UDC 504
DOI: 10.15587/1729-4061.2023.272674

ADVANCED DESIGN OF A SMALL-SCALE MINI GEROTOR PUMP IN A HIGH-TEMPERATURE AND HIGH-VISCOSITY FLUID THERMAL SYSTEM

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In this research, the development of a small-scale heat transfer gerotor pump with an advanced arrangement for high-viscosity and high-temperature fluids is described comprehensively. The small-scale pump design aims to meet the needs of modular components for industry and research, especially for small-scale heat transfer applications such as residential heating systems. VDI 2221 approach was used to construct an advanced gerotor pump with an internal reservoir unit that can generate additional pressure and minimize the slip factor, thereby increasing its volumetric efficiency. The developed small-scale pump was designed in a smart arrangement, which required fewer components than a typical heat transfer pump. This helps to reduce the maintenance of the pump and its components. Experimental tests were performed using a testing apparatus equipped with a heater, a control system using Pulse Width Modulation (PWM) adjustment, valves, pressure, and temperature gauges. The instruments in the apparatus test were used to control the flow rate and pump speed and monitor the temperature of the working fluid. The results of the experiment show that the advanced gerotor pump was able to deliver fluid with a viscosity of 307 ml/min and a temperature of up to 230 °C. The components arrangement minimizes the slip factor, which is mostly the main challenge of positive displacement pumps. The maximum slip coefficient of the advanced gerotor pump design is 0.095. The volumetric efficiency was in the range of 0.803–0.905 when the pump operated at 2,100 RPM and 230 °C. The experiment and analysis results show that the pump can be implemented for the actual application of the thermal system, for research and industry

Keywords: gerotor, slip coefficient, small-scale, thermal system, trochoid pump, VDI 2221

Received date 14.12.2022
Accepted date 16.02.2023
Published date 28.02.2023

How to Cite: *Zariatin, D. L., Suwandi, A. (2023). Advanced design of a small-scale mini gerotor pump in a high-temperature and high-viscosity fluid thermal system. Eastern-European Journal of Enterprise Technologies, 1 (8 (121)), 30–39. doi: https://doi.org/10.15587/1729-4061.2023.272674*

1. Introduction

Sustainable innovation is an effort to fulfill and improve the quality of human life. Based on the Sustainable Development Goals (SDGs) proclaimed by the United Nations, it can be observed. An example of the application of this innovation is technological development based on the utilization of renewable energy and an increase in the scope of technological applications, which were originally centralized to become modular [1]. The application of technologies used to reactivate old oil well systems [2] and improved wind turbine models [3] are examples of innovations supporting energy supply. Furthermore, this development encourages small-scale renewable energy and creative technologies that utilize alternative fuels, such as biomass and thermal energy storage, which require special adjustments to be accepted by society [4]. Thus, the role of researchers and industry practitioners is crucial in this technology diffusion.

The trend of small-scale technology utilization in the energy sector encourages the fulfillment of needs related to system equipment. For example, the use of solar energy, focused initially on electricity production, has shifted to many applications, such as industrial thermal generation, solar dryers for the agricultural sector, residential water and space heating [5]. The development of thermal energy storage technology allows for better economic value from solar thermal systems, reducing dependence on fossil fuels, which can then reduce pollution from the combustion process [6]. This technology can be developed for passive systems, such as energy materials for walls, floors and ceilings, and active systems, with a more comprehensive range of applications, especially for solar heating systems. The active system uses a coiled heat exchanger as a storage container and allows more energy to be stored with better performance [7].

Recent developments focus on the sustainability aspects of solar heating system technology. The application of composite materials for thermal energy storage allows for more prolonged use [8]. Using a parabolic-shaped modular solar collector allows for a better concentration of solar heat to achieve a more optimal working temperature. This includes adjusting the working fluid flow model to operate longer on the convective solar heating system. The arrangement between supply and demand in the convective solar heating system gets special attention to make it possible for small-scale applications [9]. Fig. 1 shows a general system of an active heating system that implements two working fluid models.

The arrangement has drawbacks related to the number of parts used in the systems, including using two different types of fluids, which requires two different pumps [10]. The arrangement causes the system to be wasteful, cannot be used on a small scale, and could be more economical. The best arrangement alternative is to use the direct system (Fig. 2) proposed by [11]. The smart arrangement allows for the use of one type of pump utilizing the same type of HTF, reducing the need for components, and enabling lower costs. The system is also more efficient, so the system cost is lower than that of the standard arrangement model. The main component of the system (Fig. 2) is a smart pump that must work at high temperatures.

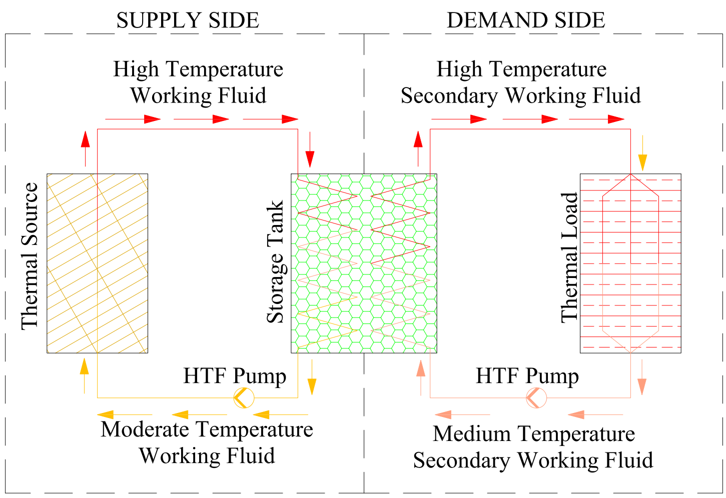

Fig. 1. Common arrangement for an active heating system

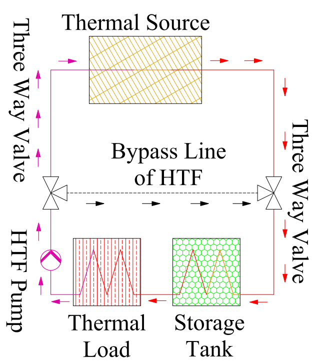

Fig. 2. Smart arrangement for an active heating system

The smart arrangement also has an optimal function to support the use of biomass-powered water heaters through the indirect burner scheme [12]. This model provides better energy utilization and reduces the need for primary energy for heating needs. The main problem is related to the small-scale pump model that needs to be produced to support the utility of the entire operation. It is still closely related to the growing trend related to the use of alternative energy sources and efforts to increase the efficiency of existing systems. Therefore, research related to the design and model of pumps for high-temperature operating purposes is still very relevant and continues to develop.

2. Literature review and problem statement

The smart pump design must cognize the working fluid's work cycle, type and temperature range. As discussed in [13], the determination of the duty cycle, type and operating temperature must be considered a fundamental aspect in determining the type of pump. More specifically, the working temperature is the most critical aspect because it relates to the components used by the pump. As in [14], the use of an airlift pump can be adapted for systems that work at high temperatures and use hazardous working fluids. However, the problem with this model is that the operation is more challenging, and the cost is relatively higher. The pump in the heating system works continuously, so its durability must be considered. The study [15] clearly shows the importance of durability in the operation of pumping systems, especially those related to high-temperature fluctuations and using incompressible fluids.

General trends point to the use of nanofluids in working fluids. The study [16] shows that using nanofluid contributes to energy exchange and provides better performance. However, more detailed pump types still need to be presented because the use of nanofluid itself requires special pump operation. The positive displacement pump model can ideally work in high-temperature conditions, has good reliability, and allows for operation on a small scale [17]. The trochoidal pump model is the best example to fulfill the functional criteria of various positive displacement pump models, as explained in [18]. The trochoidal pump, often referred to as gerotor pump, has been widely used as a vehicle oil pump, which has proven reliable and allows for small-scale use. Gerotor has only one moving component, so it is easy to operate and can be produced cheaply [19].

The application of the gerotor itself has its own challenges in adjusting the tooth shape and housing design. The numerical approach presented in the work shows the influence of the shape of the rotor and housing on the pump performance [20]. However, the actual operation requires a more in-depth study to look at pump operation more broadly, particularly concerning feasibility [21].

A more comprehensive approach is presented in [22], which provides three options for developing the gerotor design as a transfer fluid pump. Various approaches can be presented to define a gerotor model that can be used for the work function of HTF shifters. The theoretical basis presented in [23] provides an essential understanding of the gerotor pump design process. The challenge is to integrate a gerotor model that can work at high temperatures and has good flexibility in usage, including the possibility of heating on the pump side for the needs of experimental tests, which are much needed in the process of investigating the performance of small thermal systems.

As explained, the gerotor pump has ideal characteristics to be operated on a small scale. This model makes it possible to work at high temperatures, use a variety of fluid types, and use nanofluids. Thus, an actual design model of a gerotor pump that can operate based on these criteria is required. It functions as a pump for small-scale heat transfer fluids, is reliable in small-scale heating systems, can be well improved and is suitable for modular application.

3. The aim and objectives of the study

The main objective of this study is to develop a mini gerotor pump model designed for hot fluid transfer. This will assist researchers and industry players in developing smart pumps that can be used as HTF transfer tools with high reliability, easy operation, and affordability in the design process. Furthermore, using advanced mini pumps can also aid the smart heating system development process through experimental studies, especially in developing countries with limited access to research equipment.

To achieve the goal, the study was conducted in steps as follows:

– to determine the prototype model for mini gerotor pumps with high-temperature and high-viscosity fluids;

– to determine the pumping capacity in several temperature ranges;

– to estimate the slip factor based on pumping velocity;

– to estimate the volumetric efficiency based on the pump velocity and working temperature.

4. Materials and methods

This study uses a design process with stages referring to VDI 2221 [24], which is intended to be related to the design and determination of system components. Furthermore, the method of first-principal design is used to determine the function and capability of pumps specifically used for thermal systems [25]. Basically, this type of gerotor pump belongs to the category of positive displacement pumps with the primary ability to use transfer fluids with high viscosity (thick fluid) and operating temperatures [26]. The fluid transferred for the needs of the thermal system can have high viscosity and the ability to cover high temperatures so that it can withstand the decomposition process during operation. The gerotor pump has two main components: a rotor unit and a housing. The advantage of this pump model is that the rotor functions as the only moving component, so the driver model's determination tends to be flexible and can be adapted to the design plan and existing functional operations [27]. Considering these aspects, the design process can be more specific and measurable, as shown in Fig. 3.

Based on Fig. 3, at the end of the design step, special consideration is given to the features of the pump function. This aspect is the primary key in determining a pump model that can operate within the working limits determined according to its designation. The hydrostatic pressure aspect is specifically related to the capacity on the reservoir side, which will affect the value of the slip coefficient. The slip coefficient is a critical parameter that must be considered in the pump design process because there is a tendency for high losses due to the backflow of the pumped liquid and re-entering the intake port.

Volumetric efficiency is a value that determines the pump's overall performance related to the effectiveness of the pump operating to transfer fluid. Therefore, hydrostatic pressure, slip coefficient and volumetric efficiency have a strong relationship and must be considered and tested based on the design results obtained.

Higher hydrostatic pressure values can reduce the risk of slip [28]. Thus, determining the higher hydrostatic pressure value will directly reduce the value of the slip coefficient. With a lower slip coefficient value, the ability of the pump to transfer fluid becomes higher so that the optimal volumetric efficiency value of the pump can be maintained. To achieve this function, the driver model must be from outside the pump (indirect driver) to achieve a higher hydrostatic pressure function.

The indirect driver model is a better option because it allows for easier maintenance, and the rotation control function can be performed more flexibly.

Higher hydrostatic pressure values on the inlet side can be made by using the internal reservoir side before the gerotor unit (Fig. 4). It accommodates a certain amount of working fluid and provides a better hydrostatic value, reducing the chance of a slip coefficient.

Placement of the reservoir on the inlet side makes it possible to provide a heater whose role is to determine the operating fluid working temperature at the research stage and maintain optimal operation of the pump.

Based on these considerations, the advanced gerotor pump model must meet all these criteria and achieve optimal function.

The pumps developed from the design process are then tested to determine their performance and operating characteristics. This test is intended explicitly for delivery capability parameters, slip coefficient values and volumetric efficiency. For these parameter values to be obtained, the testing model must be able to measure performance measurement variables. Fig. 4 shows the apparatus design used with the specifications for the type of fluid used in Table 1.

The main fluid source is on the side of the reservoir where the Pulse Width Modulation (PWM) and heater are placed. These two components are responsible for heating the working fluid at a certain point for performance measurement purposes. The temperatures used are 30 °C, 80 °C, 130 °C, 180 °C and 230 °C. The flow rate must be well regulated. Thus, the inflow from the reservoir and the outflow from the pump assembly must be properly maintained. A needle valve is placed at the reservoir outlet to regulate the flow rate in the pump assembly. The flow from the pump is regulated using a ball valve with a constant outlet pressure of 1.5 bar.

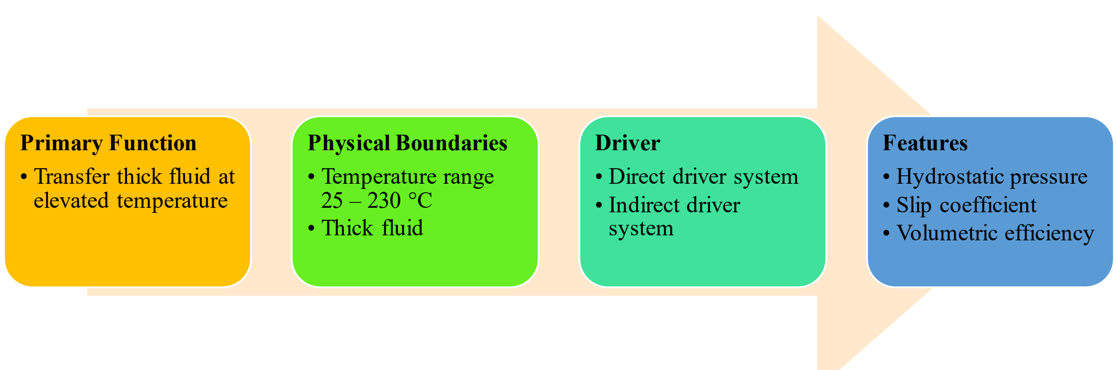

Fig. 3. Basic considerations for designing a high-temperature gerotor pump

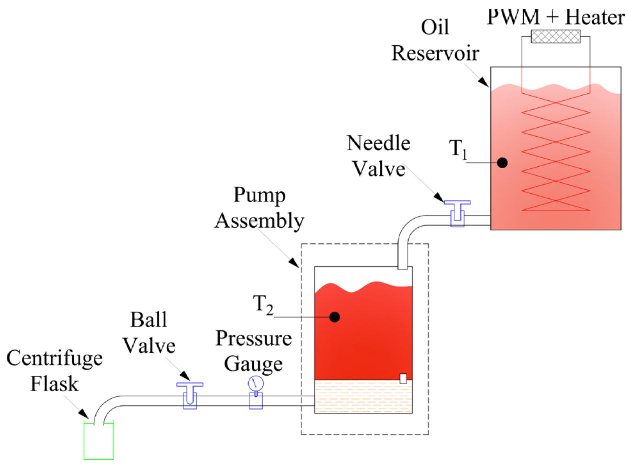

Fig. 4. Testing apparatus for evaluating the pumping performance

Table 1

Basic properties of the working fluid

Description	Unit	Type 1	Type 2
Name	–	AT400	HTMHVIS
Boiling point	°C	255	400
Viscosity (at 40 °C)	cSt	2.4	97
Density	g·cm^{-3}	1.068	0.885

The operating speed of the pump is set at a specific speed between 150–2100 RPM (150 RPM interval) with the help of PWM on the motor driver's side. The temperature values from the reservoir side and the pump assembly are viewed using a thermocouple (type K) at the measurement points T1 and T2. Two types of fluids with high and low viscosity values are used to see the ability of the pump to operate in different types of fluids. The output value of the pump is measured using a centrifuge flask (ASTM 96D) so that the volume per unit of time can be obtained more precisely.

5. Results of design and experiment of advanced gerotor pump

5. 1. Results of the prototype model for mini gerotor pump for high-temperature and high-viscosity fluid

Fig. 3 shows the primary consideration of the design process of the advanced mini generator pump. Since target fluids used tend to increase in temperature and viscosity, the use of an external mover is a critical choice. Fig. 5, *a* shows the main part of the chosen design in this study. The placement of two bearings on the top and bottom of the design ensures a proper rotation. The upper body also serves as an internal reservoir to maintain the hydrostatic pressure during pump operation. The reservoir has an internal capacity of 280 ml. The applied gerotor consists of eight teeth inner gear and seven teeth outer gear (rotor). The rotor is the only component that rotates at the point of eccentricity. Therefore, an attached shaft is connected to a pulley to move the rotor. Movement between the rotor and inner gear pushes the fluid out of the outlet port. A gasket is installed between the upper and lower bodies to prevent leaks.

All components are assembled and tested for leak resistance. Then, a high-torque DC motor is installed as the prime mover. The DC motor and the pump are connected with a belt drive (Fig. 5, *c*). The DC motor was used as an external mover to control the process using Pulse Width Modulation (PWM). Furthermore, it guarantees operational function at high temperatures and prevents the motor unit from overheating. The main advantages offered by this design are its compactness and simple assembly, high capability to operate at high temperatures, and use of viscous fluids.

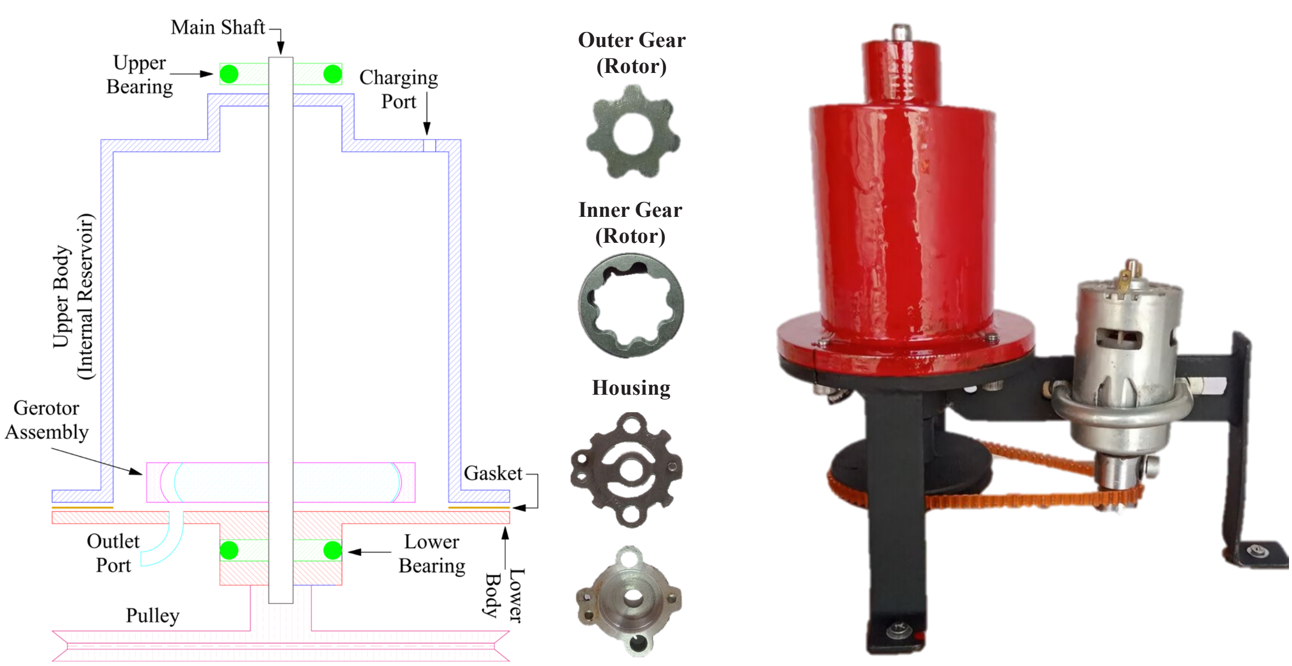

Fig. 5. Advanced gerotor pump: *a* — cutting view of the components arrangement; *b* — gerotor set; *c* — assembly of the designed mini gerotor pump

5. 2. Results of pumping capacity under different operating temperatures

Fig. 6 presents experimental test data for pumping capacity (pumping delivery) using working fluids of type 1 with the lowest viscosity. An increase in pump velocity causes an increase in pump capacity. It occurs as the increase of fluid is moved per unit of time by the pump. An increase in temperature also shows a better capacity.

At high temperatures (180 and 230 °C), higher capacity can be obtained at a lower speed; the maximum capacity is 237 ml/minute at 30 °C with a speed of 1,650 RPM. At the highest temperature, maximum capacity can be obtained at a value of 307 ml/minute with an even lower speed of 1,200 RPM. The maximum value demonstrates a pattern of a stall point when a decrease in pump capacity occurs despite the increase in velocity. A pumping delivery test is also conducted for high viscous fluid (type 2). Fig. 7 shows similar patterns caused by increased RPM, which results in higher capacity. At low temperatures (30 and 80 °C), the pump capacity tends to drop with an acquired maximum value of 176 and 192 ml/minute.

The main difference between type 1 and type 2 fluids is that type 2 fluid has no stall point at a low temperature. For type 2 fluid, stall points are only identified at a temperature of 130, 180, and 230 °C. Then, stall point values are spotted lower than for type 1 fluid. Stall point values are up to the limitation of 188, 212, and 233 ml/minute for type 2 fluid.

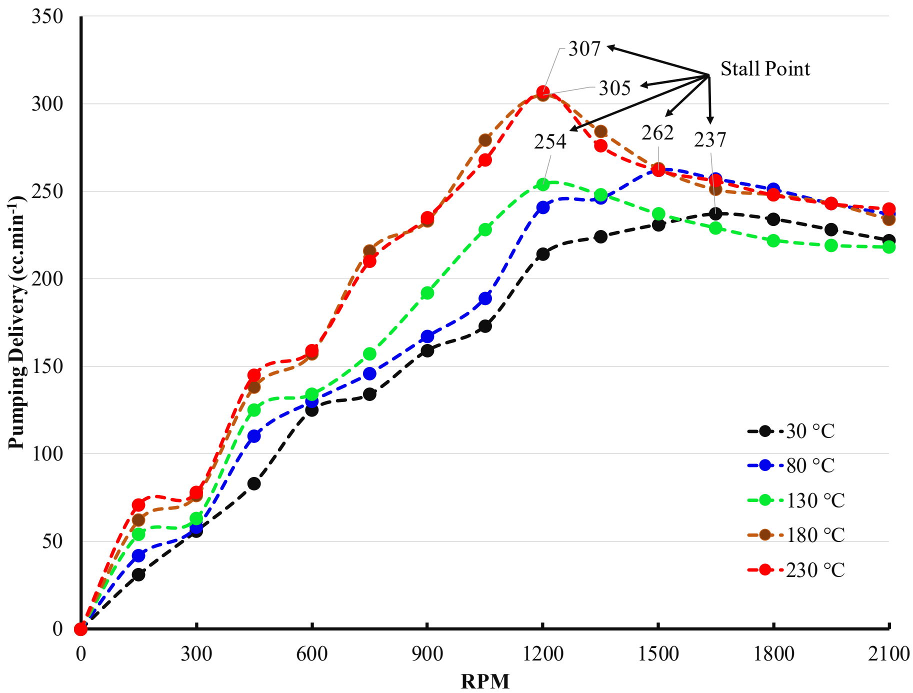

Fig. 6. Pumping capacity vs RPM under different temperatures for low viscous fluid (type 1)

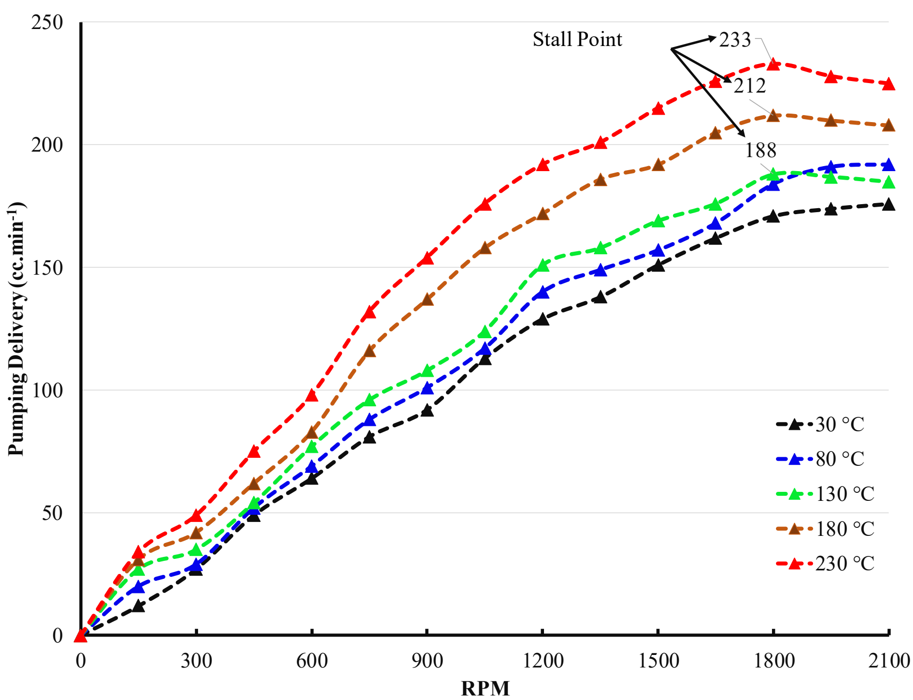

Fig. 7. Pumping capacity vs RPM under different temperatures for high viscous fluid (type 2)

Therefore, the pump can show performance for fluid types with high temperature and viscosity properties.

5. 3. Results of slip coefficient estimation under different operating temperatures

The slip factor is a common issue in hydrodynamic pumps. As shown in Fig. 8, the increase in slip coefficient is related to the change in the pump velocity and the temperature rise. Higher values of temperature trigger an increase in a poor slip coefficient.

At 30 °C, the highest slip value is obtained at 0.223 at a temperature of 230 °C, reaching 0.357. The pump velocity value also influences the increase in slip value. The increase in the pump velocity exacerbates the slip coefficient value. Variations in the types of fluid used still affect the slip coefficient of the pump and are considered natural. The main difference is that the slip coefficient value on type 2 fluid is typically lower.

In Fig. 9, the maximum slip coefficient value shown at a temperature of 30 °C is 0.127, whereas the following slip coefficient value increase tends to be lower.

The highest slip value of 0.197 is obtained at 230 °C. The use of thick fluid reduces the slip coefficient risk due to its ability to transfer high temperature and viscosity fluids under high pressure and temperature. The pump continues to function as a fluid transfer.

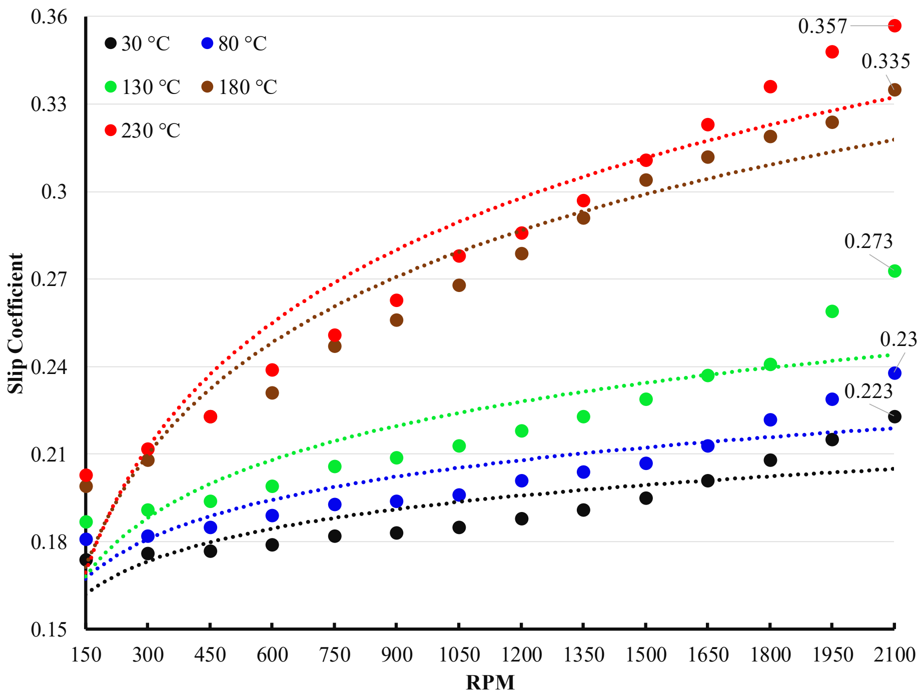

Fig. 8. Slip coefficient vs RPM under different temperatures for low viscous fluid (type 1)

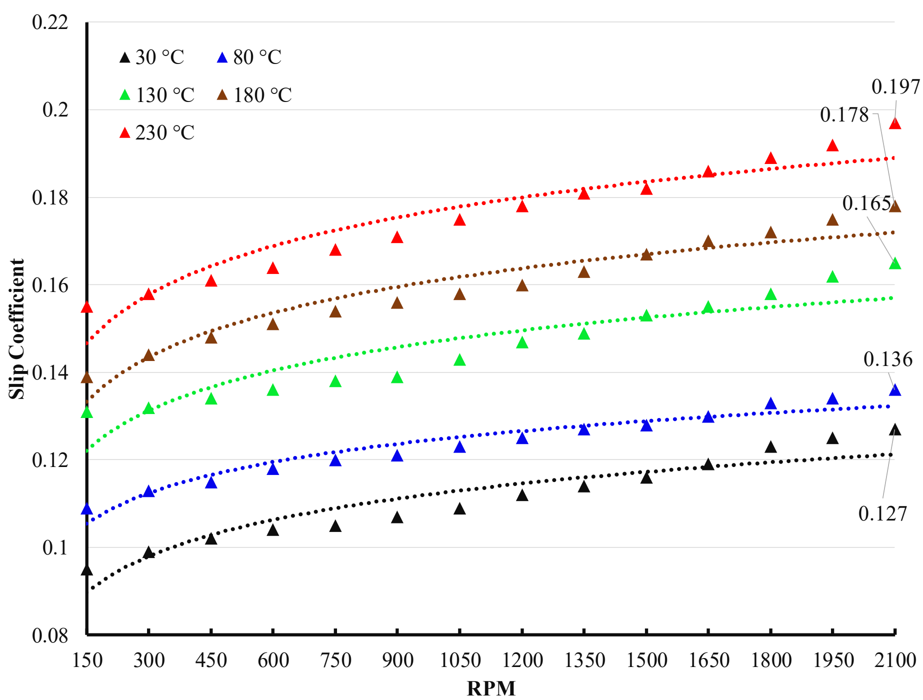

Fig. 9. Slip coefficient vs RPM under different temperatures for high viscous fluid (type 2)

5. 4. Results of volumetric efficiency under different pumping speeds and temperatures

Another critical factor of a pump system is volumetric efficiency, which becomes the primary key related to pump performance. Fig. 10 shows a comparison of volumetric efficiency at different fluid temperatures and pump velocities. Pump volumetric efficiency tends to decrease at higher temperatures, which are 180 and 230 °C.

There is also a spike in the value of decreasing volumetric efficiency between the temperatures of 130 and 180 °C. For example, the lowest volumetric efficiency at 130 °C is only 0.727, and 0.665 at 180 °C. This indicates a limitation in the pump operating temperature, thus causing significant changes. The increasing pattern at 180 and 230 °C also possess similarities in low speeds (between 150 and 750 RPM). The high-viscosity fluid still reveals a decrease in volumetric efficiency (Fig. 11).

The lowest value for thick fluid is 0.803, obtained at 230 °C with a pump velocity of 2,100 RPM. However, the decrease in volumetric efficiency for each temperature level is not as severe as in the case of thin liquids (type 1). There is a slightly significant change during the temperature increase between 80 °C and 130 °C. At 80 °C, the volumetric efficiency value can reach 0.864 with a speed of 2,100 RPM. At the same speed but with a temperature of 130 °C, the volumetric efficiency value drops to 0.835. This value is far superior to thin fluids that can only achieve 0.727 efficiencies under the same working conditions.

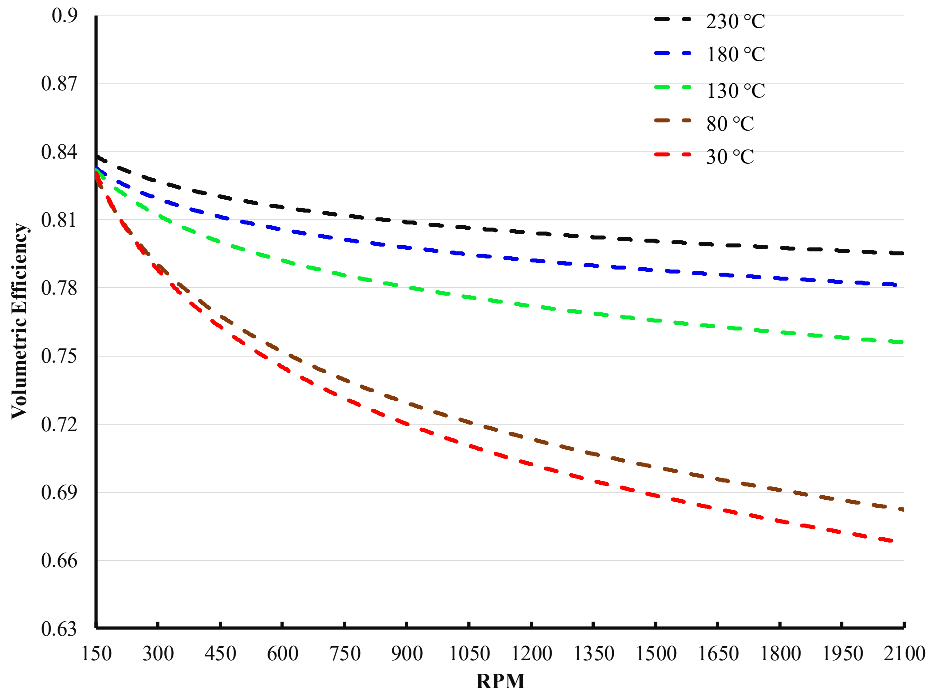

Fig. 10. Volumetric efficiency vs RPM under different temperatures for low viscous fluid (type 1)

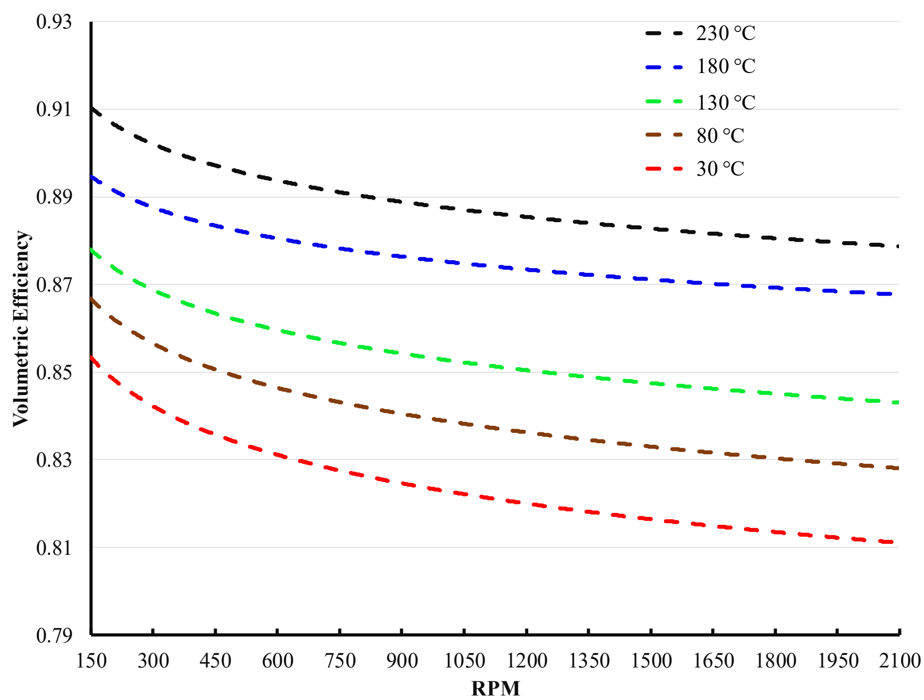

Fig. 11. Volumetric efficiency vs RPM under different temperatures for high viscous fluid (type 2)

6. Discussion of the results from the design and experiment of the advanced gerotor pump

The unique characteristics of the gerotor as a positive displacement pump make the pumping operation more reliable where extreme operations using high temperature and high viscous fluids can be carried out. This study found an advanced design model to determine the design of the small-scale gerotor pump presented in Fig. 5. A more extensive internal reservoir on the inlet side can maintain the pressure difference between the input and output sides of the pump. It allows the pump to operate more efficiently due to hydrostatic pressure, which reduces the slip coefficient. Moreover, with the placement of a more extensive reservoir, it is possible to heat fluids with the help of a heater, which can be used for industrial and research purposes. Furthermore, using indirect drivers assisted by drive motors outside the assembly makes controlling pump operations easier.

The increase in working fluid temperature causes a change in the operating characteristics of the pump. This can be seen clearly in Fig. 6, 7, where the unit pumping volume becomes higher as the temperature increases. The increase in volume is caused by changes in the value of fluid viscosity because, at higher temperatures, the level of fluid viscosity tends to decrease. Decreasing the level of viscosity causes the fluid to be pumped more easily. So that the pumping capacity becomes more significant, especially at higher pump speeds; this can also be seen more clearly in Fig. 7, where thick fluid is used for higher temperatures and shows a more significant increase in value. Even though there is a relationship of increasing capacity and temperature and pump speed, the risk of a stall point still occurs at a certain point. Thinner liquids can cause a stall point. The stall point occurs at a speed of 1,200–1,800 RPM for thin fluids (Fig. 6), while for thick fluids, it occurs at a speed range of 1,800 RPM (Fig. 7). After passing through the stall point, the pumping capacity tends to decrease.

The stall point directly impacts the slip coefficient, which provides clearer pump operating limits. As shown in Fig. 8, the slip coefficient increases with the increasing temperature of the working fluid. This condition is also strongly influenced by the type of fluid used, where fluids with low viscosity have a greater chance of internal leakage and an increase in the value of the slip coefficient. The slip coefficient value for thin fluid (Fig. 8) varies between 0.223–0.357, where the lowest value is at a temperature of 30 °C, with the highest slip occurring at 230 °C. Using thick fluid can reduce the value of the slip coefficient (Fig. 9). A more viscous fluid has a better density value and can minimize internal leakage. The slip coefficient value for thick fluid varies between 0.127–0.197. Thick fluid also has a lower stall point value (Fig. 7), which only occurs at high temperatures (>100 °C). It is of particular note because the stall point for thin fluids occurs at all temperatures (Fig. 6). This condition reinforces the primary role of the ideal gerotor for pumping fluids with high viscosity and temperature where maximum pump performance can be achieved with a low slip coefficient value in operation.

The influence of the slip coefficient is directly related to volumetric efficiency. Both types of fluids show a decrease in volumetric efficiency and an increase in the operating speed and working temperature of the fluid used. However, using an internal reservoir on the inlet side reduces the slip effect so that the volumetric efficiency value can be maintained better. This condition can be seen clearly in Fig. 10, where a thin fluid with a low pump temperature still has an excellent volumetric efficiency value (with a speed of 150–450 RPM at a temperature of 30–130 °C). The most significant advantage is obtained from the use of thick fluid, where the volumetric efficiency value of the pump is well maintained, with an overall value above 0.8 in all speed and temperature ranges used (Fig. 11). High volumetric efficiency will maintain better pump performance and reduce the power used by the pump so that the energy balance used in the system is more optimum. Therefore, the advanced gerotor pump model offered in this study can still maintain ideal performance and operating characteristics for thick fluids in high-temperature conditions, making it ideal for use in HTF functions.

The performance evaluation indicates a good performance that can be delivered by the proposed design of the advanced gerotor pump. The low pumping loss under specific operation parameters is desirable for maintaining sufficient performance of the system. This agrees well with the work [29], which demonstrates that the thermal recycling method can improve the overall operation of the thermal system. Therefore, higher efficiency combined with thermal recycling increases the overall system performance significantly. Furthermore, the integration of the system with a heat pump as presented in [30] demonstrates that the thermal performance of the system is increased along with a decrease in the power consumption of the pumping system. Thus, a high efficiency and smaller model are advantageous for delivering a significant positive impact on the thermal system, which utilizes heat transfer fluid, particularly for a small-scale thermal system.

Even though the advanced gerotor pump's function is maintained optimally, the model offers obvious drawbacks. The use of an internal reservoir causes the need for an assist pump to maintain the mass flow rate balance of the working fluid. This includes the use of a belt connection, which has the risk of slipping between the driver and the pulley. This needs to be eliminated in subsequent studies by using a gear driver and a vacuum pump to maintain the mass flow rate balance of the working fluid. This study still has limitations; the presented advanced gerotor has a relatively lower pressure, so it is not suitable for high flow rates. So, there is still room for improvement to meet the needs of high-speed pumps, especially for applications involving small tubes.

7. Conclusions

1. The designed advanced gerotor pump has an internal reservoir of 280 ml, with a rotor gear ratio of 7/8 (inner/outer gear), using a belt drive connected to a DC motor that can easily control the pumping speed and the simplified assembly between the upper and lower bodies is leak-proof, which ensures the operation of the pump.

2. The pumping capacity depends on the speed and temperature of the pump, where low viscous fluids capacity can reach 307 ml/min with 1,200 RPM speed at a temperature of 230 °C. However, high viscous fluids capacity can reach 233 ml/min with 1,800 RPM speed at the same temperature.

3. The slip factor varies between 0.174–0.357 for thin fluid and 0.095–0.197 for thick fluid. Slip factor charac-

teristic increases along with temperature and pump velocity escalation.

4. The highest volumetric efficiency is up to 0.905–0.803 for thick fluid and 0.826–0.643 for thin fluid, where the highest efficiency value is acquired at the lowest speed and temperature.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

Financing

The study was performed without financial support.

Data availability

Data will be made available on reasonable request.

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