J Ther Eng, Vol. 7, No. 7, pp. 1628–1639, November, 2021

Journal of Thermal Engineering
Web page info: https://jten.yildiz.edu.tr
DOI: 10.14744/jten.2021.0006

Journal of Thermal Engineering

Research Article

Experimental investigation on the effect of thermophysical properties of a heat transfer fluid on pumping performance for a convective heat transfer system

Reza Abdu RAHMAN[1,*], Agri SUWANDI[1], Muhammad NURTANTO[2]

[1]*Department of Mechanical Engineering, Faculty of Engineering, Universitas Pancasila, DKI Jakarta 12640, Indonesia*
[2]*Department of Mechanical Engineering, Faculty of Engineering, Universitas Sultan Ageng Tirtayasa, Serang 42124, Banten, Indonesia*

ARTICLE INFO

Article history
Received: 09 May 2021
Accepted: 22 July 2021

Key words:
Convective heat transfer; Heat transfer fluid; High-temperature pump; Slip factor; Volumetric efficiency

ABSTRACT

Pumping performance is crucial for a convective heat transfer system. However, there is a limited study that specifically discusses the relation between thermophysical properties of a Heat Transfer Fluid (HTF) and pumping performance. This study aims to find the effect of the change in thermophysical properties of HTF on the pumping performance, particularly for the delivery rate, slip factor coefficient, and volumetric efficiency. In this study, five different HTFs are used to assess the effect of working temperature and pumping speed on the pumping performance. Delivery rate is evaluated by setting the pumping speed from 0 to 1300 RPM where the working fluid temperature is set at 40, 140, and 200 °C. It shows that the HTF with a lower viscosity has a better delivery rate. The slip coefficient for all working fluid is ranging between 0.11–0.31 at temperature 200 °C. It is found that a higher working temperature for the fluid increases the slip coefficient and delivery rate. The volumetric efficiency is directly affected by the slip ranging from 69 – 89% at 200 °C. The heat transfer rate ranges from 40 – 98 °C for all fluids, which is mainly affected by the volumetric efficiency of the pump and also pumping speed where a higher pumping speed decreases the heat transfer rate. It can be concluded that the change in thermophysical properties of the working fluid will change the pumping performance. Therefore, it is important to adjust the pumping operation according to the temperature and properties of the working fluid to achieve the highest heat transfer rate for a convective heat transfer system.

Cite this article as: Rahman RA, Suwandi A, Nurtanto M. Experimental investigation on the effect of thermophysical properties of a heat transfer fluid on pumping performance for a convective heat transfer system. J Ther Eng 2021;7(7):1628–1639.

*Corresponding author.
*E-mail address: reza.a@univpancasila.ac.id

This paper was recommended for publication in revised form by Regional Editor Chandramohan VP

Published by Yıldız Technical University Press, İstanbul, Turkey

INTRODUCTION

The innovation and improvement in thermal engineering are intended to enhance the overall efficiency of the system. It is driven by the motivation to respond to global pressure on crisis energy and reduce the carbon footprint [1]. A good example is an increase in the utilization of concentrated solar power (CSP) and thermal waste for food and crops drying across the globe [2,3]. These innovations are feasible to be carried out on a small scale, hoping that more renewable energy can be utilized and broader renewable energy distribution [4].

Small scale solar dryer is operating at a temperature between 60 – 200 °C. The drying process by using an indirect solar dryer is generally better than the open-air drying method [5]. An indirect solar dryer uses Heat Transfer Fluid (HTF) to distribute the heat energy from the solar collector to the liquid-to-air heat exchanger [6]. Using HTF also improves the dryer's reliability, where thermal energy storage can be installed to extend the operation time at nighttime [7]. The operation of an indirect solar dryer during daytime is as follows: the HTF circulates from the thermal collector to the thermal storage for charging and then continues to the heat exchanger. The stored heat in thermal storage is released and absorbed by the HTF at nighttime, then distributes to the heat exchanger using a bypass line to minimize the heat losses. The method is proven more reliable since the drying process could be extended and increase production capacity [8].

Indirect solar dryers are generally equipped with a solar collector, thermal storage, heat exchanger, heat transfer fluid, and process pump to distribute the HTF [9]. As a small-scale system, the economic aspect should be carefully taken into account. Thus, several improvements to the equipment have been studied to increase the reliability of the system. The improvement is mainly intended to increase the overall thermal efficiency of the system [10]. For example, an advanced design of heat exchanger by using copper tubing for the dryer able to save the drying time between 10 – 21 hours compared to the open-air drying method [11]. The enhancement also can be done by using an integrated system with a parabolic trough collector equipped with thermal energy storage (PCM based) to promote a better quality of the dried product. For the working fluid, using nanofluid, oil and glycerin are proven better than using water by increasing the efficiency by 9.7%, 20.2% dan 12.4%, respectively [12]. Thermo-optical efficiency at the solar collector can be increased up to 34% by using CeO_2-water base nanofluid with the volumetric concentration of CeO_2 is 0.035% [13].

It seems the improvement on the equipment for indirect solar dryer system has a good outcome. Nonetheless, there is limited study which specifically focused on the aspect of the working fluid and process pump for indirect solar dryer. As is well known, the stability of the heat transfer fluid and the pumping performance has a mutual influence [14]. It is essential to address the issue since the process pump is a critical component for the convective heat transfer application [15]. The relationship between the working fluid and process pump in convective heat transfer directly influences the heat transfer rate and power specific consumption of the pump [16].

The interconnection between working fluid and pump for indirect solar dryer system or identical application might be rarely discussed since it is hard to get a small high-temperature process pump [17]. The process pump for a thermal system is quite distinct compare to a typical liquid pump for low-temperature application [18]. Specifically, it is associated with the nature of the application, which involves high temperature and high viscous fluid [19]. This type of pump usually requires a higher cost, make it is rarely used as a research topic for solar dryer systems, especially the use of indirect solar dryers in many developing countries with many technological and cost limitations [20].

In this case, the pump must meet several ideal aspects to be applied on a small scale, namely: broadly available, ideal for pumping viscous fluids, working under an elevated temperature, easy to manufacture, and low cost [21]. Screening from many types of pumps that can meet these criteria, a rotary-type positive displacement pump is a reasonable choice [22]. Among the rotary–type positive displacement pumps, a trochoid–gerotor pump is the most suitable option [23]. The fundamental aspect to consider a trochoid–gerotor pump as a suitable candidate is driven by its advantages; it has a compact design, simple, and already designed for pumping high viscous fluid [24]. With only one moving part (the inner rotor), customization can be easily made to create an HTF process pump for an indirect solar dryer. Moreover, the size of the gerotor is considerably tiny and commonly used as a lubricant pump for automobiles or motorcycles where it operates at elevated temperature with high viscous fluid (engine oil) [25], and also, the cost of the pump is considerably cheap.

There is limited research which discussed precisely the pumping performance under various HTFs for indirect solar dryer system, even though this issue is critical for heat transfer performance of the system [26–28]. The relationship between HTF and pumping performance can be observed based on delivery rate, slip factor dan volumetric efficiency. Since this research is related to convective heat transfer, the relationship between pumping performance and heat transfer must also be considered. In order to answer these challenges, this research aims to study the relation between thermophysical heat transfer fluid properties and pump performance by using a modified trochoid–gerotor pump as a process pump. The result of the study can be used as an essential reference to understand the heat transfer performance of the system based on the pumping characteristic and type of heat transfer fluid and also provide an alternative option to develop a small process pump

J Ther Eng, Vol. 7, No. 7, pp. 1628–1639, November, 2021

for convective heat transfer by using a trochoid–gerotor pump.

PUMP DESIGN AND EXPERIMENTAL SETUP

Gerotor Design Model

The gerotor consists of rotor and housing, which rotor is the only moving part [29]. The relative motion between rotor and housing creates a vacuum, where the fluid enters from the suction port to the housing and forced out by the rotor through delivery port. The steady motion of the rotor generates sufficient force for suction and discharge of the working fluid. Modification of the gerotor is relatively easy because the model is compact, small, and adjustable. For this research, a gerotor oil pump for motorcycles is chosen because it is low-cost and vastly available.

Several additional parts are made in one assembly to develop a gerotor-based process pump. Figure 1 shows the cutting view from the designed gerotor assy. The detail of Figure 1 is as follow: a solid shaft (A) is installed at the center axis of the gerotor pump, the shaft is supported by upper (B) and lower (C) bearing, a rubber seal is installed at the lower shaft (D) to prevent leakage due to shaft rotation, a gasket is placed at the joint between lower and upper assy. The upper assy is manufactured from carbon steel to stand with high-temperature fluid and support external heating through conduction from the reservoir body. In this scenario, an electrical heater is placed within the reservoir. The void volume of the reservoir is 280 ml.

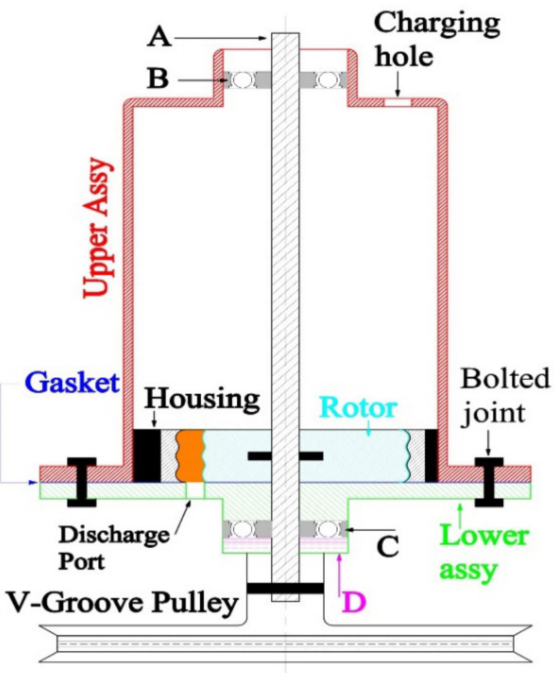

Figure 1. Cutting view of the designed gerotor pump.

In order to spin the rotor, an external motive source is needed. Considering the flexibility of the application and the controllability of the pump speed, a high torque DC motor (working voltage: 12–24 VDC, maximum working current: 5A, self-cooling fan) is used as the driver for the pump (close to standard reference IE1 [30]).

The power from the motor is transmitted to the pump by belt-pulley. After the final concept is decided, all components are assembled, as seen in Figure 2. The final design is taken after considering the characteristic of the pump, its application, manufacturability, and replicability for the other researcher. Several initial tests were done to ensure the workability of the pump as well as the leakage test [31]. Minor adjustments were made after the initial test, especially the adjustment of the belt tension.

Experimental Setup

The main objective of the experiment is to study the effect of the thermophysical properties of the working fluid on the pump performance. Five different working fluids are used to evaluate the performance of the pump. Two working fluids are commonly used for heat transfer fluid application (thermal oil and smooth fluid), where the rest of the working fluids are mineral oil-based lubricants (Table 1). Five different working fluids with various viscosity are used to observe the delivery rate of the pump at the given speed under specified temperature. A different working fluid is also used to study the effect of the change in physical properties of the working fluid based on temperature for the slip factor and volumetric efficiency of the pump.

The next step is setting the apparatus. Gerotor is working at relatively low pressure [32]. Consequently, it has to be modified to elevate the internal pressure at the suction port

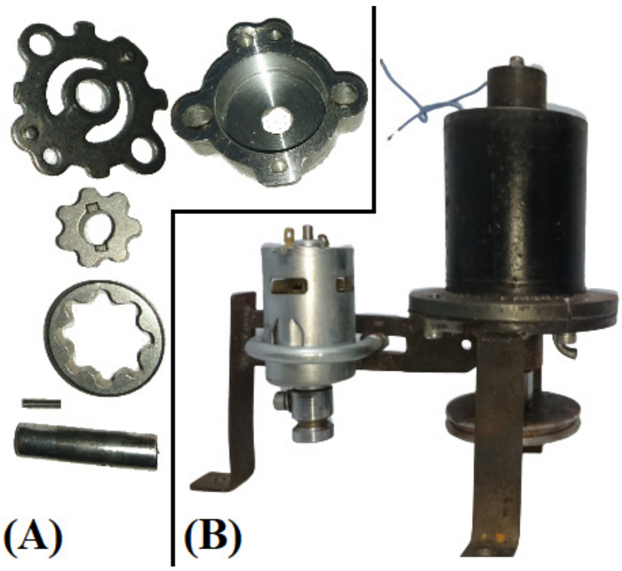

Figure 2. Final assembly for the gerotor pump and motor driver.

Table 1. Physical properties of the working fluids

Fluid	Boiling point (°C)	Viscosity (@ 40 °C, cSt)	Density (g/cm³)	Code
Thermal Oil (AT-400)	253–258	2.2–2.8	1.048 (@40 °C)	HTF_1
Smooth fluid (SF-05)	251–255	2.5–3.5	1.031 (@40 °C)	HTF_2
Oil SAE 30	280	96	0.873 (@15 °C)	HTF_3
Oil SAE 40	280	129.5	0.881 (@15 °C)	HTF_4
Oil SAE 50	320	223	0.890 (@15.6 °C)	HTF_5

to minimize the working load of the gerotor. The gerotor assy is adjusted relatively higher than the heat exchanger (Figure 3). The adjustment is intended to raise the hydrostatic pressure at the suction port, where it can be taken without changing the design of the gerotor assy. The gerotor assy is also designed for heating the fluid. Hence it required a reservoir to accommodate the fluid. Locating the reservoir before the suction port will elevate the hydrostatic pressure significantly where it helps to decrease the slip factor of the pump [33]. Since the gerotor is working at relatively low pressure, therefore an auxiliary pump is required for assisting the liquid to return to the gerotor assy. The auxiliary pump is a typical transfer pump which generally available and cheap since it works at relatively low temperatures. Figure 3 shows the schematic experimental, where the detail of the component is summarized in Table 2.

The arrangement of the apparatus and measurement can be adjusted flexibly. The critical aspect of designed apparatus is that the high-temperature fluid is pumped by gerotor assy, where the temperature of the fluid can be controlled easily by setting the electric heater within the gerotor assy. As the heat exchanger absorbs the heat from the HTF, the HTF temperature falls then collected into the reservoir. Inside the reservoir, a submersible pump (or any regular pump) works as an auxiliary pump, pumping the cold HTF to the preheater and return to the gerotor assy. Thus, the auxiliary pump is merely pushing the cold fluid, which is safe for the most available pump. For the record, the preheater is designed to heat the working fluid before entering the gerotor assy to minimize the heating time inside the assy.

The gerotor is driven by an electric DC motor and controlled by Pulse Width Modulation (PWM). PWM also controls the submersible pump (turbine type) inside the reservoir. Both of the flow (high-temperature low-pressure flow from gerotor assy and high-pressure low-temperature flow from reservoir) are monitored through flowmeter (F_1 and F_2) and pressure gauge (P_1 and P_2). The flow is adjusted carefully to maintain the pumped working fluid between the reservoir and gerotor assy. Thermocouple type K is attached to measure the temperature of the discharged working fluid from gerotor (T_1) and after the heat exchanger (T_2). All instrumentation for the measurement can be arranged freely based on the research purposes to support the flexibility of the application. In this study, the flow meter is used to measure the delivery rate and slip coefficient of the pump. Pressure gauges are used to maintain the working pressure within the system and thermocouple to measure the working fluid temperature before and after the heat exchanger.

The gerotor is driven by an electric DC motor and controlled by Pulse Width Modulation (PWM). PWM also controls the submersible pump (turbine type) inside the reservoir. Both of the flow (high-temperature low-pressure flow from gerotor assy and high-pressure low-temperature flow from reservoir) are monitored through flowmeter (F_1 and F_2) and pressure gauge (P_1 and P_2). The flow is adjusted carefully to maintain the pumped working fluid between the

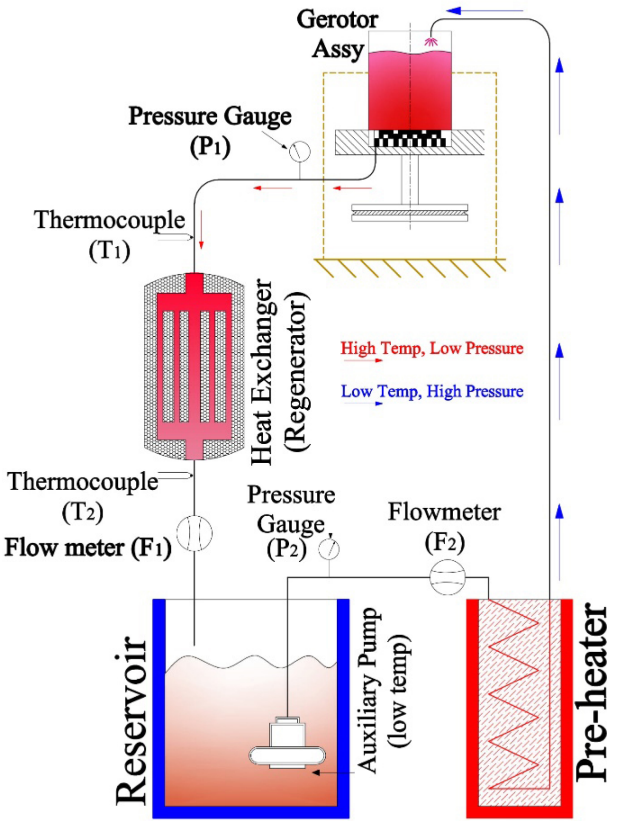

Figure 3. The schematic of the apparatus for experimental validation.

Table 2. Detail components of the apparatus

Components		**Detail**
Heat Exchanger	$Ø_{s,i}$	50 mm
	$Ø_{s,o}$	70 mm
	$Ø_{t,i}$	5 mm
	$Ø_{t,o}$	6.35 mm
	n_t	19
	h	85 mm
	Shell material	Polyoxymethylene
	Tube material	Copper
Gerotor Assy	Vr	280 ml
	N_1	8
	N_2	7
Instrumentation	P_1, P_2	Bourdon tube, 0 – 2.5 bar
	F_1, F_2	Oval gear flow meter
	T_1, T_2	Thermocouple type K
Auxiliary part	$Ø_{p,i}$	5 mm
	$Ø_{p,0}$	4 mm
	Submersible pump	Turbine type, 12V
	PWM controller	12 – 24 V, 10A, 400 W
	Electric heater	Cartridge type heater

reservoir and gerotor assy. Thermocouple type K is attached to measure the temperature of the discharged working fluid from gerotor (T_1) and after the heat exchanger (T_2). All instrumentation for the measurement can be arranged freely based on the research purposes to support the flexibility of the application. In this study, the flow meter is used to measure the delivery rate and slip coefficient of the pump. Pressure gauges are used to maintain the working pressure within the system and thermocouple to measure the working fluid temperature before and after the heat exchanger.

Two types of tubes are used for distributing the working fluid, namely the Polytetrafluoroethylene (PTFE) tube, and copper tube. The PTFE is used to minimizing the convective heat losses from the working fluid, while the copper tube is used to maximizing the heat transfer inside the preheater. The maximum temperature of the experiment is limited at 200 °C to prevent the melting of PTFE tube. This temperature is also suitable for indirect solar dryers and widely used for thermal systems related to industrial heat waste and other solar thermal applications at medium temperature (i.e., indirect solar water heater).

RESULTS AND DISCUSSION

Delivery Rate

The delivery rate (also called pumping capacity) provides vital pump information since it shows how much fluid can be pumped under a specified time. The Heat Transfer Fluids (HTFs) are tested to observe the delivery rate of the pump. As a variation, each fluid is tested by using three different temperatures. Figure 4 presents the delivery rate of each fluid at temperature 40 °C. There is an identic pattern for all fluids where the delivery rate increases as the pumping speed increases. It is the main characteristic of a positive displacement pump. From Figure 4, it also can be observed that delivery rate is varied for each HTF. The significant contributions for the differences are the physical properties of the HTF, particularly the viscosity. Fluid with lower viscosity is easier to be pumped than a fluid with a higher viscosity. It can be seen that at a speed of 1200 RPM, the delivery rate for HTF_1 reaches 229 ml/min, while for HTF_5, it is only 49 ml/min. HTF_5 has the highest viscosity compare to all HTF (Table 1) which is why it has the lowest delivery rate. The relation between viscosity and delivery rate also can be observed for HTF_2, HTF_3 and HTF_4, where lower viscosity value leads to higher delivery rate.

Increasing the temperature of the working fluid is affecting the total delivery rate of the pump. Figure 5 presents the graphical plot for the delivery rate by setting the fluid temperature at 140 °C. As shown, the delivery rate for each HTF is increased compared to the previous test, where the working fluid is set at 40 °C.

The delivery rate of each fluid is higher as the temperature rises. It is mainly affected by the change of viscosity for each fluid. Increasing the temperature will decrease the viscosity of each fluid. When the fluid temperature is increased, it affects the attractive intermolecular force within the fluid, which causes kinetic energy of the fluid increases. As a result of these changes, the attractive binding energy within the fluid is reduced, which reduces the fluid's viscosity. Fluid will be easier to flow if the attractive intermolecular force within its molecule reduced. It is the main reason why at elevated temperatures, the delivery rate for each liquid is increased.

According to Figure 5, HTF_1 and HTF_2 have a higher increment than HTF_3, HTF_4 and HTF_5. Compared to delivery rate at temperature 40°C, the average increment for HTF_1 and HTF_2 is 42.5 ml/min, while for HTF_3, HTF_4 and HTF_5 is only 25 ml/min. An escalate delivery rate for HTF_1 and HTF_2 is correlated with its nature as heat transfer fluid. HTF_1 and HTF_2 are designed to work as heat transfer fluid that has good thermal stability under its operating temperature range [14]. HTF_3, HTF_4 and HTF_5 are mineral oil-based lubricants, where the main feature of these liquids is stable viscosity under elevated temperature. Increasing the temperature for HTF_3, HTF_4, and HTF_5 has a minor effect on reducing its viscosity [34]. Figure 6 shows the delivery rate for temperature fluid at 200 °C, where the same phenomenon also can be observed.

HTF_1 and HTF_2 show a significant increment for the delivery rate, while the HTF_3, HTF_4 and HTF_5 still low. The delivery rate test under different working fluids with

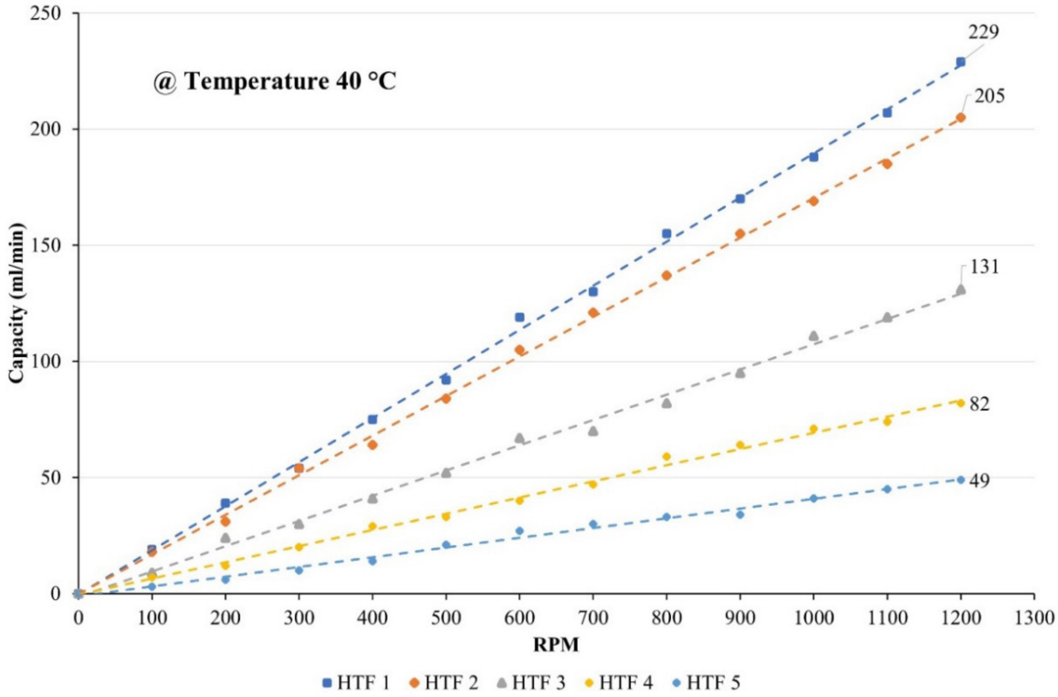

Figure 4. Delivery rate at temperature 40 °C.

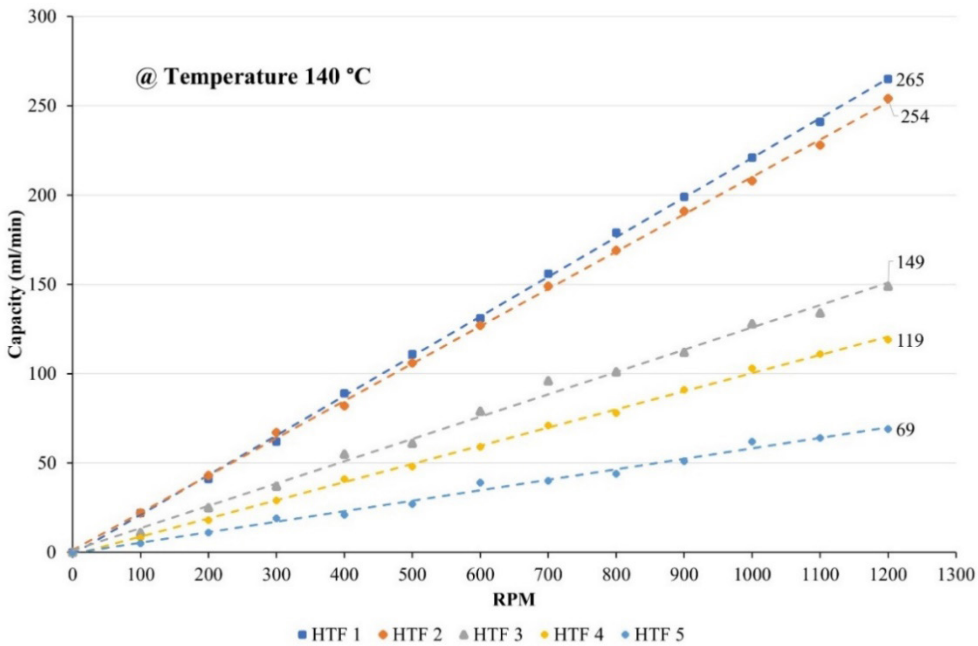

Figure 5. Delivery rate at temperature 140 °C.

different temperatures shows excellent information that the change of temperature for the working fluid affects the delivery rate of the pump. The temperature change will change the properties of the fluid. Thus, it can be concluded that the delivery rate is directly related to the properties of the working fluid, which from this test can be concluded that the fluid with a lower viscosity value has better fluidity and led to a higher delivery rate.

Slip Factor and Volumetric Efficiency

The slip factor is an essential parameter in a pumping system because it affects the operational performance of

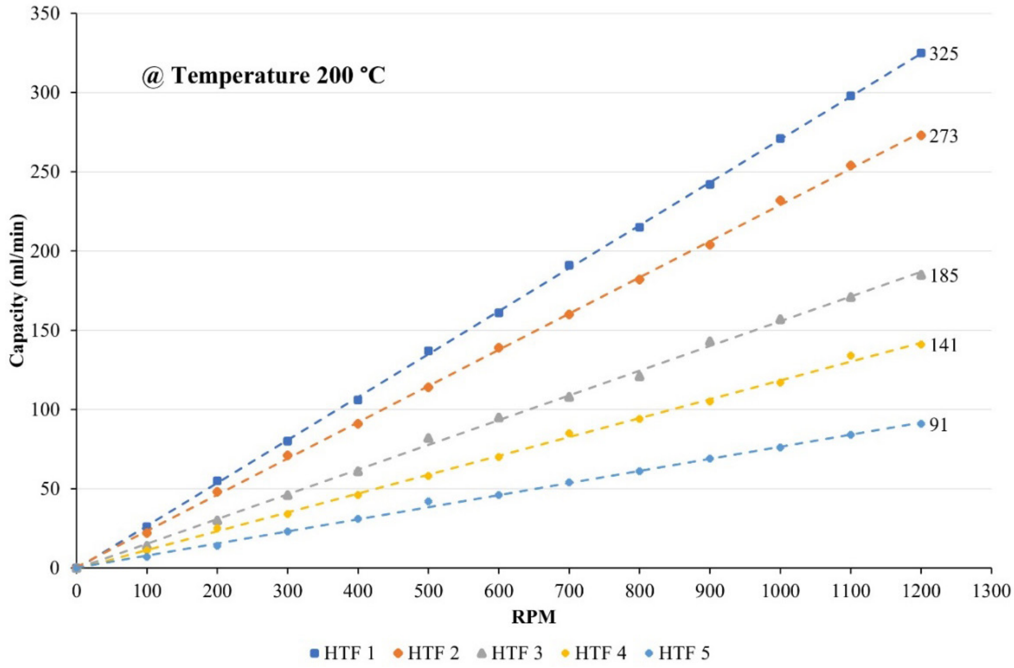

Figure 6. Delivery rate at temperature 200 °C.

the pump. For the gerotor pump, the slip occurs because of the pressure difference between suction and discharges port from the pump. The slip factor is defined as the amount of fluid leaked through the internal clearance of the pump (unit in m^3/h or cc/mins) or can be defined as the ratio or coefficient from the actual vs theoretical delivery rate [35]. Since there is no exact standard that regulates the standard test for measuring slip in a gerotor pump [36], therefore this study uses the slip factor coefficient for simplicity.

The slip coefficient for the gerotor pump is controlled by the internal clearance between the rotor and the housing, differential pressure between suction and discharge port and the pumping speed. The internal clearance at this point is fixed, while the discharge pressure is also set at a fixed value (1.5 bar at P_1) and including the pumping speed which is at 400 RPM. In order to observe the effect of thermophysical properties from the heat transfer fluid to the slip coefficient of the pump, the test uses temperature as variation. The working fluid temperature is started at 40 °C and increased steeply by 10 °C till it reaches the maximum temperature at 200 °C. The actual delivery rate is measured at a specific temperature. The measurement is taken 10 times at a given temperature to ensure the reliability of the measurement.

According to Figure 7, the slip coefficient increases with increasing fluid temperature. HTF_1 and HTF_2 have the highest slip coefficient with a significant rate of increment of slip values, while the slip coefficient for HTF_3, HTF_4 and HTF_5 is generally low with a minor effect as the temperature rises. The first aspect that makes the slip coefficient different in the two types of fluid is viscosity. The possibility of internal leakage between the rotor and the housing is reduced when a thicker fluid (high viscous fluid) is used. It can be observed that HTF_3, HTF_4 and HTF_5 at temperature 40 °C has a slip coefficient less than 0.1, while HTF_1 and HTF_2 have a higher value with slip coefficient 0.18 and 0.17, respectively.

The slip coefficient is increasing with the change in fluid properties. Besides the viscosity of the liquid, the density is also playing a role in the slip coefficient. Increasing the temperature will make the fluid expand, causing its density to decrease. The effect of temperature, which associated with a reduction in liquid density, is the expansion of molecules within the fluid. As the result of the reduction in density, the hydrostatic pressure at the suction port decreases and makes the differential pressure between suction and discharge port rise [37]. Although the density at room temperature for HTF_3, HTF_4 and HTF_5 is lower than HTF_1 and HTF_2, the effect of increment on HTF_3, HTF_4 and HTF_5 is negligible. These fluids have better density at elevated temperature since it designed as a lubricant which requires stable density.

HTF_3, HTF_4 and HTF_5 also have a higher viscosity and gives an additional advantage to reduce the slip coefficient. It can be recognized from Figure 7 where the increase in slip coefficient as the effect of temperature is relatively small, particularly for HTF_5 as the fluid with the highest viscosity; it has the lowest slip coefficient with the maximum value of 0.11 at temperature 200 °C. In contrast, the increasing temperature for HTF_1 and HTF_2 makes the slip coefficient

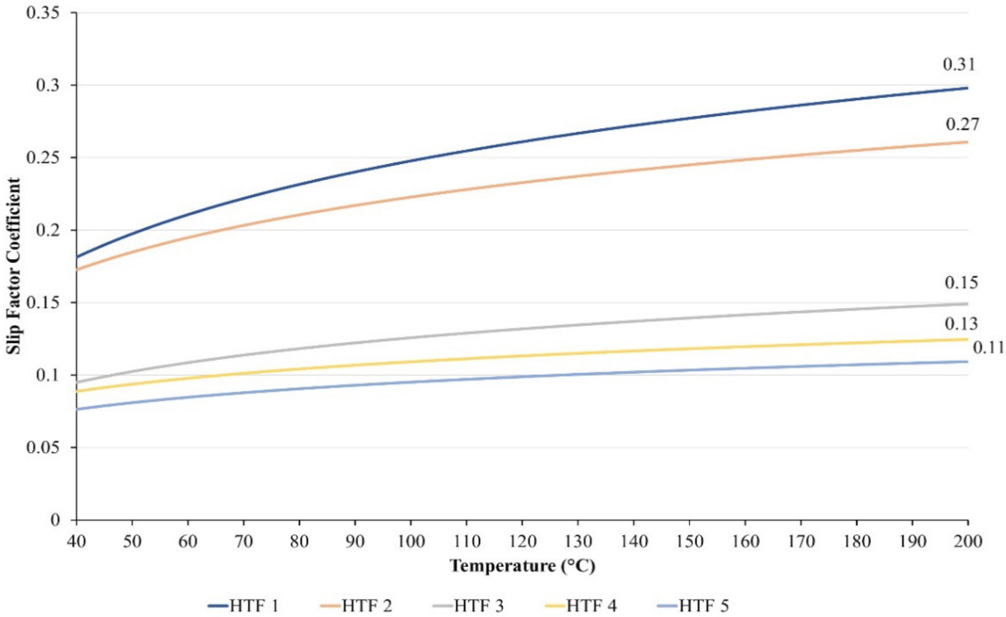

Figure 7. The effect of HTF temperature on slip factor coefficient.

increase significantly, which directly caused by its low viscosity. According to this phenomenon, it can be concluded that viscosity has more influence on slip coefficient than density, in good accordance with [38], where more viscous fluids tend to experience lower slip than non-viscous fluids.

From assessing the slip coefficient under different working temperatures of the fluid, the pump's volumetric efficiency can be obtained. Volumetric efficiency is used as a representation of the capability of the pump to distribute the fluid. Since the slip coefficient is a ratio between the actual and theoretical capacity, then the effect of the slip losses can be proven by the change of volumetric efficiency. The effect of slip losses based on fluid temperature against the volumetric efficiency is plotted in Figure 8. Gerotor is a pump with high volumetric efficiency. It can be validated by the initial value of volumetric efficiency for all fluids is above 80%. However, the changes in fluid properties due to temperature changes can be observed on the decrease in the pump's volumetric efficiency.

The change in thermophysical properties of the fluid is directly associated with the decrease in volumetric efficiency of the pump. It is essential information for a high-temperature process pump. The temperature of the working fluid affects its volumetric efficiency significantly. Although the volumetric efficiency for HTF_3, HTF_4, and HTF_5 is better than HTF_1 and HTF_2, it has to be considered that the first three fluids have a lower delivery rate than HTF_1 and HTF_2. Therefore, selecting a high-temperature process pump for heat transfer system should consider both the volumetric efficiency and delivery rate. The consideration is essential since the delivery rate will relate to the mass flow rate of the fluid, and volumetric efficiency will relate to the specific power consumption of the pump. These aspects should be taken care of for designing the thermal system to obtain the ideal working conditions for the pump to maximize the volumetric efficiency of the pump that will eventually improve the system's overall efficiency.

Heat Transfer Rate

The heat transfer rate is assessed by evaluating the heat transfer process between the fluid and the thermal load. The thermal load in this scenario is using a regenerator type heat exchanger. The heat transfer activity is measured by observing the temperature difference of the fluid before and after the heat exchanger (Figure 4). The heat exchanger is set at a specific dimension, so the assessment is entirely dependent on the temperature difference of the working fluid. The inlet temperature of the working fluid is kept constantly at 200 °C while the pumping speed is taken as the variation. Figure 9 plot the relationship between the pumping speed and temperature difference of the fluid before and after the heat exchanger.

The effect of the pumping speed for all working fluid is the same, where the temperature difference is decreased as the pumping speed increased. Increasing the pumping speed accelerates the flow rate, which causes heat transfer rate between the fluid and heat exchanger to drop significantly as the fluid flows faster inside the heat exchanger. A slower pumping speed can maximize the heat transfer since there is enough time for the fluid to release the heat. Besides the pumping speed, volumetric efficiency also plays a role in the heat transfer rate of each fluid. It can be observed during low pumping speed where temperature differences for all HTF are above 80 °C. The drawback of low delivery

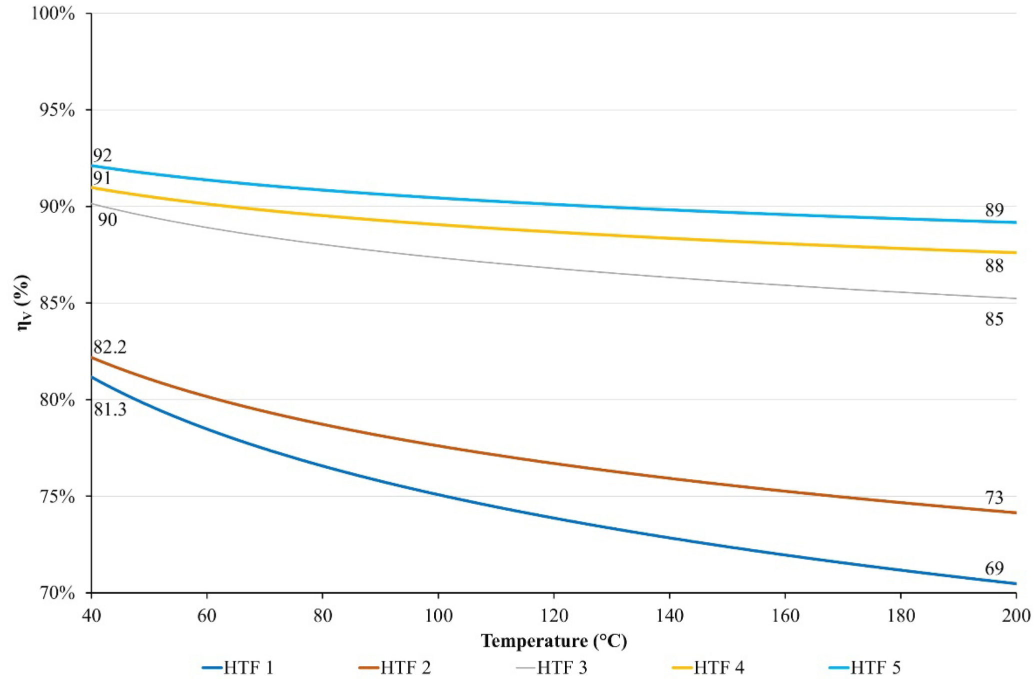

Figure 8. Volumetric efficiency under different temperature of working fluid.

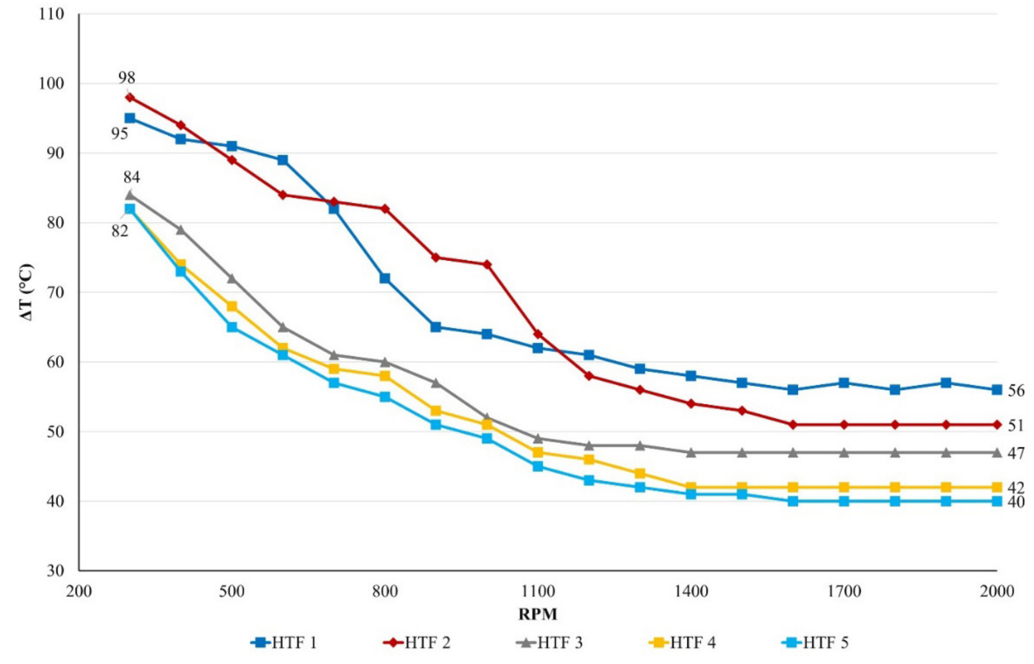

Figure 9. The heat transfer rate of the working fluid under different pumping speed.

rate for HTF_3, $HTF_{4,}$ and HTF_5 can be minimized as the volumetric efficiency for these fluids are high. Oppositely, HTF_1 and HTF_2 with low volumetric efficiency can still reach a noticeable performance for heat transfer rate since these fluids have a high delivery rate.

Unstable heat transfer rates for HTF_1 and HTF_2 at pumping speed 500 – 1100 RPM are observed. As the viscosity and density for these fluids fall at temperature 200 °C, it makes the slip increased, which causes the heat transfer rate of HTF_1 and HTF_2 less stable in that speed range. In contrast, lubricant-based HTF tends to be able to maintain heat transfer stability at the same speed. Apart from the stability of heat transfer, HTF-based fluids have a better thermal performance compared to lubricant-based fluids.

It is affected by the nature of HTF-based fluid, where the properties of these fluids are already designed to heat transfer application. Under this circumstance, the slip coefficient decreases the overall thermal performance of the HTF-based fluids at higher pumping speeds. Oppositely, the overall thermal performance for all lubricant-based fluids does not drop significantly as the pumping speed increased since the slip coefficient for these fluids is relatively lower than HTF-based fluids. Despite all that, all working fluid shows a satisfactory performance as heat transfer fluid.

CONCLUSION

The thermophysical properties of Heat Transfer Fluid (HTF) influence the pumping performance of the process pump. Delivery rate is directly related to the viscosity of the HTF. The increase in temperature of the working fluid makes the viscosity drop; it makes the delivery rate increase. Thermal Oil (AT–400), which has the lowest viscosity, can reach the highest delivery rate at 40, 140, and 200 °C with delivery rate 229, 265, and 325 ml/min, respectively. As the viscosity increased from smooth fluid, oil SAE 30, oil SAE 40 and oil SAE 50, the delivery constantly decreased which oil SAE 50 has a delivery rate of 49, 69, and 91 ml/min since it has the highest viscosity. It is clear evidence that fluid with lower viscosity has a better delivery rate. Changes in thermophysical properties due to temperature increases also affect the slip coefficient. As the temperature rises, the fluid's density and viscosity decrease and make the slip coefficient increase. It is observed from Thermal Oil (AT–400) with the highest slip coefficient at 0.31, where oil SAE 50 is 0.11. As the effect of slip coefficient, volumetric efficiency is reduced significantly for Thermal Oil (AT–400) where at temperature 200 °C and pumping speed 400 RPM, the volumetric efficiency of the pump drops to 69% where oil SAE 50 able to maintain a noticeable volumetric efficiency up to 89%.

Eventually, the effect of thermophysical properties of the working fluid on pump performance impacts the heat transfer rate of the system. The highest heat transfer rate is obtained by Thermal Oil (AT–400) and Smooth fluid (SF-05), though it is mainly affected by the nature of the fluid itself as heat transfer fluid. However, the slip coefficient of the pump for Thermal Oil (AT–400) and Smooth fluid (SF-05) is relatively high, which leads to an unstable heat transfer process from the fluid to the thermal load. For the others fluid, oil SAE 30, SAE 40, and SAE 50, stable heat transfer rates are observed since the pump can reach a suitable volumetric efficiency. This study provides essential information on the relationship of fluid properties with pumping performance on heat transfer rate in convective heat transfer systems. It can use as a reference to understand the effect of working fluid on pumping performance for a high-temperature process pump in a convective heat transfer system. Although the results of this study can provide important information related to the relationship between the thermophysical properties of HTF and pumping performance, more in-depth research is still needed relating to fluid flow analysis, especially to study the type of fluid flow under certain pumping conditions associated with the properties of the fluid used. The use of an effective pump can increase the heat transfer rate, which in turn can maximize the overall efficiency of the heat transfer system.

NOMENCLATURE

F_1	Flowmeter from the heat exchanger to the reservoir (ml/min)
F_2	Flowmeter from the reservoir to gerotor assy (ml/min)
h	Height of tubes (mm)
HTF	Heat Transfer Fluid
N_1	Number of housing teeth
N_2	Number of rotor teeth
n_t	Number of tubes
P_1	Pressure gauge for gerotor pump (bar)
P_2	Pressure gauge for submersible pump (bar)
$Ø_{p,i}$	Inside diameter of PTFE tube (mm)
$Ø_{p,o}$	Outside diameter of PTFE tube (mm)
$Ø_{s,i}$	Inside diameter shell side (mm)
$Ø_{s,o}$	Outside diameter shell side (mm)
$Ø_{t,i}$	Inside diameter tube side (mm)
$Ø_{t,o}$	Outside diameter tube side (mm)
Vr	Volume reservoir at gerotor assy (ml)

ACKNOWLEDGEMENT

The authors would like to thanks Rohadi Yusuf from SMKN 2 Kota Bekasi for manufacturing the pump.

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.

REFERENCES

[1] Ismail I, Rahman RA, Haryanto G, Pane EA. The optimal pitch distance for maximizing the power ratio for savonius turbine on inline configuration. Int J Renew Energy Res 2021;11:595–599.

[2] Reddy VS. Portable solar drying system with inbuilt pv module forstandalone forced convection operation .J Therm Eng 2020;6:92–98. [CrossRef]

[3] Oyedepo SO, Fakeye BA. Waste heat recovery technologies: pathway to sustainable energy development. J Therm Eng 2021;7:324–348. [CrossRef]

[4] Ismail I, Pane EA, Haryanto G, Okviyanto T, Rahman RA. A better approach for modified bach-type savonius turbine optimization. Int Rev Aerosp Eng 2021;14:159–165. [CrossRef]

[5] Lingayat A, Chandramohan VP, Raju VRK, Kumar A. Development of indirect type solar dryer and experiments for estimation of drying parameters of apple and watermelon. Therm Sci Eng Prog 2020;16:100477. [CrossRef]

[6] Asirvatham Lg. nanofluid heat transfer and applications. J Therm Eng 2015;1:113–115. [CrossRef]

[7] Yadav S, Chandramohan VP. Performance comparison of thermal energy storage system for indirect solar dryer with and without finned copper tube. Sustain Energy Technol Assess 2020;37:100609. [CrossRef]

[8] Lakshmi DVN, Muthukumar P, Nayak PK. Experimental investigations on active solar dryers integrated with thermal storage for drying of black pepper. Renew Energy 2021;167:728–739. [CrossRef]

[9] Güler HÖ, Sözen A, Tuncer AD, Afshari F, Khanlari A, Şirin C, et al. Experimental and CFD survey of indirect solar dryer modified with low-cost iron mesh. Sol Energy 2020;197:371–384. [CrossRef]

[10] Lingayat AB, Chandramohan VP, Raju VRK, Meda V. A review on indirect type solar dryers for agricultural crops – Dryer setup, its performance, energy storage and important highlights. Appl Energy 2020;258:114005. [CrossRef]

[11] Ndukwu MC, Simo-Tagne M, Abam FI, Onwuka OS, Prince S, Bennamoun L. Exergetic sustainability and economic analysis of hybrid solar-biomass dryer integrated with copper tubing as heat exchanger. Heliyon 2020;6:e03401. [CrossRef]

[12] Alimohammadi Z, Samimi Akhijahani H, Salami P. Thermal analysis of a solar dryer equipped with PTSC and PCM using experimental and numerical methods. Sol Energy 2020;201:157–177. [CrossRef]

[13] Sharafeldin MA, Gróf G. Evacuated tube solar collector performance using CeO2/water nanofluid. J Clean Prod 2018;185:347–356. [CrossRef]

[14] Wright C. The stability of thermal fluids.World Pumps 2016;2016:32–35. [CrossRef]

[15] Chaciga J, Tabu B, Okello D, Karidewa N. Construction of a High Temperature (~ 200 °C) Oil Pump for Solar Thermal Energy Storage System for Cooking Applications. Tanzania J Sci 2018;44:21–35.

[16] Rahmanian S, Hamzavi A. Effects of pump power on performance analysis of photovoltaic thermal system using CNT nanofluid. Sol Energy 2020;201:787–797. [CrossRef]

[17] Mohana Y, Mohanapriya R, Anukiruthika T, Yoha KS, Moses JA, Anandharamakrishnan C. Solar dryers for food applications: Concepts, designs, and recent advances. Sol Energy 2020;208:321–344. [CrossRef]

[18] Arneth H, Dötsch S. Pumps for heat transfer applications. World Pumps 2006;2006:22–27. [CrossRef]

[19] Yedidiah S. Effects of Temperature and Viscosity on Pump Performance. In: Centrifugal Pump User's Guidebook. Boston,MA: Springer; 1996. p. 279–290. [CrossRef]

[20] Kumar P, Singh D. Advanced technologies and performance investigations of solar dryers: a review. Renew Energy Focus 2020;35:148–158. [CrossRef]

[21] Bebee RS. Predicitive Maintenance of Pumps Using Condition Monitoring. Amsterdam, Netherlands: Elsevier Science; 2004. [CrossRef]

[22] Bill Forsthoffer WE. Pump characteristics—positive displacement vs centrifugal (kinetic). Washington: Elsevier Science; 2005;2:25–36. [CrossRef]

[23] Parr A. Hydraulic Pumps and Pressure Regulation. 1st ed. Amsterdam: Elsevier Ltd; 2011. [CrossRef]

[24] Kwon SM, Kang HS, Shin JH. Rotor profile design in a hypogerotor pump. J Mech Sci Technol 2009;23:3459–3470. [CrossRef]

[25] Lee SH, Kwak HS, Han G Bin, Kim C. Design of gerotor oil pump with 2-expanded cardioids lobe shape for noise reduction. Energies 2019;12:1–16. [CrossRef]

[26] Hodel AE. Heat Transfer Fluids. Chem Process 2004;67:45.

[27] Abu Talib AR, Hilo AK. Fluid flow and heat transfer over corrugated backward facing step channel. Case Stud Therm Eng 2021;24:100862. [CrossRef]

[28] Krishna Y, Faizal M, Saidur R, Ng KC, Aslfattahi N. State-of-the-art heat transfer fluids for parabolic trough collector. Int J Heat Mass Transf 2020;152:119541. [CrossRef]

[29] Stryczek J, Bednarczyk S, Codina E, Gamez-Montero PJ, Ivanovic L, Matejic M. Gears or rotors - three approaches to design of working units of hydraulic machines. IOP Conf Ser Mater Sci Eng 2019;659: 012006. [CrossRef]

[30] Fong J, Ferreira FJTE, Silva AM, de Almeida AT. IEC61800-9 system standards as a tool to boost the efficiency of electric motor driven systems worldwide. Inventions 2020;5:1–15. [CrossRef]

[31] Ismail I, John J, Pane EA, Maulana R, Rahman RA, Suwandi A. Experimental evaluation for the feasibility of test chamber in the open-loop wind tunnel. WSEAS Trans Fluid Mech 2021;16:120–126. [CrossRef]
[32] Jacazio G, De Martin A. Influence of rotor profile geometry on the performance of an original low-pressure gerotor pump. Mech Mach Theory 2016;100:296–312. [CrossRef]
[33] Qu D, Luo W, Liu Y, Fu B, Zhou Y, Zhang F. Simulation and experimental study on the pump efficiency improvement of continuously variable transmission. Mech Mach Theory 2019;131:137–151. [CrossRef]
[34] Walvekar R, Singh A, Khalid M, Gupta T,Wong WY. Thermophysical properties of deep eutectic solvent-carbon nanotubes (des-cnt) based nanolubricant. J Therm Eng 2020;6:53–64. [CrossRef]
[35] Nesbitt B. Handbook of Pumps and Pumping: Pumping Manual International. 1st ed. Washington: Elsevier Science; 2006.
[36] Zhang YL, Zhu ZC, Dou HS, Cui BL, Li Y, Xiao JJ. A method to determine the slip factor of centrifugal pumps through experiment. Int J Turbo Jet Engines 2015;32:59–64. [CrossRef]
[37] Chen B, Song B, Tu B, Zhang Y, Li X, Li Z, et al. Effect of rotation speed and flow rate on slip factor in a centrifugal pump. Shock Vib 2021;2021:6614981. [CrossRef]
[38] Li WG. Effects of flow rate and viscosity on slip factor of centrifugal pump handling viscous oils. Int J Rotating Mach 2013;2013:317473. [CrossRef]