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Experimental investigation of convective heat transfer properties of synthetic fluid

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Abstract

Heat transfer fluids are important component in transferring heat through heat exchangers in variety of industrial applications including solar energy. Measurement of convective heat transfer coefficients in experimental setup simulating as much actual operating conditions as possible is one reliable method. Experimenting with fully synthetic heat transfer oil meant for use in concentrated solar power plants, the paper presents experimental data for the oil run in a closed-loop indoor test setup up to high temperatures of 200 °C and at two flow rates of 900 and 1200 kg h^{-1}. Convective heat transfer coefficients were calculated based on actual steady-state heat transfer taking place between the hot oil and cold water flowing in a counterflow shell and tube heat exchanger. It was observed that the convective heat transfer coefficient is higher at lower oil flow rate and there is more variation in the experimental values at lower flow rates of oil. On the contrary, the coefficients of convective heat transfer on the basis of empirical correlations at same two oil flow rates were calculated to be higher at higher oil flow rate with the variation uniformly patterned. With respect to calculations based on empirical correlations and experimentally observed values, a comparison of convective heat transfer coefficient "hi" for oil at the two flow rates, the empirically calculated heat transfer coefficients show an increasing trend with a definite gradient, while the experimental values show variable trend which is increasing initially with temperature, then drops slightly and then again starts to increase. In view of the fact that the empirical correlations do not take into account the nature and chemistry of the oil, it has been concluded that the experimental determination of heat transfer coefficient is reliable and feasible, though it may not necessarily correlate with the theoretically derived values.

Keywords Thermofluids · Heat transport properties · Solar thermal energy · Concentrated solar power (CSP) plants · Heat transfer coefficient

Introduction

In the present day, science and technology are mandating using heat transfer oils for high-temperature applications, such as concentrated solar power plants. Devising and designing suitable oil formulations for such severe operating conditions shall require a precise and accurate test method for laboratory evaluation of heat transport properties. If the oils are to be used at elevated temperatures, like those in the case of concentrated solar power generation where temperatures go as high as 400 °C, there is an urgent need for a laboratory test method for determination of heat transport phenomenon of heat transfer fluids (HTFs) which can be utilized in formulating new generation oils based on novel chemistries and technologies like nanofluids, etc. [1].

Heat transport properties of oil are important in determining the amount of heat transferred from the oil as well as in arriving at an optimum plant design in terms of selection of right type and configuration of heat exchangers, pipelines, pumps, etc. Density, viscosity, specific heat capacity and thermal conductivity are generally regarded as the important heat transport properties requiring consideration for a given configuration of heat transfer mechanism.

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For few specialized industrial applications such as CSP generating electricity using sun's heat stored and transported through a heat transfer fluid—*generally oil of suitable chemical nature*, heat transport properties are very critical in arriving at the least cost of electricity generated. There are a variety of standard as well as nonstandard test methods being employed by researchers in evaluation of heat transport properties of oils. Most of these test methods evaluate the oil properties at low to medium temperatures and under atmospheric or lower pressures ranges [2].

The HTF used in solar CSP applications encounters system pressures of 15–30 bar and temperatures as high as 250 °C or above. Under such severe operating conditions, the oil has to perform satisfactorily and it would be useful if the oil's heat transport properties are evaluated at these operating conditions. Though all the four properties play crucial role in actual heat transfer, properties like density and viscosity are more related to flow of oils, while specific heat capacity and thermal conductivity are two important properties that govern the heat transfer taking place through the oil. Among these, thermal conductivity is the most important heat transport characteristic of oil which though, has been studied and experimented very extensively in the past, still offers controversial and debatable results of evaluation.

Thermal conductivity can be evaluated using steady-state test techniques as well as transient test techniques. Steady-state test techniques require more time and are more complex in experimentation, while transient tests are relatively more prevalent and adoptable. All the test methods for evaluation of thermal conductivity of oil result into widely scattered values of thermal conductivity with poor repeatability of results. This is owing to the fact that measuring the "pure" thermal conductivity of oils, that too at elevated temperatures, is very difficult because the oil tends to go into the convection region following the Brownian motion in its molecules. For industrial applications, determination of thermal conductivity and specific heat capacity of oils has been carried out on the basis of certain empirical equations derived by oil formulators based on their experience in collecting, collating and disseminating technical information needed by practicing engineers [3]. Quite a few researchers have reported evaluation of thermal conductivity of oils by different methods, but there still remains a huge scope of debate and conflicting discussions on the reliability of such testing, *specially for high temperatures*. Determination of convective heat transfer coefficient of oil is an alternative, but comparatively more meaningful way of assessing the heat transport property of oils during their formulation and research. Convective heat transfer coefficients can be regarded as the overall effect of all the heat transport characteristics of oil such as the density, viscosity, specific heat capacity and conductivity or heat diffusivity [4].

High-performance heat transfer oil development and evaluation

Researchers in the field of heat transfer fluids are experimenting with several novel compositions of nanofluids and other high performing chemicals in their efforts to increase the heat transfer capability of fluids. Enhancement of heat transport properties of fluid is required to improve the cycle efficiency of heat transfer oils used in solar energy application in particular. During the course of formulating high-performance heat transfer fluids, while most of them have reportedly used combination of heat transport test techniques as mentioned in this paper, some have resorted to testing the oil in simulated test setups evaluating the convective heat transfer properties of the fluid instead of depending upon the individual heat transport properties of thermal conductivity, heat capacity, etc. [5].

Hoffman and Cohen [6] experimentally studied heat transfer coefficients of molten salt mixtures of $NaNO_2$–$NaNO_3$–KNO_3 under forced circulation in circular tubes with parameters such as Reynolds number between 4800 and 24,000 and Prandtl number of 4.2–9.1 and concluded that heat transfer with tested salt mixture could be represented by general corrections for heat transfer fluids flowing through circular pipes. Wu et al. [7] used a specially designed system of circular tube to study the heat transfer characteristics of molten salt for about 1000 h and validate the known heat transfer equation such as Hausen and Gnielinski. Based upon the measurements of flow rate and temperatures, the heat transfer coefficients were determined by least square method and transient flow heat transfer correlations were obtained with reasonable agreement. Garg et al. [8] evaluated four different samples with different duration of ultrasonication in nanofluids prepared using 1% mass of multi-walled carbon nanotubes (MWCNT) and 0.25% mass of gum arabic (GA) in deionized (DI) water. They measured the viscosity through a rotational type viscometer and thermal conductivity using a thermal property analyzer having a probe of 60 mm length and 1.3 mm diameter which had heating element, thermoresistor and a microprocessor to control and measure the conduction in the probe. They also used a specially fabricated convective heat transfer coefficient measurement test setup comprising of copper heat transfer section (length 914.4 mm, inner diameter 1.55 mm and outer diameter 3.175 mm), data acquisition system, DC power supply, a syringe metering pump and a computer. They reported maximum of 20% increase in thermal conductivity at temperatures greater than 24 °C and 32% increase in convective heat transfer along axial distance.

Sohal et al. [9] in a project work at Idaho National Laboratory in USA proposed a conceptual design of a forced convection test loop to verify the convective heat transfer, thermophysical and thermochemical properties, corrosion properties and any other thermal–hydraulic characteristics that are essential in developing high-temperature solar salts. Mohammed et al. [10] reviewed several aspects of microchannel heat exchangers such as channel geometry, fluid inlet and outlet arrangement, type of construction with respect to past works from the literatures. They also reviewed heat transfer and fluid flow properties in such microheat exchangers using nanofluids as well as conventional fluids and emphasized the need for a coordinated approach among researchers so as to develop an accurate, reliable and standardized test technique for study of heat transfer effects of newer generation HTFs. Sarkar [11] reviewed the correlations used in heat transfer and pressure drop for laminar and turbulent flows of nanofluids under natural as well as forced conditions. He concluded that for the nanofluids which have spherical nanoparticles, there is an appreciable agreement of the pressure drop values between experimental and predicted results based on established correlations under laminar as well as turbulent flows. But the established correlations do not match with experimental values in case of heat transfer coefficients of nanofluids and different correlations were proposed for Nusselt number pertaining to laminar and turbulent flow. The author emphasized concerted studies of nanofluids with regard to development of heat transfer correlations and attributed the observed big differences in Nusselt numbers with the proposed correlations for laminar and turbulent flows owing to variety of factors such as interdependence of particle properties and fluid composition on flow and heat transfer characteristics, erroneous data of viscosity and thermal conductivity, absence of substantiated knowledge of flow mechanism of nanofluids and most importantly inadequate experimental methodology and data of nanofluids. Tumuluri et al. [12] experimented on three heat transfer fluids consisting of multiwalled carbon nanotubes (MWCNT), slurry of microencapsulated octadecane-based phase change materials (MPCMs) and their blends, evaluating thermal conductivity using transient hot wire (THW) apparatus, viscosity using spindle type viscometer and heat transfer performance through a specially designed loop consisting of pipings made of copper, heat exchanger, variable voltage transformers, water chiller, pump, motor, flow meter, data acquisition units, thermocouples, pressure transducers, etc. A maximum thermal conductivity increase of 8.1% was observed for MWCNTs having diameter of 60–100 nm and length 0.5–40 microns and showed 20–25% convective heat transfer enhancement in turbulent flow conditions. While MPCM slurry showed good agreement of heat transfer rates with respect to the published literature, the blend showed lower heat transfer rates and pressure drops owing to having higher viscosity.

You et al. [13] developed a mathematical fluid flow and heat transfer model in a direct steam generation (DSG) system using a parabolic trough solar power plant. They actually constructed a pilot test setup using a concentrating trough and evacuated absorber tube for studying the authenticity of the developed mathematical models. They concluded that the models developed by them and verified using the experimentation are in agreement with each other within the permissible engineering ranges and can be used for analyzing the operational properties of the DSG systems for industrial applications. Gonzalez et al. [14] also resorted to designing a special purpose pilot plant primarily to study the degradation of HTFs at high temperatures and used six varieties of HTFs in the plant to understand their degradation majorly with respect to the oils' viscosity change before and after the evaluation exercise. On the basis of the experimental values, the authors also estimated the heat capacity and overall heat transfer coefficients and concluded that both these heat transport parameters follow opposite trends. Lu et al. [15] utilized an experimental facility consisting of oil tank, pump, heat exchanger, data management system and an HTF flow loop to study the convective heat transfer of a molten salt made of $NaNO_2$–KNO_3–$NaNO_3$ chemistry used as HTF in solar energy application. During the experiments, HTF temperature ranged between 250 and 400 °C and Reynolds number between 4000 and 10,000. The authors reported good agreement with experimental data, but the deviations were nonuniform probability under different fluid temperatures. Chen et al. [16] experimentally studied the comparative performance of a commercial grade molten salt Hitec in transversally corrugated and smooth tube. Based on experimental results, the authors observed about 17% correlation between experimental and empirical data. The authors also obtained higher drag coefficient for transversally corrugated tube than the smooth tube on the basis of variation in drag coefficient and Reynolds number. Lu et al. [17] utilized an electrically heated spirally grooved tube to study the heat transfer performance of molten salts. Based on the obtained Nusselt numbers, it was concluded that heat transfer through spiral tube was higher than the smooth tube and that the groove height is helpful in increasing the heat transfer. The authors also found reasonable correlation between the experimental results and modified correlation coefficients based on empirical correlations by Sieder and Tate and Gnielinski.

Ahmad et al. [18] studied over/Cu nanofluid using double lid-driven cavity by lattice Boltzmann method at constant Richardson no, Grashof no, temperature phase deviation and Prandtl no. The analysis showed that the flow pattern of thermal phase deviation change with high Richardson numbers is apparent, while Nusselt number and heat transfer coefficient enhance by decreasing Richardson number. It also seems that average Nusselt number is higher at constant properties than variable. Haghighi et al. [19] measured the thermal conductivity and viscosity of 9% by mass solid nanoparticles of alumina (Al_2O_3), zirconia (ZrO_2) and titania (TiO_2) in water-based nanofluids (NFs) at 20 °C using the transient plane source (TPS-Hot Disk) technique for thermal conductivity and a coaxial cylinder viscometer to measure the viscosity. Their measurement showed agreement with distilled water results at 20 °C to an extent of 2% in thermal conductivity and 4% for viscosity. They also measured the heat transfer coefficients for the same NFs using a straight tube having 1.5 m length and 3.7 mm diameter and observed contradictory results with heat transfer coefficients increasing in a range of 8–51% when using equal Reynolds number and decreasing by 17–63% when compared at equal pumping power. They concluded that when measuring heat transfer coefficients, results should be compared using equal pumping power and not at equal Reynolds number as NFs have higher viscosity than the base fluids and would require higher volumetric flow at equal Reynolds number. Biencinto et al. [20] used simulation methods to comparatively study use of pressurized nitrogen in place of conventional synthetic oil based HTF in parabolic trough plants using the coordinates of an existing Spanish 50MWe parabolic trough plants with 6 h of thermal storage and observed that almost similar net annual electricity productions can happen by replacing the conventional synthetic HTF with pressurized nitrogen which will also be environmentally safe.

Muñoz-Anton et al. [21] studied theoretically the substitution of gas as working fluid in a parabolic trough solar power plant for overcoming flammability and environmental problems associated with conventional synthetic oils. They also described a test loop developed at solar research center of Plataforma Solar de Almería (PSA, Spain) for evaluating the effects and technical feasibility of such new concepts in heat transfer oils. They concluded that the high gas pressure can offset pumping power to better acceptable levels, but also reported absence of technique to detect the gas leakages from ball joints as a major drawback of using gas in place of oil as HTF. Nikkam et al. [22] studied the heat transfer characteristics of nanofluid having α-SiC particle concentration of 3, 6 and 9 wt% and different base fluids using distilled water and distilled water/ethylene glycol mixture. Their thermal conductivity with a Hot Disk apparatus working on the transient plane source method and viscosity using viscometer were evaluated at 20 °C. They found that the loading of 9% by mass of α-SiC particle in distilled water/ethylene glycol base liquid exhibited the best combination of thermophysical properties, which was then taken up for heat transfer coefficient evaluation using a closed-loop system consisting of straight tube of 1.5 m length and 3.7 mm diameter. HTC measurements showed enhancement of 13% under equal Reynolds number, 5.5% at equal pumping power and 8.5% at equal flow rates. Selvakumar [23] et al. used mineral-based commercial heat transfer oil Therminol D-12 to study efficiency of a evacuated tube built in a parabolic trough-based solar collector heating water and observed 30% improvement in efficiency under low solar radiations. Suganthi et al. [24] studied heat transport properties of ZnO–ethylene glycol and ZnO–ethylene glycol–water nanofluid coolants using transient heat transfer test setup consisting of stainless steel sample holder, electrical heating coil and temperature sensor connected with data logger, developing empirical models to calculate the heat transport properties of the fluids. The experiments showed increase in thermal conductivity to the tune of 39 and 17% in case of ZnO–EG and ZnO–EG–water nanofluids over their respective base fluids. Harris [25] reported a modified transient plane source method to measure the thermal conductivity which is able to reduce the convection currents in oil by employing a very short time interval of 0.8 s of measurement, using extremely less sample quantity of 1.25 mL and utilizing a very low-energy 2600 W m^{-2} heat flux to the test specimen holding the oil.

Derakhshan et al. [26] studied the convective heat transfer properties of a heat transfer oil fortified with spherical shaped multi-walled carbon nanotubes (MWCNTs) by developing laminar mixed convection in a dedicated experimental setup utilizing microfin as well as plain tubes in studying the heat transfer characteristics under laminar mixed convection conditions. They proposed two new correlations to predict the mean Nusselt number in horizontal and vertical microfin tubes within an error band of − 6% to + 4% and − 10% to + 8%, respectively. Mohammad et al. [27] proposed a new correlation for assessing the thermal conductivity of COOH-functionalized MWCNTs/water nanofluids. They utilized COOH-functionalized MWCNTs nanoparticles in water as base fluid and measured the thermal conductivity in various MWCNT solid concentrations, up to 1%, but at low temperatures of 25–55 °C only. They then applied artificial neural network analysis using temperature and solid volume fraction as input and thermal conductivity as output and observed good agreement between the ANN and experimental value. Mehdi et al. [28] experimentally evaluated the hydrothermal properties such as particle concentration, baffles overlapping and helix angle of Al_2O_3 nanoparticles in water using shell and tube heat exchanger having helical baffles. Using artificial neural network, they carried out modeling of heat transfer and pressure drop of the nanofluid and predicted optimal cases for various situations.

Masoud et al. [29] studied the natural convection of an electrically conducting fluid under magnetic field in an inclined cylindrical annulus using simulation. In the study, the authors varied the inclination angle from 0° to 90° and Hartmann number up to 60 and found that magnetic field has good impact in controlling the convection of the electrically conducting fluid. They concluded that the convection motion can be suppressed by varying the direction and intensity of the magnetic field. Mohammad et al. [30] investigated the thermal conductivity and viscosity of FE/water nanofluid in varying concentrations and diameter of the nanoparticles. They obtained an increase in thermal conductivity with concentration and decrease with nanoparticle size. They also obtained increased viscosity with increase in concentration and diameter. Mahdi and Mehdi [31] used shell and tube heat exchanger with helical baffles to study the impact and overlap of helix angle on hydrothermal features. They also developed an artificial neural network to assess heat transfer and pressure drop in the heat exchanger and optimized the developed model to arrive at 38 different cases.

Raei et al. [32] measured the convective heat transfer coefficient in a novel experimental setup using double-tube counterflow heat exchanger using Al_2O_3/water nanofluid in flow rates of 7, 9 and 11 L min^{-1}, but again here the inlet fluid temperatures were low at 45, 55 and 65 °C. The authors reported large increase in heat transfer coefficient up to 23% and friction factor up to 25% at 0.15 vol% concentration of nanoparticles. Hosseinzadeh et al. [33] reported using a test setup consisting of a horizontal circular tube in length of 1000 mm and 7 mm in diameter to study the effect of magnetic field on heat transfer increase and friction factor on Fe_3O_4/water nanofluid experimentally at different Reynolds numbers and magnetic field strengths. They have reported improvement in Nusselt number with increase in Reynolds number and percentage concentration of nanopowders. Mehdi [34] did an exhaustive review of the migration of particles in nanofluids. With the issue of migration of particles, he looked into all the previous work done by several researchers and through different methods used such as Eulerian–Lagrangian, Buongiorno model, molecular dynamics simulation as well as theoretical approaches. Several important issues are highlighted that deserve greater attention. In the review, he has been able to clearly pin point the main bottlenecks and directions for further work on this subject. Marjan et al. [35] experimentally determined the thermal conductivity, specific heat capacity and viscosity and convective heat transfer behavior of nitrogen-doped graphene nanofluids flowing in a double-pipe heat exchanger. Using Matlab software analysis, the authors concluded that heat transfer of the nitrogen-doped graphene nanofluid increases with the increase in Reynolds number as well as with increase in quantity of nanoparticles in the fluid.

Mehdi and Masoud [36] carried out experimental studies on multi-walled carbon nanotubes dispersed in ethylene glycol–water mixture in the ratio of 40:60, temperature range between 25 and 50 °C and solid volume fraction range of 0–1.0%. They observed that with increase in solid volume fraction and temperature, the thermal conductivity ratio also increases, and at the same time, temperature effective on thermal conductivity is more predominant at higher concentrations. The authors also used Maxwell model to compare the experimental data and observed that the experimental data do not correlate with the simulation studies, and hence, they proposed correlations for the same. Mohsen et al. [37] examined the dynamic viscosity of single-wall carbon nanotubes in ethylene glycol in temperature range from 30 to 60 °C and solid volume fractions up to 0.1% and observed that the fluid behaves in Newtonian fashion under all concentrations and up to highest temperatures studied. The measurements also indicated that dynamic viscosity increases with increasing solid volume fraction and decreases with increasing temperature. Hamid and Masoud [38] studied the rheological behavior of COOH-functionalized MWCNTs–SiO_2/EG–water hybrid nanocoolant for cooling systems at temperatures ranging from 27.5 to 50 °C and solid volume fractions ranging from 0.0625 to 2%. Viscosity measurements were performed at the shear rate range of 0.612–122.3 s^{-1} for each nanocoolant sample. Results showed that the base fluid exhibits Newtonian behavior and the nanocoolant samples exhibit a pseudoplastic rheological behavior with a power law index of less than unity ($n < 1$) and also that the apparent viscosity generally increases with an increase in the solid volume fraction and decreases with increasing temperature.

Ebrahim et al. [39] experimented with Al_2O_3-MWCNT nanoadditives (0–1.0% and 25–50 °C) in a SAE40 engine oil formulation to obtain their rheological characteristics and observed that the nanofluid behaves in Newtonian manner. They also observed that with decrease in temperature and increase in nanoadditive concentrations, the viscosity of the fluid increases. They performed sensitivity analysis and found that while viscosity is less sensitive to temperature, it is more sensitive to solid volume fractions and proposed correlations to calculate the viscosity of engine nanofluid based on experimental values. Masoud et al. [40] vouched for higher accuracy of the optimized artificial neural network model compared to experimentally derived correlations for predicting the thermal conductivity of Fe_3O_4 magnetic nanofluids. They observed a deviation of 5% for correlations and 1.5% for artificial neural network-based thermal conductivity values of the magnetic fluid studied experimentally.

Mohammad et al. [41] used an SAE40 viscosity grade engine oil mixed with MWCNTs and SiO_2 in varying volume fractions up to 2% and temperature range from 25 to 50 °C to investigate the effect on rheological performance of the fluid. The authors concluded that the fluid exhibits Newtonian behavior in volume fraction up to 1% and non-Newtonian beyond that. They also observed viscosity decrease with temperatures and increase with concentration. Davood et al. [42] evaluated the viscosity of magnetic Fe_2O_3 nanofluid in a range of temperatures and solid volume fractions and observed that the viscosity reduces appreciably with temperature and increases with the nanoparticle concentrations. The authors proposed a correlation for viscosity of the magnetic nanofluid and obtained good accuracy with the experimental results. Mohammad et al. [43] predicted the thermal conductivity of ethylene glycol–water mixed with alumina nanofluids using artificial neural network as well as on the basis of correlation coefficients derived out of experimental studies. The authors concluded a high correlation between the experimental results and those obtained using artificial neural network. Majid et al. [44] experimented with the water mixed with CuO nanoparticles in several volume fractions up to 2% flowing in turbulence inside a double-tube, counterflow heat exchanger. They observed that the rate of increase in heat transfer coefficient of the fluid is more at lower ranges of Reynolds number and concluded that the effect of increasing percentage of nanoparticle in lower Reynolds number was more. Mohammad et al. [45] studied the dynamic viscosity of engine oil containing alumina nanoparticles in solid volume ratios and temperature up to 65 °C using Brookfield viscometer. They observed that nanofluids follow Newtonian behavior with viscosity increasing with nanoparticle concentration to as high as 132% at highest additive dosage of 2%, decreasing with temperature and derived mathematical correlations which were found not to predict the correct viscosity under experimentation.

Saman and Davood [46] studied the effects of variation in Reynolds number and Darcy number on the Nusselt number and the convection heat transfer coefficient using a water–copper oxide (CuO) nanofluid in a sinusoidal channel with a porous medium and observed increasing trends. They further concluded that temperature gradient along the sinusoidal channel increases with Reynolds number and that the porous portion helps control temperature differential, thus enhancing the convective heat transfer coefficient. Amir et al. [47] investigated the heat transfer coefficient, Nusselt number and pressure loss of water/CuO nanofluid in single phase for laminar study flow having Reynolds number 100 with boundary conditions firstly constant heat flux for all sides and secondly two side constant heat flux and constant temperature one side. They observed that pressure loss increases as CuO concentration increases, while velocity distribution is fully developed. It also shows that the size of nanoparticle does not effect the heat transfer properties. Omid et al. [48] concluded in their studies that the heat transfer increases with an increase in the volume fraction of solid alumina nanoparticles mixed into water-based carboxy methyl cellulose (CMC) solution and reduction in the diameter of the nanoparticles significantly in Reynolds number. Mohammad et al. [49] examined the viscosity of MWCNTs/ZnO-SAE40 hybrid nanolubricants at different temperature and volume fraction ranges. They observed that viscosity decreases with rise of temperature and enhances with the increase in volume fractions. Maximum viscosity seems at 33% of nanolubricants. And nanolubricants act like Newtonian fluid.

Mohammad et al. [50] investigated the optimization of nanofluid aluminum oxide in water and ethylene glycol in the ratio 40:60. They applied NSGA II algorithm in neural network modeling to decrease viscosity and increase thermal conductivity. The observation showed that optimal thermal conductivity and viscosity occur at the highest temperature. Mohammad et al. [51] experimentally studied and proposed correlations for thermal conductivity of COOH-functionalized multi-walled carbon nanotubes in water with varying solid volume fractions up to 1%, temperature range of 25–55 °C and using different dispersion techniques. They also trained their data obtained through an artificial neural network model with good degree of confidence. Roozbeh et al. [52] used ethylene glycol in water at 50% concentration, added functionalized single walled carbon nanotubes (F-SWCNTs) in varying volume concentrations, evaluated the thermal conductivity of the samples at different temperatures and observed that thermal conductivity varied directly with the additive concentrations and temperature. The authors also studied the effect of absence and presence F-SWCNTs in the base fluid by measuring the forced convective heat transfer in a smooth tube and concluded that the nanofluid was efficient as long as relative viscosity is lower than the 0.465 power of the thermal conductivity ratio.

Masoud [53] carried out an exhaustive experimental work on thermal conductivity of magnesium oxide and functionalized multi-walled carbon nanotubes hybrid nanofluid in temperature range of 25–50 °C and nanoparticle concentration range of 0–0.6% and found high increase in thermal conductivity in the order of 21%. He also proposed mathematical correlation for prediction of thermal conductivity based on his experiments. Mehdi et al. [54] studied the heat transfer characteristics of green tea leaves oil containing silver nanoparticles in a miniature counterflow double-tube heat exchanger. They found that at high concentrations and Reynolds numbers, nanoparticle migration have appreciable impact on efficacy of the heat exchanger. Pumping power increases with Reynolds number and decreases with increase in concentration. They concluded that nanofluids in high concentration are advantageous to be used in heat exchangers because there is an increase in the ratio of heat transfer rate to pressure drop with concentration increment. Mehdi et al. [55] investigated in an annuli the convective heat transfer and flow properties of a copper nanoparticle in a non-Newtonian base fluid containing 0.4 mass% carboxymethyl cellulose (CMC) solubilized in water. They observed higher impact on pressure drop with changing concentration of nanoparticles and comparatively lower impact by changing radius ratio and particle size. They also carried out modeling through artificial neural network (ANN) to find out convective heat transfer coefficients in both wall, and the pressure drop in terms of radius ratio, volume concentration, and particle size.

Thus, it can be seen from the above literature review that different researchers have used several different methods for evaluating the convective heat transfer coefficient of oils, and so far no test method has been standardized for the same. There is a need for developing a uniformly acceptable test method that can determine the overall heat transport properties of oil, such as the convective heat transfer coefficients. The paper presents experimental data for the oil run in a closed-loop indoor test setup up to high temperatures of 200 °C and at two flow rates of 900 and 1200 kg h^{-1}, employing fully synthetic heat transfer oil meant for use in concentrated solar power plants. Convective heat transfer coefficients are calculated based on actual steady-state heat transfer taking place between the hot oil and cold water flowing in a counterflow shell and tube heat exchanger.

Experimental

A typical solar heating operation as taking place in actual outdoor condition in a solar thermal plant with two-tank thermal oil storage system is represented in Fig. 1.

In order to study the convective heat transfer characteristics of heat transfer fluid at high operating temperatures normally encountered in solar thermal plants, a test setup was designed and built with the objective of determining the mechanisms by which the heat transfer is enhanced in heat exchangers and related piping in various flow regimes such as laminar and turbulent. The test setup consists of a closed-loop test system of pipe test section, temperature measurement devices, flow meters, heat exchangers (tube and shell co-current and countercurrent type) data acquisition system and software. Figure 2 depicts a schematic diagram of the testing setup used for evaluation of thermic fluid for their intended application in solar thermal systems.

The highly instrumented test setup consists of electrical mechanism for oil heating to high temperatures and heat exchangers having equal heat transfer area to study the effect of variations in exchanger designs. Shell and tube type heat exchangers with co-current and countercurrent fluid movement were made. The setup was designed in such a way that only one heat exchanger was able to operate at a time. It was possible to emulate the solar charging process usually performed by a solar collector [56]. The discharging cycle was obtained by passing the hot thermic fluid through an oil/water heat exchanger. DM water was used as the other fluid stream, and temperatures of inlet water and outlet steam were recorded precisely during experimentation.

Control parameters:

1. Oil heater through electrical heat load and agitation.
2. Temperature of hot oil coming from oil heater—t_{h1}.
3. Temperature of oil leaving the heat exchanger—t_{h2}.
4. Oil flow rate to heat exchanger—m_h.

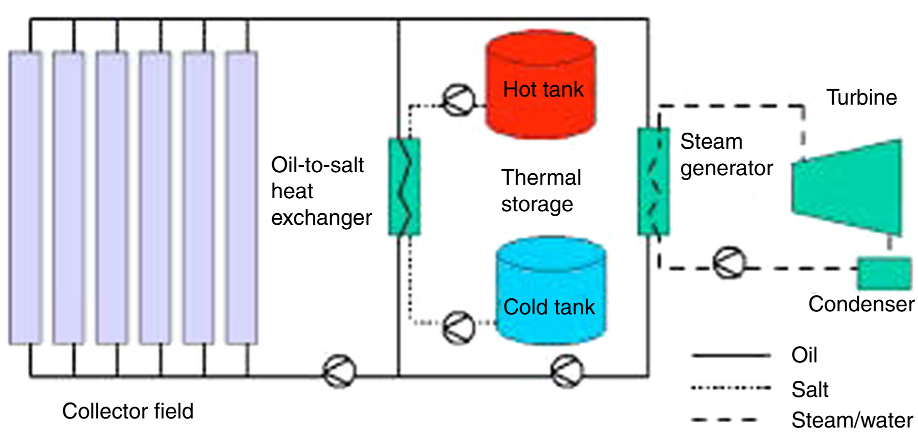

Fig. 1 Typical field with indirect two-tank thermal oil storage system

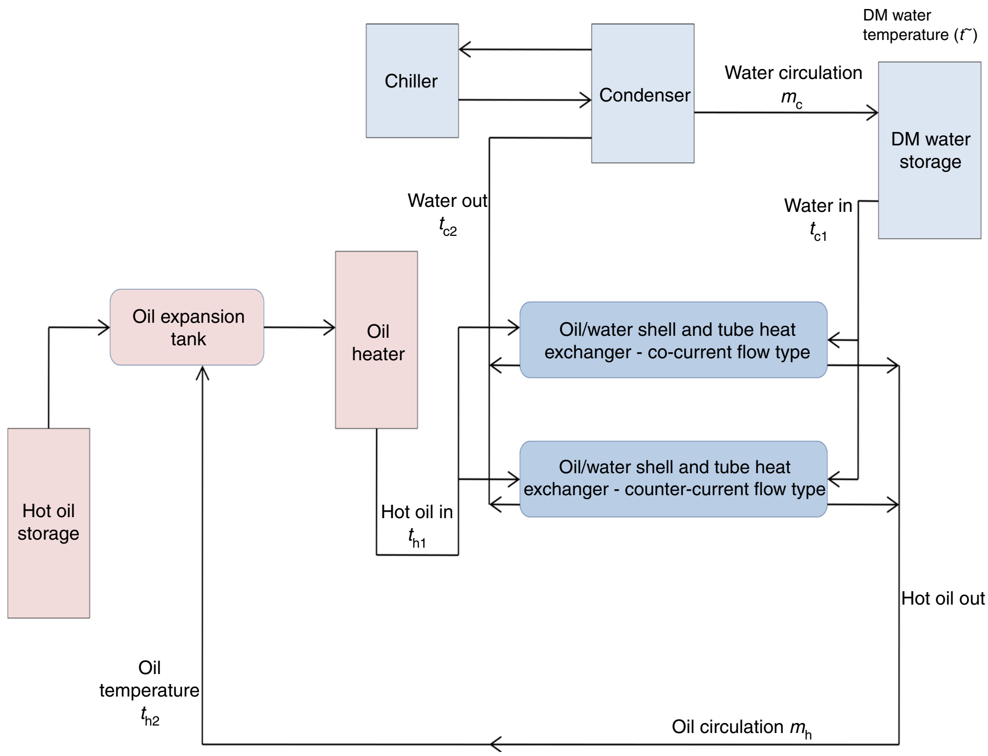

Fig. 2 Schematic diagram of testing setup for evaluation of thermic fluid

5. Water inlet and outlet temperature—t_{c1} and t_{c2}.
6. Water circulation rate—m_c.

Experimental methodology

Tests were performed to validate the test setup capabilities with a fully synthetic grade heat transfer oil as working fluid having kinematic viscosity of 2.48 cst @ 40 °C, flash point of 124 °C and pour point of 12 °C. The following experimental study was performed pertaining to measurement of heat transfer coefficients:

- The test loop was run with the objective of achieving the steady state of heat transfer in the heat exchanger at a predefined flow rate of water, oil and hot oil temperature.
- Flow rates and temperatures at the heat exchangers inlet and outlet for the oil and the water side were then measured.
- Recording of inlet and outlet temperatures of hot fluid and cold water giving delta *T* on both sides, respectively.

Both the heat exchangers had almost identical design with the following parameters:

Inner diameter of shell $D_o = 0.2$ m (200 NB pipe, SCH 20).
Total number of tubes = 18.
Outer diameter of tube $d_o = 0.01905$ m.
Thickness of tube wall $t = 0.00165$ m.
Inner diameter of tube $d_i = 0.01575$ m.
Total tube length $L = 1.25$ m.
Pitch = triangular, 25.4 mm.
Baffle spacing = 175 mm (5 Nos).

The following parameters were obtained from the software-assisted heat transfer test setup:

Temperature of hot oil entering the exchanger (°C) = t_{h1}.
Temperature of oil exiting the exchanger (°C) = t_{h2}.
Oil flow rate to heat exchanger (kg h^{-1}) = m_h.
Density of oil (kg m^{-3}) = ρ_o.
Water inlet temperature (°C) = t_{c1}.
Water outlet temperature (°C) = t_{c2}.
Water circulation rate (kg h^{-1}) = m_c.

Since water was used as the cold stream whose heat transfer properties are already well established and researched, Prandtl number *Pr*, viscosity μ_w, thermal conductivity *K*, specific heat capacity Cp_w and density ρ_w were taken from the reported literature [57].

$$\text{Velocity of water} = \frac{m_c}{(\rho_w \times A_o)} \tag{1}$$

$$\begin{aligned} A_o &= \text{Effective cross sectional area of shell} \\ &= \frac{\pi}{4}D_o^2 - \text{Total cross sectional area of tube} \\ &= \frac{\pi}{4}D_o^2 - 18\frac{\pi}{4}d_o^2 \\ &= 0.0263\,\text{m}^2 \end{aligned}$$

Substituting value of A_o in Eq. (1)

$$\text{Velocity of water in the shell } V_w = \frac{m_c}{(0.0263 \times \rho_w)} \qquad Re = \frac{\rho_w V_w D_o}{\mu_w} \tag{2}$$

Now, heat transfer rate in terms of the mass flow rates and temperatures entering and exiting the heat exchanger can be written as:

$$\text{At oil (hot) side}: q_h = m_h C_{ph} dT_h \tag{3}$$

$$\text{At water (cold) side}: q_c = m_w C_{pw} dT_w \tag{4}$$

where q_h and q_c are the heat transfer rates of hot and cold fluids, respectively; m_h and m_c are the corresponding mass flow rates; and C_{ph} and C_{pc} are the corresponding specific heat capacities of the hot fluid oil and cold fluid water, respectively.

Under steady-state conditions, equating Eqs. (3) and (4):

$$C_{ph} = \frac{(m_w C_{pw} dT_w)}{m_h d_{Th}} \tag{5}$$

Overall heat transfer rate under steady-state conditions

$$q = m_h C_{ph} d_{Th} = UA_i \times \text{LMTD} = \frac{\text{LMTD}}{\text{RT}} \tag{6}$$

$$\text{RT} = \frac{1}{A_i U}$$

$$U = \frac{m_h C_{ph} dT_h}{A_i \times \text{LMTD}}$$

where U = overall heat transfer coefficient, A_i = heat exchanger or heat transfer area $(A_i) = 18 \times \pi d_i L$, R_T = overall resistance in heat exchanger.

$$\text{LMTD} = \text{Log mean temperature difference} = \frac{\text{d}T_h - \text{d}T_w}{\log \frac{\text{d}T_h}{\text{d}T_w}} \tag{7}$$

The overall resistances can be calculated based on hot side fluid resistance (R_{hf}), wall resistance (R_w) and cold side fluid resistance (R_{cf}) using:

$$R_T = R_{hf} + \left(\frac{F_f}{A_i}\right)_h + R_w + \left(\frac{F_f}{A_{osurface}}\right)_W + R_{Wf} \tag{8}$$

Here, F_{fh} and F_{fw} are the fouling factor on hot and cold sides, respectively, whose values have been obtained from the reported literature [58] for oil and water, respectively.

Nusselt number for water side was calculated using Gnielinski's equation for turbulent flow in smooth tubes as per the following:

$$N_u = \frac{\left(\frac{f}{8}\right)(R_e - 1000)Pr}{1.0 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)}\left[1 + \left(\frac{D}{L}\right)^{\frac{2}{3}}\right] \tag{9}$$

The above Gnielinski's equation is applicable for fully developed flow characterized by $D/L \sim 0$. In the study that was performed, $D = 0.01905$ m and $L = 1.25$ m. Hence, condition of $D/L \sim 0$ was fulfilled.

Further, Gnielinski's equation is applicable in the following cases which was completely fulfilled during experimentations in the heat transfer test setup:

- Developing or fully developed turbulent flow.
- Smooth tubes.
- $2300 < Re < 5 \times 10^{6.}$
- $0.5 < Pr < 2000.$
- $0 < D/L < 1.$

f = Darcy friction factor obtained from Blasius correlation applicable for turbulent flow in smooth pipes for Reynolds number ranging between 3×10^3 and 2×10^5

$$f = \frac{0.3164}{R_e^{0.25}} \tag{10}$$

Having known the Nu, h_w, the heat transfer coefficient at the cold water side was calculated using the following equation:

$$Nu = \frac{h_w \times d_o}{K} \tag{11}$$

where K is the thermal conductivity of water (W m^{-1} °K^{-1}).

Substituting the above value of h_W in Eq. (8)

$$R_T = R_{hf} + \left(\frac{F_f}{A_i}\right)_h + R_w + \left(\frac{F_f}{A_{osurface}}\right)_W + R_{Wf}$$

where

$$R_{hf} = \frac{1}{A_i h_i}$$

$$R_w = \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi L K_w}\ (K_w \text{ taken as } 54\,\text{W m}^{-1}\,\text{K}^{-1} \text{ for stainless steel tube metallurgy})$$

$$R_{wf} = \frac{1}{A_{osurface} h_w}$$

Now since in the above equation, only unknown remaining is R_{hf}, by substitution of all other known values, R_{hf}, therefore, h_i was calculated using the following experimental protocol:

- The oil was circulated in both counterflow and co-current heat exchangers of tube and shell type—to experimentally assess the effectiveness of each other.
- Distilled water was taken as the cold fluid receiving heat from oil.
- Flow rate of water was maintained around 1200 kg h^{-1} on the basis of the initial benchmarking studies and based on the observation that best steady-state condition for the experiments was being achieved at this flow rate.
- Oil flow rate was varied from 900 up to 1200 kg h^{-1} in steps of 100 kg h^{-1} within the limitation and capacity of the oil pump that was used in the setup.
- At each of the flow rates of oil, once steady state is arrived, inlet and outlet temperature values for both oil and water were recorded.

Literature-reported correlations

For solar energy applications of concentrated solar power, the heat transfer fluid generally used is a eutectic mixture of two pure synthetic compounds "diphenyl-oxide/diphenyl" and whose heat transport properties like density, viscosity, thermal conductivity and specific heat capacity can be derived using suitable mathematical modeling software by using the known physicochemical properties of the two components of the oil. For one such solar grade heat transfer fluid operating up to temperatures of 400 °C, the suppliers [59] have mentioned the heat transport properties in tabular form based on empirically derived formulae reproduced below:

$$\rho = -0.90797 \times t + 0.0078116 \times t^2 - 2.367 \times 10^{-6} \times t^3 + 1083.25 \quad (12)$$

$$\mu = \mathrm{e}^{\left(\frac{544.189}{t+114.43} - 2.59578\right)} \quad (13)$$

$$C_p = +0.002414t + 5.9591 \times 10^{-6}t^2 - 2.9879 \times 10^{-8}t^3 + 4.4172 \times 10^{-11}t^4 + 1.498 \quad (14)$$

$$k = -8.19477 \times 10^{-5}t - 1.92257 \times 10^{-7}t^2 + 2.5034 \times 10^{-11}t^3 - 7.2974 \times 10^{-15}t^4 + 0.137743 \quad (15)$$

where ρ = density in (kg m^{-3}), μ = kinematic viscosity (mm^2 s^{-1}), C_p = specific heat capacity in (kJ kg^{-1} °K^{-1}), k = thermal conductivity in (W m^{-1} °K^{-1}), t = temperature of oil in °C.

Using the above literature-reported correlation, the calculated values of *Re*, *Pr* and subsequently *Nu* were derived. Once *Nu* was known, convective heat transfer coefficient was also be calculated.

Analysis and comparison of results

Based on both the experimentation and calculations based on simulated empirical relationships, the convective heat transfer coefficient "h_i" was obtained. The following analysis was arrived in the studies.

Co-current v/s countercurrent heat exchanger studies

The purpose of using two different heat exchangers viz. co-current and countercurrent exchangers was to compare the heat transfer ability of the two different types of exchangers under identical experimental conditions. As depicted in Fig. 3, utilizing the experimentally arrived values of convective heat transfer coefficients and overall heat transfer coefficients, regression analysis was carried out to assess the best curve fitting values for the two types of heat exchangers used in the study. It can be seen that the highest regression coefficients were obtained for polynomial curves and also within the polynomial curves, the highest values were obtained in the case of counterflow heat exchanger. Thus, counterflow heat exchanger was taken for all subsequent studies based on the fact that it resulted into highly correlated, significantly uniform and consistent readings of temperature differentials during experimentations. Therefore, all the experiments were then performed using the counterflow heat exchanger and the results being reported in this paper pertain to counterflow heat exchanger only.

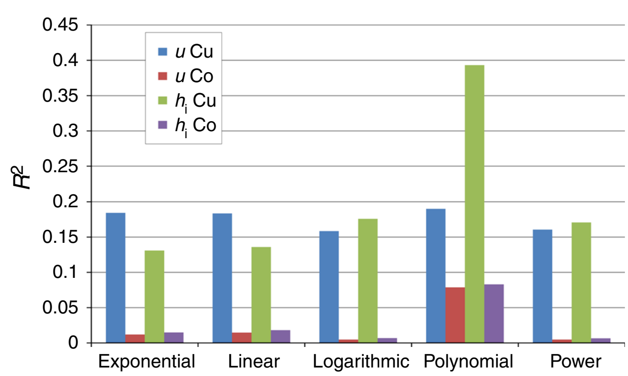

Fig. 3 Regression coefficients for different curve fittings in co-current and countercurrent heat exchanger

Comparison of literature-reported correlation v/s experimental properties

Important oil properties like density and kinematic viscosity of oil were calculated using the empirically simulated relationship as well as observed experimentally during the studies using a high precision, reliable and low maintenance Krohne make mass flow meter model Optimass 6400F suitable for high temperatures and pressures. Figure 4 depicts the calculated and the experimental density values of the oil from where it is seen that the experimental values of the density are lower than the calculated values which will have implication at a later stage when using the experimental values the convective heat transfer coefficients are obtained for subsequent comparison.

Similarly, Fig. 5 depicts the calculated and the experimental values of kinematic viscosity of the oil from where it is seen that the experimental values of the kinematic viscosity are higher than the calculated values having significant implication on the convective heat transfer coefficients that are obtained subsequently.

From Figs. 4 and 5, it is observed that experimental values of density are low and the viscosity value is higher. Generally, the oil having low density should have lower viscosity values also. However, in the present study, the oil used is a fully synthetic one and hence can show this kind of nature in actual use. These two values will have their combined effect on the subsequently derived heat transfer coefficients.

Experiments under different flow rates of oil

Within the limitation of the pump used in the experimental setup, the oil flow rate could be varied between 900 and 1200 kg h^{-1}. On both the lower and higher limits of oil flow rates, the convective heat transfer coefficient "h_i" was calculated experimentally. Figure 6 shows the variation in the convective heat transfer coefficient at 900 and 1200 kg h^{-1} of oil flow rate from where it can be seen that the convective heat transfer coefficient is higher at lower oil flow rate and also that the lower oil flow rate shows more variation in the values of heat transfer coefficient obtained. This observation is contrary to the theory that the heat transfer will be invariably more at higher turbulences. One reason that can be understood is that the heat transfer is not a function of turbulence alone, but it is also function of residence time for which the oil is in contact with the surface of heat exchanger. Since, at lower flow rates, the oil gets more residence time of contact with the surface, the heat transfer rates are higher at lower flow rates. Further,

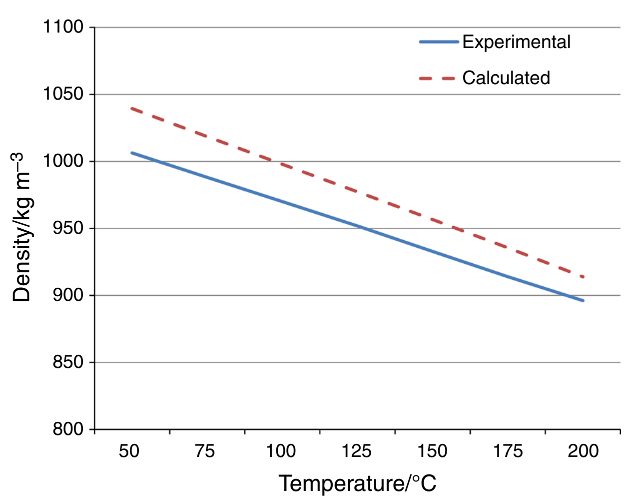

Fig. 4 Calculated v/s experimental density of oil

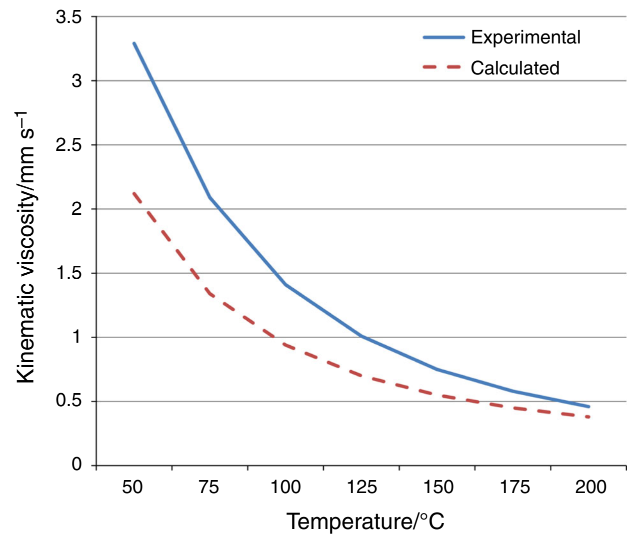

Fig. 5 Calculated v/s experimental kinematic viscosity of oil

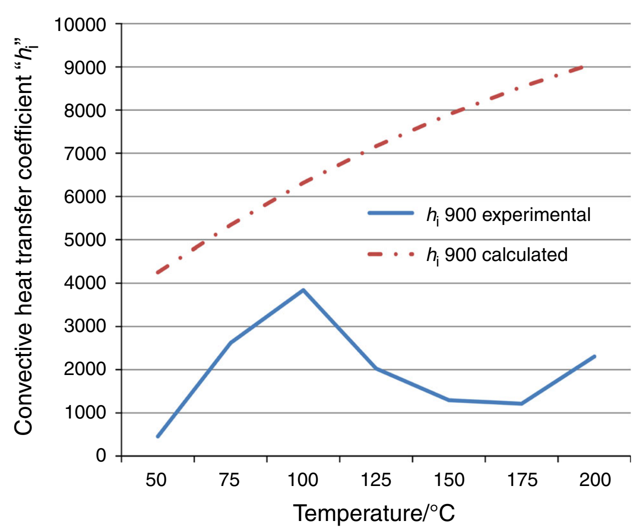

Fig. 6 Variation in experimental convective heat transfer coefficients at different flow rates

because it gets more time, the heat transfer phenomenon is erratic and nonuniform with higher degree of variations. Another observation that can be drawn from Fig. 6 is that the heat transfer rate at higher flow rate shows a fast change in gradient much below the temperature of 100 °C. This may be because of the fact that water pressurized up to 5 bar is used on the other side and is receiving heat from hot oil through the heat exchanger. The water under this condition gets vaporized and forms a vapor blanket faster owing to the heat it receives from more turbulent oil flow from the other side.

Literature-reported correlation values of convective heat transfer coefficient for different flow rates

Similarly, Fig. 7 shows the variation in the convective heat transfer coefficient calculated on the basis of empirical correlations at 900 and 1200 kg h^{-1} of oil flow rate. It can be seen that the convective heat transfer coefficient is higher at higher oil flow rate and also that the variation in the values of heat transfer coefficients is almost uniformly patterned matching the theoretical understanding of the phenomenon. This is contrary to the experimentally observed values as discussed with Fig. 6 in the previous section. However, the theoretical and empirical relationships do not give consideration to the time factor and to the nature and chemistry of oil. The actual heat transport phenomena may sometimes vary because the oil may behave in different manner depending upon its chemistry, which in turn may affect its wettability of the contacting surfaces and upon the time for which the oil is in contact with the heat transferring surfaces.

Comparison of experimental v/s literature-reported correlation values with respect to flow rate

Convective heat transfer coefficient "h_i" comparison at 900 kg h^{-1} of oil flow rate is shown in Fig. 8. While the simulated heat transfer coefficient shows an increasing trend with a definite gradient, the experimental values show increasing trend initially with respect to increase in temperature up to a certain limit, and then, it drops drastically to low value from 100 °C up to about 150 °C from where it again starts to increase. This trend can be attributed to the fact that the heat from oil is getting transferred to water through the heat exchanger surface. Once the steady-state temperature on all the contact surfaces reaches 100 °C, there is a small amount of vapor blanketing effect on the outer surface where water is the other fluid, owing to which the heat transfer gets reduced. However, in the experiment within the capability of the test setup the water used was at about 5 bar pressure as well as in actual application of concentrated solar power plants where water is at a pressure of about 15 bars. Subsequently, under the combined effect of reduction in the vapor blanket with time as well as the increase in oil temperature, the heat transfer starts to increase beyond 150 °C. However, from this point onward, the water is under a two-phase flow with some amount of vapors in it, and hence, the oil is not able to transfer much heat. Such two-phase flow systems are beyond the scope of present study and hence have not been considered here.

Convective heat transfer coefficient "h_i" comparison at 1200 kg h^{-1} of oil flow rate is shown in Fig. 9. While the simulated heat transfer coefficient shows an increasing trend with a definite gradient, the experimental values show

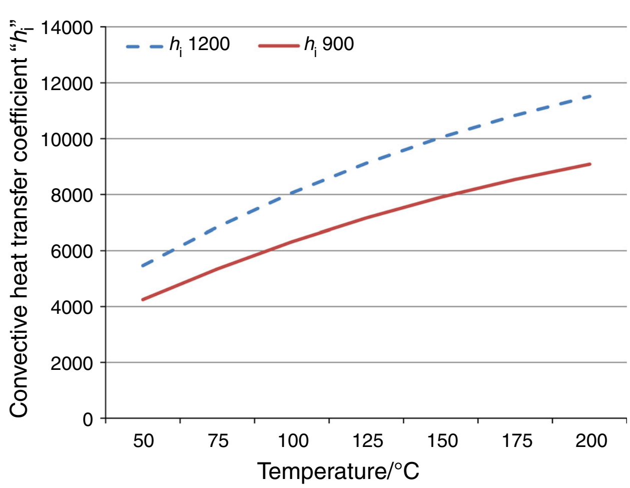

Fig. 7 Variation in simulated convective heat transfer coefficients at different flow rates

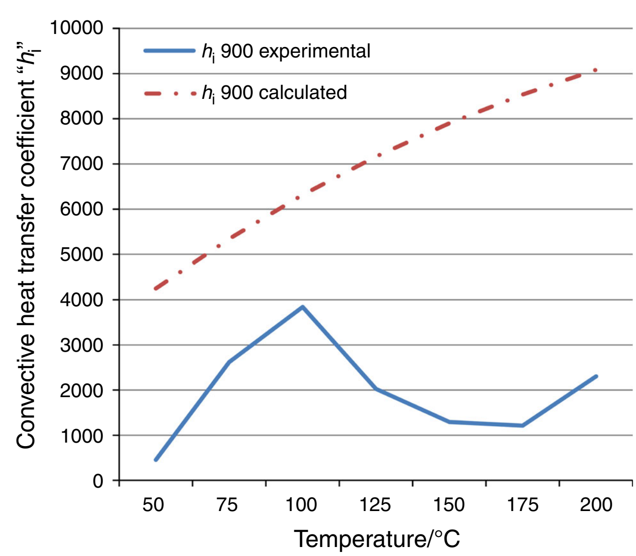

Fig. 8 Convective heat transfer coefficient comparison at 900 kg h^{-1} of oil flow

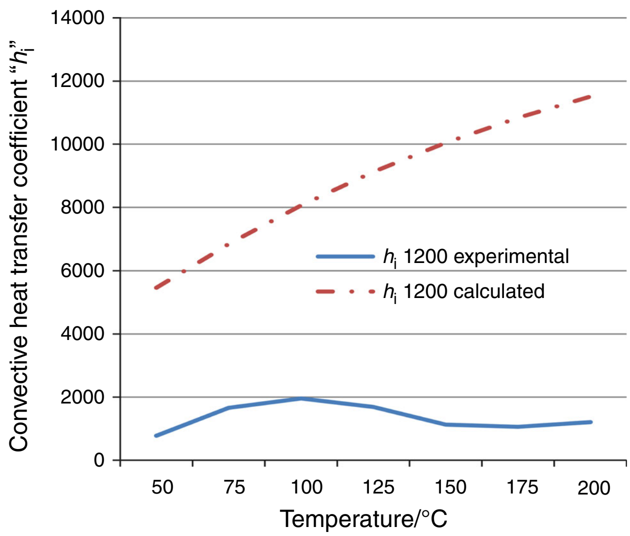

Fig. 9 Convective heat transfer coefficient comparison at 1200 kg h^{-1} of oil flow

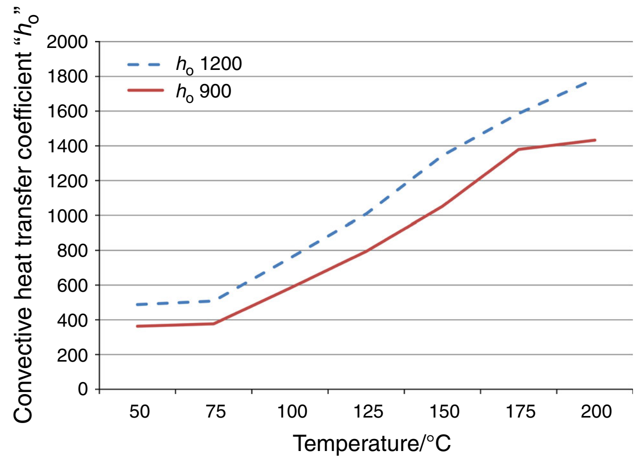

Fig. 10 Experimental convective heat transfer coefficients at different flow rates on water side

increasing trend initially with respect to increase in temperature up to a certain limit and then drop slightly from 100 °C up to about 150 °C from where it again starts to increase slowly. This trend can be attributed to the fact that the heat from oil is getting transferred to water through the heat exchanger surface. Once the steady-state temperature on the entire contact surface reaches 100 °C, there is a small amount of vapor blanketing effect on the outer surface where water is the other fluid, owing to which the heat transfer gets reduced. However, in the experiment within the capability of the test setup the water was used at 5 bar pressure as well as in actual application of concentrated solar power plants where water is at a pressure of about 15 bars. Subsequently, under the combined effect of reduction in the vapor blanket with time as well as the increase in oil temperature, the heat transfer starts to increase beyond 150 °C. However, from this point onward, the water is under a two-phase flow with some amount of vapors in it, and hence, the oil is not able to transfer much heat to water. Such two-phase flow systems are beyond the scope of the present study and hence have not been considered here.

Water side heat transfer coefficient variation

The experimental heat transfer coefficients on the water side are plotted in Fig. 10 at two different oil flow rates of 900 and 1200 kg h^{-1}, while the water flow rate in the experiments was kept constant at 1200 kg h^{-1} based on initial bench marking studies. The heat transfer coefficient on the water side is higher for higher flow rate of oil, signifying that more heat is transferred to water at high turbulence of heated oil. Further, the heat transfer coefficients are uniformly increasing with temperature following almost the same gradient and pattern.

Conclusions

Heat transfer oils are widely used in several industrial applications including the recently increasing trend of their use in solar energy applications as thermofluids for heat as well as power generation. One of the important parameters to assess the heat transfer capability of oil is thermal conductivity, but it has been widely reported in the literatures that, especially in case of fluids, and also in cases of fluids at high temperatures, the measurement methods of thermal conductivities give high variation and results are not very reliable. Hence, the obvious difficulty in using the indirect method of calculating the dimensionless quantities such as Reynolds numbers, Prandtl number and Nusselt number based on thermal conductivity, viscosity, density etc. from which convective heat transfer coefficient can be derived.

Heat transport properties are very important parameters in selection of heat transfer oils for a particular application. Most heat transport properties are determined either at lower temperatures or based on certain empirical correlations which are normally independent of the nature and chemistry of the oil. There are a variety of standard as well as nonstandard test methods being employed by researchers in evaluation of heat transport properties of oils. Most of these test methods evaluate the oil properties at low temperatures of below 100 °C and under atmospheric or lower pressures in the range of 1 bar. Thus, measurement of heat transport properties such as convective heat transfer coefficients of the oil, in experimental setup correlating as much actual operating conditions as possible, is one reliable method.

The authors have experimented with a fully synthetic grade heat transfer oil having kinematic viscosity of 2.48 cst @ 40 °C, flash point of 124 °C and pour point of 12 °C and used as working fluid in concentrated solar power plants. The oil was run in a specially designed closed-loop test setup comprising of electrical heaters to heat the oil to desired high temperatures of up to 200 °C, heat exchangers, oil pump and instruments to measure online oil properties like viscosity, density, temperatures, flow rates, etc. Convective heat transfer coefficients were then calculated based on the actual heat transfer that takes place from hot oil flowing in the tubes of the countercurrent heat exchanger and cold water flowing in the shell side with temperatures and flow rates being measured after steady-state conditions are achieved. Based on the extensive test run with the oil, authors have concluded the following in the study:

- A comparison of convective heat transfer coefficients and overall heat transfer coefficients of same oil under same conditions was carried out in co-current and countercurrent heat exchangers. Based on the experiments, it was observed that polynomial is the best curve obtained in case of counterflow heat exchanger with highest regression coefficients. Counterflow heat exchanger was then chosen by the authors to carry out all further analyses and evaluation of heat transfer properties of oil presented in the paper.
- Contrary to the common belief that the oil that has low density should also have lower viscosity, a comparison of the calculated and experimental values of density and viscosity was made. It was observed that the experimental values of the density are lower than the calculated values, while the experimental values of kinematic viscosity of the oil were found higher than the calculated values. This phenomenon was attributed to the synthetic nature of oil.
- The experiments were carried out under two flow rates of oil, i.e., 900 and 1200 kg h^{-1}. It was seen that the convective heat transfer coefficient is higher at lower oil flow rate, with the lower oil flow rate showing more variation in the values of heat transfer coefficients obtained. This was attributed to the fact that heat transfer is not a function of turbulence alone, but it is also a function of residence time for which the oil is in contact with the surface of heat exchanger.
- The convective heat transfer coefficients were also calculated on the basis of empirical correlations at 900 and 1200 kg h^{-1} of oil flow rate. It was seen that the convective heat transfer coefficient is higher at higher oil flow rate and that the variation in the heat transfer coefficients is almost uniformly patterned contrary to the experimentally observed values in the study. This explains that the empirical correlation does not take into consideration the residence time of oil's contact with the surface of heat exchanger.
- With respect to calculations based on empirical correlations and experimentally observed values, a comparison of convective heat transfer coefficient "h_i" for oil at the two flow rates of 900 and 1200 kg h^{-1} was also carried out. Under both the flow rates, the empirically calculated heat transfer coefficients show an increasing trend with a definite gradient. However, the experimental values show variable trend which is increasing initially with temperature up to a certain limit and then drops slightly from a temperature range of 100 to 150 °C, and then, it again starts to increase. This trend was attributed to the fact that the heat from oil is getting transferred to water where a small amount of vapor blanketing effect on the outer surface of tube is observed. Subsequently, under the combined effect of reduction in the vapor blanket with time as well as the increase in oil temperature, the heat transfer starts to increase in the experiments.
- In case of the convective heat transfer coefficient for water flowing on the shell side under constant flow rate, the heat transfer coefficient was found to be higher for higher flow rate of oil, signifying that more heat is transferred to water at high turbulence of heated oil. Further, the heat transfer coefficients are uniformly increasing with temperature following almost the same gradient and pattern.

The actual heat transport phenomena may sometimes vary because the oil may behave in different manner depending upon its chemistry, which in turn may affect its wettability of the heat transferring surfaces and also upon the time for which the oil is in contact with the heat transferring surfaces. The theoretical or empirical relationships do not give consideration to the time factor and to the nature and chemistry of oil. While selecting the oil for an application, specially for high temperatures beyond 100 °C, it is always good if actual heat transfer phenomenon can be studied experimentally under laboratory setup in the best possible actual operating conditions. This is important because oil may behave differently in actual operating conditions than that postulated in theory and the experiments shall be useful in optimizing the oil formulation as well as the overall operational cost. Determination of convective heat transfer coefficient of oil is a useful way of assessing the heat transport property of oils during its formulation and research. Convective heat transfer coefficients can be regarded as the overall effect of all the heat transport characteristics of oil such as the density, viscosity, specific heat capacity and conductivity or heat diffusivity. Different researchers have used different methods for evaluating the convective heat transfer coefficient of oils, and so far no test method has been standardized for the same. There was a need felt for developing a uniformly acceptable test method that can determine the overall heat transport properties of oil, such as the convective heat transfer coefficients.

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References

1. Srivastva U, Malhotra RK, Kaushik SC. Recent developments in heat transfer fluids used for solar thermal energy applications. J Fundam Renew Energy Appl. 2015. https://doi.org/10.4172/20904541.1000189.
2. Srivastva U, Malhotra RK, Kaushik SC. Advancements in heat transfer fluids for concentrated solar power plants. In: Petrotech 2016, New Delhi; 2016.
3. Srivastva U, Malhotra RK, Kaushik SC. Comparative review of thermal conductivity measurements techniques of heat transfer oils. In: International symposium on fuels and lubricants, Indian Society of Fuels and Lubricants, New Delhi; 2016.
4. Srivastva U, Malhotra RK, Kaushik SC. Review of heat transport properties of solar heat transfer fluids. J Therm Anal Calorim. 2017;130–132:605–21.
5. Parker WJ, Jenkins RJ, Butler CP, Abbott GL. Method of determining thermal diffusivity, heat capacity and thermal conductivity. J Appl Phys. 1961;32–9:1679–84.
6. Hoffman HW, Cohen SJ. Fused salt heat transfer, Part III: forced convection heat transfer in circular tubes containing $NaNO_2$–KNO_3–$NaNO_3$ eutectic. Report No. ORNL-2433; 1960.
7. Yu-ting Wu, Bin Liu, Chong-fang Ma, Hang Guo. Convective heat transfer in the laminar–turbulent transition region with molten salt in a circular tube. Exp Therm. Fluid Sci. 2009;33:1128–32.
8. Garg P, Alvarado JL, Marsh C, Carlson TA, Kessler DA, Annamalai K. An experimental study on the effect of ultrasonication on viscosity and heat transfer performance of multi-wall carbon nanotube-based aqueous nanofluids. Int J Heat Mass Transf. 2009;52:5090–101.
9. Sohal MS, Sabharwall P, Calderoni P, Wertsching AK, Grover BS, Sharpe P. Conceptual design of forced convection molten salt heat transfer testing loop. Idaho Falls: Idaho National Laboratory; 2010.
10. Mohammed HA, Bhaskarana G, Shuaib NH, Saidur R. Heat transfer and fluid flow characteristics in microchannels heat exchanger using nanofluids: a review. Renew Sustain Energy Rev. 2011;15:1502–12.
11. Sarkar Jahar. A critical review on convective heat transfer correlations of nanofluids. Renew Sustain Energy Rev. 2011;15:3271–7.
12. Tumuluri K, Alvarado JL, Taherian H, Marsh C. Thermal performance of a novel heat transfer fluid containing multi walled carbon nanotubes and microencapsulated phase change materials. Int J Heat Mass Transf. 2011;54:5554–67.
13. You C, Zhang W, Yin Z. Modeling of fluid flow and heat transfer in a trough solar collector. Appl Therm Eng. 2013;54:247–54.
14. Lopez-Gonzalez D, Valverde JL, Sanchez P, Sanchez-Silva L. Characterization of different heat transfer fluids and degradation study by using a pilot plant device operating at real conditions. Energy. 2013;54:240–50.
15. Lu J, He S, Liang J, Ding J, Yang J. Convective heat transfer in the laminar–turbulent transition region of molten salt in annular passage. Exp Therm Fluid Sci. 2013;51:71–6.
16. Chen C, Wu Y-T, Wang S-T, Ma C-F. Experimental investigation on enhanced heat transfer in transversally corrugated tube with molten salt. Exp Therm Fluid Sci. 2013;47:108–16.
17. Lu J, Sheng X, Ding J, Yang J. Transition and turbulent convective heat transfer of molten salt in spirally grooved tube. Exp Therm Fluid Sci. 2013;47:180–5.
18. Rahmati AR, Roknabadi AR, Abbaszadeh M. Analysis of laminar mixed convection in an inclined square lid-driven cavity with a nanofluid by using an artificial neural network. Heat Transf Res. 2014;45:361–90.
19. Haghighi EB, Saleemi M, Nikkam N, Khodabandeh R, Toprak MS, Muhammed M, Palm B. Accurate basis of comparison for convective heat transfer in nanofluids. Int Commun Heat Mass Transf. 2014;52:1–7.
20. Biencinto M, González L, Zarza E, Díez LE, Javier M-A. Performance model and annual yield comparison of parabolic-trough solar thermal power plants with either nitrogen or synthetic oil as heat transfer fluid. Energy Convers Manag. 2014;87:238–49.
21. Muñoz-Anton J, Biencinto M, Zarza E, Díez LE. Theoretical basis and experimental facility for parabolic trough collectors at high temperature using gas as heat transfer fluid. Appl Energy. 2014;135:373–81.
22. Nikkam N, Haghighi EB, Saleemi M, Behi M, Khodabandeh R, Muhammed M, Palm B, Toprak MS. Experimental study on preparation and base liquid effect on thermo-physical and heat transport characteristics of α-SiC nanofluids. Int Commun Heat Mass Transf. 2014;55:38–44.
23. Selvakumar P, Somasundaram P, Thangavel P. Performance study on evacuated tube solar collector using therminol D-12 as heat transfer fluid coupled with parabolic trough. Energy Convers Manag. 2014;85:505–10.
24. Suganthi KS, Vinodhan VL, Rajan KS. Heat transfer performance and transport properties of ZnO–ethylene glycol and ZnO–ethylene glycol–water nanofluid coolants. Appl Energy. 2014;135:548–59.
25. Harris A, Kazachenko S, Bateman R. Measuring the thermal conductivity of heat transfer fluids via the modified transient plane source (MTPS). J Therm Anal Calorim. 2014;116:1309–14.
26. Derakhshan MM, Akhavan-Behabadi MA, Mohseni SG. Experiments on mixed convection heat transfer and performance evaluation of MWCNT–oil nanofluid flow in horizontal and vertical microfin tubes. Exp Therm Fluid Sci. 2015;61:241–8.
27. Esfe MH, Naderi A, Akbari M, Afrand M, Karimipour A. Evaluation of thermal conductivity of COOH-functionalized MWCNTs/water via temperature and solid volume fraction by using experimental data and ANN methods. J Therm Anal Calorim. 2015;121:1273–8.
28. Bahiraei M, Hangi M, Saeedan M. A novel application for energy efficiency improvement using nanofluid in shell and tube heat exchanger equipped with helical baffles. Energy. 2015;93:2229–40.
29. Afrand M, Sina N, Teimouri H, Mazaheri A, Safaei MR, Esfe MH, Kamali J, Toghraie D. Effect of magnetic field on free convection in inclined cylindrical annulus containing molten potassium. Int J Appl Mech. 2015;7:1550052–68.
30. Esfe MH, Saedodin S, Wongwises S, Toghraie D. An experimental study on the effect of diameter on thermal conductivity

and dynamic viscosity of Fe/water nanofluids. J Therm Anal Calorim. 2015;119:1817–24.
31. Saeedan M, Bahiraei M. Effects of geometrical parameters on hydrothermal characteristics of shell-and-tube heat exchanger with helical baffles: numerical investigation, modeling and optimization. Chem Eng Res Des. 2015;96:43–53.
32. Raei B, Shahraki F, Jamialahmadi M, Petyghambarzadeh SM. Experimental study on the heat transfer and flow properties of γ-Al_2O_3/water nanofluid in a double-tube heat exchanger. J Therm Anal Calorim. 2017;127:2561–75.
33. Hosseinzadeh M, Heris SZ, Beheshti A, Shanbedi M. Convective heat transfer and friction factor of aqueous Fe_3O_4 nanofluid flow under laminar regime. J Therm Anal Calorim. 2016;124:827–38.
34. Bahiraei M. Particle migration in nanofluids: a critical review. Int J Therm Sci. 2016;109:90–113.
35. Goodarzi M, Kherbeet AS, Afrand M, Sadeghinezhad E, Mehrali M, Zahedi P, Wongwises S, Daharih M. Investigation of heat transfer performance and friction factor of a counter-flow double-pipe heat exchanger using nitrogen-doped, graphene-based nanofluids. Int Commun Heat Mass Transfer. 2016;76:16–23.
36. Soltanimehr M, Afrand M. Thermal conductivity enhancement of COOH-functionalized MWCNTs/ethylene glycol-water nanofluid for application in heating and cooling systems. Appl Therm Eng. 2016;105:716–23.
37. Baratpoura M, Karimipour A, Afrand M, Wongwises S. Effects of temperature and concentration on the viscosity of nanofluids made of single-wall carbon nanotubes in ethylene glycol. Int Commun Heat Mass Transfer. 2016;74:108–13.
38. Eshgarf H, Afrand M. An experimental study on rheological behavior of non-Newtonian hybrid nano-coolant for application in cooling and heating systems. Exp Therm Fluid Sci. 2016;76:221–7.
39. Dardan E, Afrand M, Isfahani AHM. Effect of suspending hybrid nano-additives on rheological behavior of engine oil and pumping power. Appl Therm Eng. 2016;25A:524–34.
40. Afrand M, Toghraie D, Sina N. Experimental study on thermal conductivity of water-based Fe_3O_4 nanofluid: development of a new correlation and modeled by artificial neural network. Int Commun Heat Mass Transf. 2016;75:262–9.
41. Esfe MH, Afrand M, Yan W-M, Yarmand H, Toghraie D, Dahari M. Effects of temperature and concentration on rheological behavior of MWCNTs/SiO_2 (20–80)-SAE40 hybrid nano-lubricant. Int Commun Heat Mass Transf. 2016;76:133–8.
42. Toghraie D, Alempour SM, Afrand M. Experimental determination of viscosity of water based magnetite nanofluid for application in heating and cooling systems. J Magn Mater. 2016;417:243–8.
43. Esfe MH, Yan W-M, Afrand M, Sarraf M, Toghraie D, Dahari M. Estimation of thermal conductivity of Al_2O_3/water (40%)-ethylene glycol (60%) by artificial neural network and correlation using experimental data. Int Commun Heat Mass Transf. 2016;74:125–8.
44. Zarringhalam M, Karimipour A, Toghraie D. Experimental study of the effect of solid volume fraction and Reynolds number on heat transfer coefficient and pressure drop of CuO-water nanofluid. Exp Therm Fluid Sci. 2016;76:342–51.
45. Esfe MH, Afrand M, Gharehkhani S, Rostamian H, Toghraie D, Dahari M. An experimental study on viscosity of alumina-engine oil: effects of temperature and nanoparticles concentration. Int Commun Heat Mass Transf. 2016;76:202–8.
46. Nazari S, Toghraie D. Numerical simulation of heat transfer and fluid flow of water–CuO Nanofluid in a sinusoidal channel with a porous medium. Physica E. 2017;87:134–40.
47. Aghanajafi A, Toghraie D, Mehmandoust B. Numerical simulation of laminar forced convection of water–CuO nanofluid inside a triangular duct. Physica E. 2017;85:103–8.
48. Akbaria OA, Toghraie D, Karimipour A, Marzband A, Ahmadi GR. The effect of velocity and dimension of solid nanoparticles on heat transfer in non-Newtonian nanofluid. Physica E. 2017;86:68–75.
49. Esfe MH, Afrand M, Rostamian SH, Toghraie D. Examination of rheological behavior of MWCNTs/ZnO-SAE40 hybrid nano-lubricants under various temperatures and solid volume fractions. Exp Therm Fluid Sci. 2017;80:384–90.
50. Esfe MH, Rostamian H, Toghraie D, Yan W-M. Using artificial neural network to predict thermal conductivity of ethylene glycol with alumina nanoparticle. J Therm Anal Calorim. 2017;126:643–8.
51. Esfe MH, Razi P, Hajmohammad MH, Rostamian SH, Sarsam WS, Arani AAA, Dahari M. Optimization, modeling and accurate prediction of thermal conductivity and dynamic viscosity of stabilized ethylene glycol and water mixture Al_2O_3 nanofluids by NSGA-II using ANN. Int Commun Heat Mass Transf. 2017;82:154–60.
52. Dehkordi RA, Esfe MH, Afrand M. Effect of functionalized single walled carbon nanotubes on thermal performance of antifreeze: an experimental study on thermal conductivity. Appl Therm Eng. 2017;120:358–66.
53. Afrand M. Experimental study on thermal conductivity of ethylene glycol containing hybrid nano-additives and development of a new correlation. Appl Therm Eng. 2017;110:1111–9.
54. Bahiraei M, Naghibzadeh SM, Jamshidmofid M. Efficacy of an eco-friendly nanofluid in a miniature heat exchanger regarding to arrangement of silver nanoparticles. Energy Convers Manag. 2017;144:224–34.
55. Bahiraei M, Khosravi R, Heshmatian S. Assessment and optimization of hydrothermal characteristics for a non-Newtonian nanofluid flow within miniaturized concentric-tube heat exchanger considering designer's viewpoint. Appl Therm Eng. 2017;123:266–76.
56. Foster R, Ghassemi M, Cota A. Solar energy: renewable energy and the environment. New York: CRC Press; 2009. ISBN 978-1-4200-7566-3.
57. Thomas LC. Heat transfer. Englewood Cliffs: Prentice Hall; 1991. ISBN 0-13-384942-2.
58. Mcdonald AG, Maganda HL. Thermo-fluids system design. West Sussex: Wiley; 2012. ISBN 978-1-118313633.
59. Solutia Europe S.A./N.V. Therminol VP-1—heat transfer fluid. http://www.solutia.com; Product information.