enhance the performance of double-pipe heat exchangers under a constant wall temperature achieved by a refrigeration unit. A counterflow is applied in the heat exchanger, where the nanofluid passes through the inner tube and the refrigerant vapour passes through the annular tube. The heat transfer enhancement due to the utilization of nanofluid in turbulent flow is an important phenomenon. Moreover, the use of refrigerant vapor can further enhance performance. It concluded that the heat transfer enhancement is affected by the type of nanoparticle, its size, and its concentration in the base fluid, as well as the ratio between thermal conductivity and viscosity of the nanofluid, an important parameter in the enhancement process. The mass flow rate of the fluid also influences heat transfer enhancement. The study showed that the biggest enhancement of the heat transfer coefficient and Nusselt number in the inner tube was 13.4% and 10.7%, respectively, when using the nanofluid with a volume concentration of 0.7% compared to pure water under Reynolds number 11900. In addition, the use of refrigerant vapor instead of water in the annular tube enhances performance more, according to a comparison with a similar study that used water in the annular tube.

NOMENCLATURE

A	tube area (m^2)
C	specific heat (J/kg.°C)
D	diameter of tube (m)
F	refrigerant vapor enthaly (J/kg)
I	compressor current (A)
L	tube length (m)
Q	heat transfer rate (W)
T	temperature (°C)
U	overall heat transfer coefficient ($W/m^2.°C$)
V	compressor voltage (V)
h	convection heat transfer coefficient ($W/m^2.°C$)
k	thermal conductivity (W/m.°C)
$\dot{m}$	mass flow rate (kg/s)
v	velocity flow (m/s)
D_{hvd}	hydraulic diameter (m)
Nu	Nusselt number (-)
Re	Reynolds number (-)
ΔT_{LM}	logarithmic mean temperature difference (°C)
Δx	dryness fraction (-)

Greek symbols

cosθ=	power factor (-)
μ	viscosity ($N.s/m^2$)
ρ	density (Kg/m^3)
φ	volume concentration (-)
η	efficiency (-)

Subscripts

c	cold
h	hot
i	inner
o	outer
bf	base fliud
fg	wet refrigerant vapor
in	inlet
out	outlet

nf nanofluid
np nanoparticles

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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