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# External Gear Pump Varying Temperature Investigation

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**Abstract**

Gear pumps are most common types of pumps having excessive heat during pumping process. In this research, effects of varying outlet temperature are investigated on external gear pump in diesel oil example. For temperature values ofexternal gear pump theoretical approaches are analysed and compared to practical results to identify the factors and parameters affecting it. Positive and negative outcomes of excessive and less temperature values on performance of gear pump are investigated depending on requirements of application type.

**Keywords:** External gear pump excessive heating ; Turbulency in rising temperature; Laminarity in rising temperature; Effects of variation of temperature in external gear pump ; Difference of external gear pump temperature depending factors; Theoretical approach to parameters of heated pump.

## THEORETICAL APPROACH TO TEMPERATURE DIFFERENCE FORMULA

Initially we need to get theoretical formula for temperature of external gear pumpindicating relationships with other parameters. The conservative approach to the thermal design of a system is to consider that all the pump input power is converted to heat by the various system componentsand must eventually be rejected as heat. In essence this is a restatement of the First Law of Thermodynamics which states that in a closed system the net work input is proportional to the net heat output. The pump introduces a given amount of mechanical energy into the system, and this energy isused to provide useful work such as turning a shaft or is dissipated in the form of thermal energy at various stations in the system.

First Law states:

$$\dot{Q} - \dot{W} = \Delta\dot{E} \qquad (1)$$

$\dot{W}$ - work input during process

$\dot{Q}$ - heat output during the process

$\Delta\dot{E}$ - rate of change of the system energy.

$$\left[\begin{array}{l}\text{Net amount of energy}\\ \text{to system as heat or}\\ \text{work for a given}\\ \text{time interval}\end{array}\right] + \left[\begin{array}{l}\text{Stored energy of}\\ \text{mass entering}\\ \text{system for the}\\ \text{same time inter-}\\ \text{val}\end{array}\right] - \left[\begin{array}{l}\text{Stored energy}\\ \text{of mass leaving}\\ \text{system for the}\\ \text{same time in-}\\ \text{terval}\end{array}\right] = 0$$

Written symbolically with formula:

$$\dot{Q} - \dot{W} + \dot{m}\, p_1 v_1 - \dot{m}\, p_2 v_2 \ + \ \dot{E}_1 - \dot{E}_2 \qquad (2)$$

$\dot{Q}$ - net amount of heat added to the system for a given time interval

$\dot{W}$ - net amount of work, excluding flow work, done by system for the sametime interval

$\dot{m}\, p_1 v_1$ - amount of flow work on the system by the enteringfor the interval

$\dot{m}\, p_2 v_2$ - amount of flow work done by system or fluid leaving

$\dot{m}$ - mass rate of fluid flow

$p$ -flow pressure

$v$ - specific volume

$\dot{E}_1$ - stored energy rate of fluid entering the system

$\dot{E}_2$ -stored energy rate of fluid leaving the system

The energy term neglecting the effects of electricity,magnetism, and surface tension may be written as:

$$\dot{E} = \dot{m}\left(u + \frac{V^2}{2g_c} + \frac{g}{g_c} z\right) \qquad (3)$$

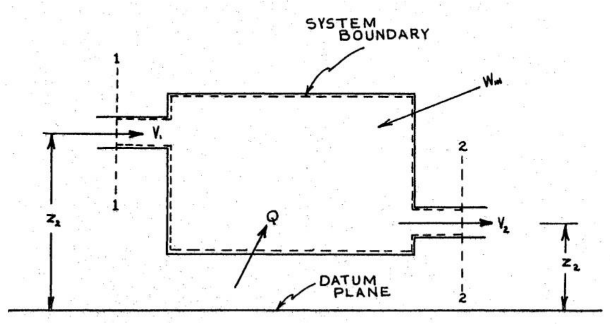
$u$- internal energy

$V$- average velocity of the fluid

Z- elevation above an arbitrary datum

g -gravitational acceleration.

Figure 1




Substituting Equation 3 into Equation 2, and taking other parameters into consideration:

$$\dot{Q} + \dot{W}_{in} = \dot{m}(u_2 - u_1) + \dot{m}(p_2 v_2 - p_1 v_1) + \dot{m}\frac{(V_2^2 - V_1^2)}{2g_c} + \dot{m}(z_2 - z_1)\frac{g}{g_c} \quad (4)$$

In the thermal analysis of the pump the kinetic and potential energy terms are negligible and, therefore, are canceled. Also, it is approximated that adiabatic conditions prevail. By recalling that

$$h=u+pv \quad (5)$$

h- enthalpy

remembering the previous approximations, Equation 4 can be written as

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}(u_2 - u_1) + \dot{m}(p_2 v_2 - p_1 v_1) \quad (6)$$

The enthalpy for the liquid varies as a function of both pressure and temperature; therefore, the change in enthalpy in equation 6 can be expressed as

$$\Delta h = \Delta h_p + \Delta h_t \quad (7)$$

$\Delta h_p$ - enthalpy change at constant pressure.

$\Delta h_t$ enthalpy change at constant temperature.

The specific heat at constant pressure is defined as

$$c_p = \left(\frac{dh}{dt}\right)_p \quad (8)$$

Therefore, it can be stated that

$(h_2 - h_1)_p = \int c_p \, dt = c_p \ (t_2 - t_1)$ (9)

Where

$c_p$ is considered to remain constant with respect to temperature.

In the determination of the change in enthalpy at constant temperature, the relation

$$\left(\frac{dh}{dp}\right)_t = v - t\left(\frac{dv}{dt}\right)_p \qquad (10)$$

Integration with respect to pressure produces

$$(h_2 - h_1)_t = v(p_2 - p_1) - t\left(\frac{dv}{dt}\right)_p (p_2 - p_1) \qquad (11)$$

The $t\left(\frac{dv}{dt}\right)_p (p_2 - p_1)$ term is small and can be neglected

Therefore, the power input can be expressed as:

$$\dot{W}_{in} = \dot{m}\, c_p (t_2 - t_1) + \dot{m}v(p_2 - p_1) \qquad (12)$$

Analysis of the energy equation states that the pump work done on a mass of incompressible fluid in a reversible and adiabatic process during a given time interval is

$$\dot{W}_{in} = \dot{m}v(p_2 - p_1) \qquad (13)$$

Again, the kinetic and potential energy terms are neglected. The power input denoted by Equation 13 is the ideal amount required. If the overall pump efficiency is represented by η0, the actual power input to the system is

$$\dot{W}_{in} = \frac{\dot{m}v(p_2 - p_1)}{\eta_0} \qquad (14)$$

Substituting the above relationship into equation 12 and solving for temperature change across the pump results:

$$t_2 - t_1 = \frac{v(p_2 - p_1)}{c_p} \left(\frac{1 - \eta_0}{\eta_0}\right) \qquad (15)$$

In terms of specific gravity, equation 15 becomes:

$$t_2 - t_1 = \frac{(p2 - p1)}{\rho\, c_p} \left(\frac{1 - \eta 0}{\eta_0}\right) \qquad (16)$$

## ANALYSIS OF MEASURED DATA

To check relationships between parameters in equation 16 experimental result are needed on external gear pump.

### Experimental test bench and simulation model

The test bench consists of tank, external gear pump, electric motor, hose, pressure relief valve, variable orifice, cooler, flow meter, pressure and temperature transducers and other accessories for connections. The specifications of the system are explained in detail below.

### System description

The system is a simple model as shown in figure 2 with electric motor is mounted above the tank and pump is connected to electric motor which comes inside the tank that is not visible in actual system. One end of the hose is connected to the outlet of the pump and other end to pressure relief valve which acts as load in this system. One end of the cooler is connected to pressure relief valve and other end tothe tank. Flow meter is fixed to inlet of hose to measure flow. Temperature sensor is fixed at inlet of hose to measure the temperature of the diesel oil. Signals from the sensors of the system was taken by using multimeter.

Schematic diagram of experimental test bench
1.Electric motor; 2.External Gear Pump; 3.Filter; 4,5,6.Pressure transducers; 7. Temperature sensor; 8.Hose; 9.Variable orifice; 10.Pressure relief valve; 11.Cooler; 12.Tank; 13. Flow meter.

System component specifications

Electric motor : Power 7.5 kW

Nominal speed: 1440 rpm

Hydraulic pump

| | |
|---|---|
| Type of pump | External gear pump |
| Medium | Diesel oil |
| Displacement | 35.2 cc/rev |
| Maximum pressure | 155 bar |
| Maximum speed | 2500 rpm |
| Temperature sensor | 0-100 °C / 0-10 v |

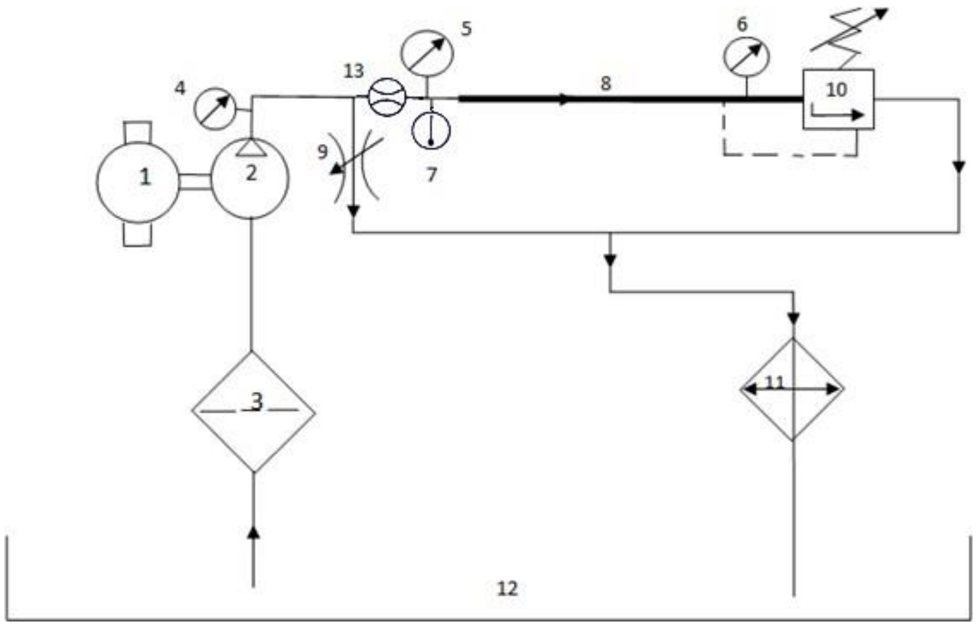
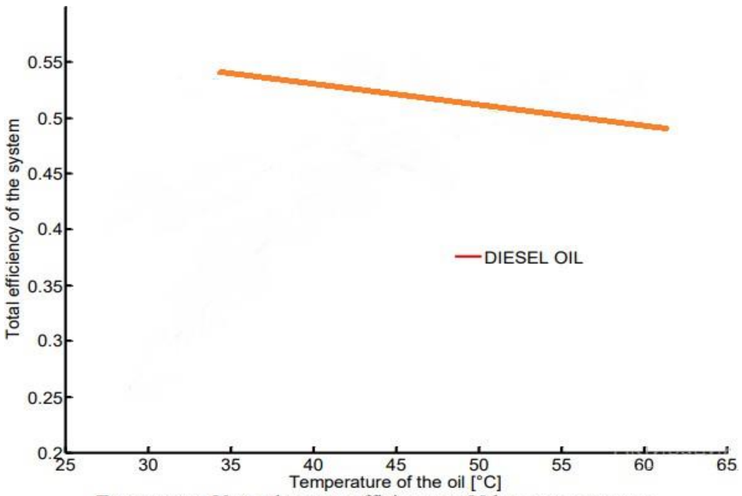


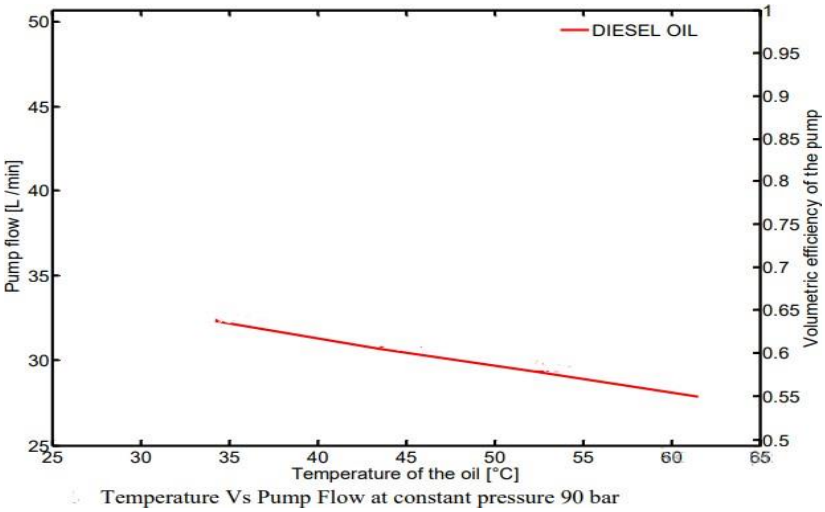

Figure 2

According to all measured values three graphs are plotted in different condition:

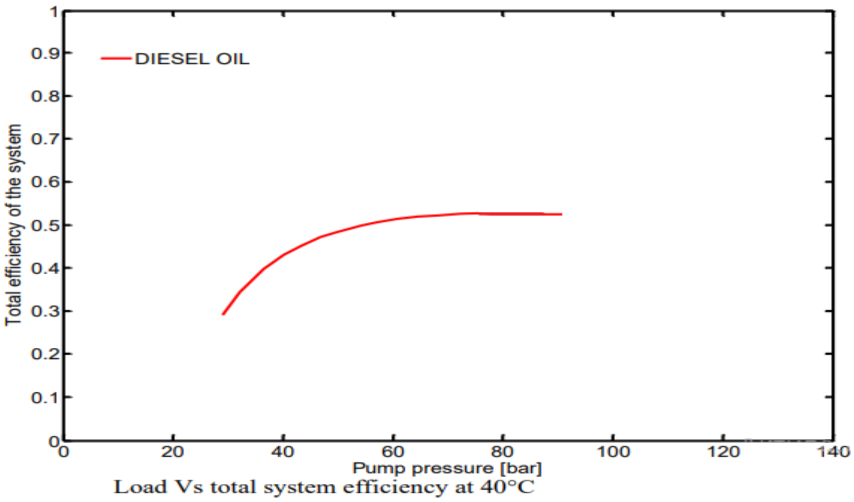



Temperature Vs total system efficiency at 90 bar pump pressure

Graph 1 P=const T vs total efficiency (inlet T and P are constant)




Graph 2 P=const T vs Q ( inlet T and P are constant)




Graph 3 T=const P vs Efficiency ( inlet T and P are constant)

As it seems from each graph theoretical equation is practically correct due toproportionality of parameters in equation 16:

- Increasing outlet temperature is the result reduced total efficiency in constant outlet pressure
- Decreasing flow rate causes more heat in outlet pipe in constant outlet pressure
- Increasing outlet pumping pressure is the result of greater total efficiency in constant outlet temperature

## IMPORTANCE OF LAMINARITY IN VARYING TEMPERATURE

As outlet temperature increases kinematic viscosity value drops down and in constant flow rate decreasing viscosity values increase Reynold number according to:

$$Re = \left(\frac{4Q}{\pi D v_{kinematic}}\right) \qquad (17)$$

D=0.01m - Pipe outlet diameter (hose diameter)

According to measured parameters through measurement devices in constant flow rate $\dot{Q}$ =25 L/min at exit pipe from 30℃ to 60℃ increasing temperature interval, kinematic viscosities are accordingly taken from tables as function of outlet pressure and outlet temperature. Along with these Reynold number is calculated for each value due to equation 17 through Engineering Equation Solver(EES) software. Kinematic viscosity values vary in [2.1 - 3.4] CentiStokes (cSt) interval with 0.1 cSt increment in each step, converted to m2/min.

EES Demonstration Version: C:\EES_DEMO\HELLO.TXT - [Parametric Table]

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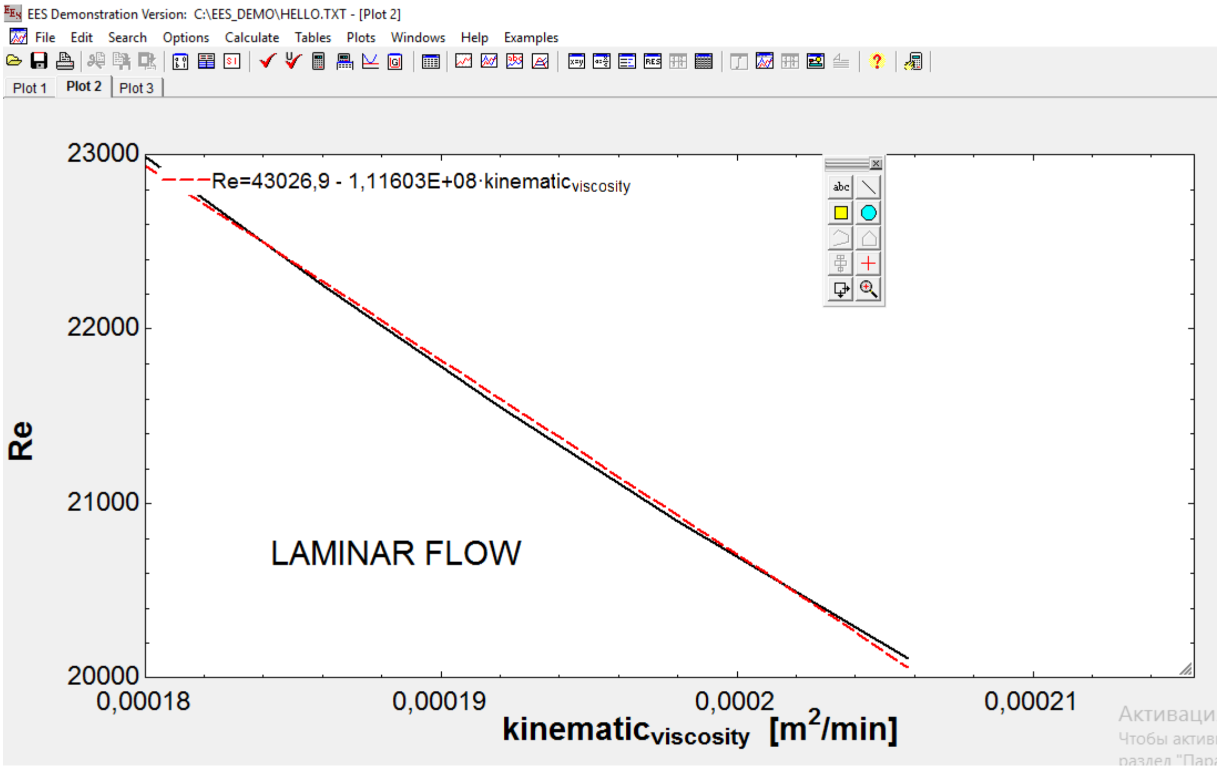
Table 1 Table 2

| 1..50 | $kinematic_{viscosity}$ [m2/min] | Re |
|---|---|---|
| Run 1 | 0,0002058 | 20107 |
| Run 2 | 0,000198 | 20899 |
| Run 3 | 0,000192 | 21552 |
| Run 4 | 0,000186 | 22247 |
| Run 5 | 0,00018 | 22989 |
| Run 6 | 0,000174 | 23782 |
| Run 7 | 0,000168 | 24631 |
| Run 8 | 0,000162 | 25543 |
| Run 9 | 0,000156 | 26526 |
| Run 10 | 0,00015 | 27587 |
| Run 11 | 0,000144 | 28736 |
| Run 12 | 0,000138 | 29986 |
| Run 13 | 0,000132 | 31349 |
| Run 14 | 0,000126 | 32841 |

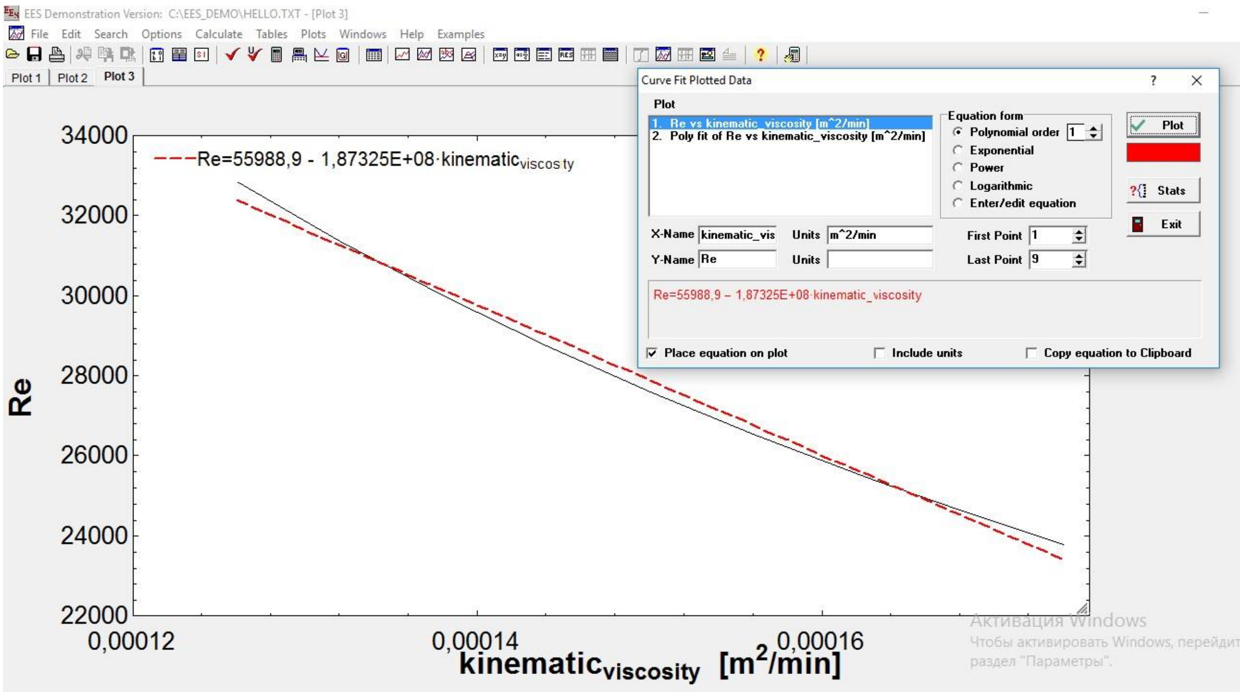
Figure 3 EES output due to eq 17

In 30°C- 60°C interval real data of kinematic values and Reynold Number calculated due to equation 17 are plotted for laminar region (Re<2300, till Run 6 )in graph 4:




Graph 4 Laminar region

At the the same condition for turbulent flow (after Run 5):




Graph 5 (Transient+Turbulent region)

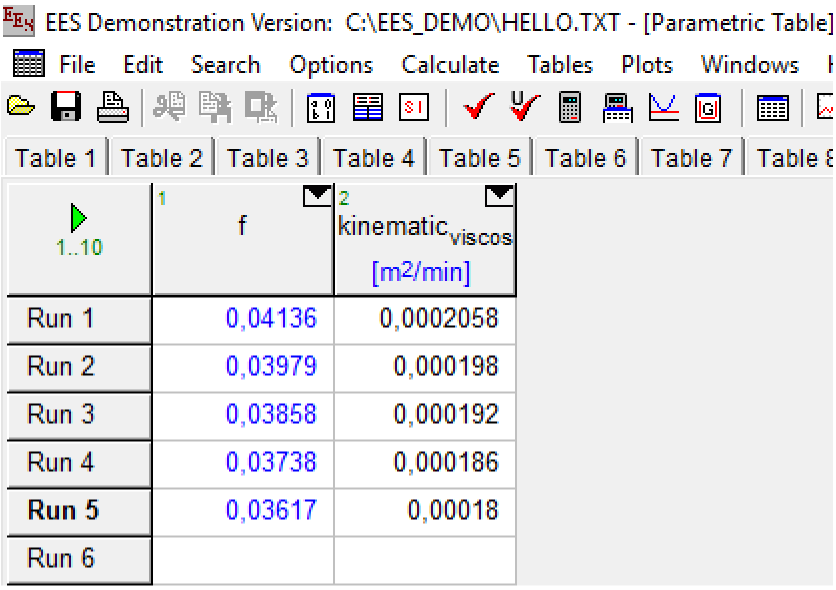
From both graph 4 and 5 it is apparent that in laminar and transient+turbulent region in increasing temperature decreasing rate of kinematic viscosities are the same due to both graph and their approximated formulas (red dashed lines formula).

## VARIATION OF DARCY–WEISBACH FRICTION FACTOR

Now changing rate of friction factor is studied in both laminar and transient+turbulent region. For Laminar flow friction factor:

$$f = Re/64 \qquad (18)$$

Calculated Reynold number values in figure 3 is next input in EES and for laminar flow EES calculates outputs due to eq (18) in Graph 6:




| 1..10 | f | kinematic$_{viscos}$ [m2/min] |
|---|---|---|
| Run 1 | 0,04136 | 0,0002058 |
| Run 2 | 0,03979 | 0,000198 |
| Run 3 | 0,03858 | 0,000192 |
| Run 4 | 0,03738 | 0,000186 |
| Run 5 | 0,03617 | 0,00018 |
| Run 6 | | |

Figure 4

For transient+turbulent region friction factor is found:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon}{3.7\,D} + \frac{2.51}{Re\sqrt{f}}\right) \qquad (19)$$

ε= 0.00015m - pipe roughness

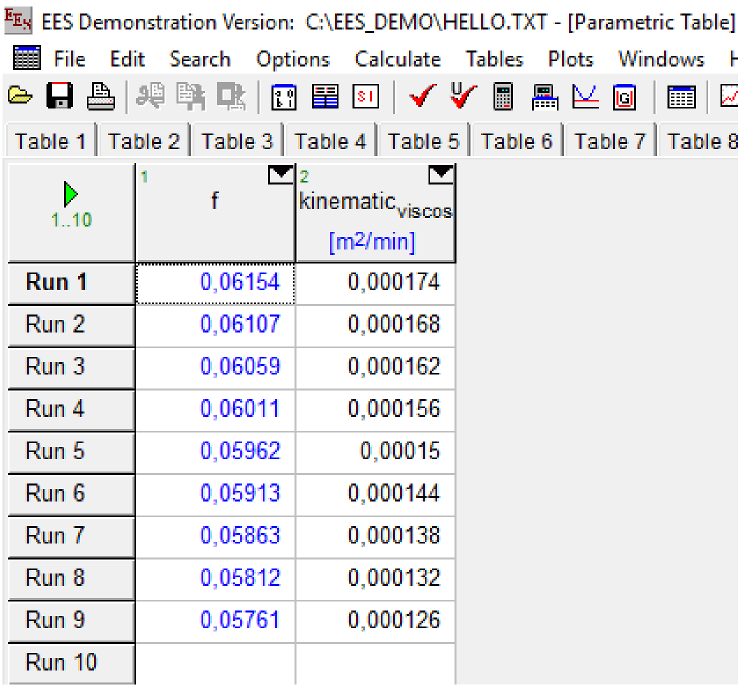
L= 15 m - selected length of outlet pipe

For calculated values of Reynold number above 2300 EES solves equations with following sequence:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.00015}{3.7*(0.01)} + \frac{2.51}{0.000174*\sqrt{f}}\right)$$

- $$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.00015}{3.7*(0.01)} + \frac{2.51}{0.000168*\sqrt{f}}\right)$$
- $$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.00015}{3.7*(0.01)} + \frac{2.51}{0.000162*\sqrt{f}}\right) \ldots$$

All outputs by EES below:




| 1..10 | f | kinematic$_{viscos}$ [m2/min] |
|---|---|---|
| Run 1 | 0,06154 | 0,000174 |
| Run 2 | 0,06107 | 0,000168 |
| Run 3 | 0,06059 | 0,000162 |
| Run 4 | 0,06011 | 0,000156 |
| Run 5 | 0,05962 | 0,00015 |
| Run 6 | 0,05913 | 0,000144 |
| Run 7 | 0,05863 | 0,000138 |
| Run 8 | 0,05812 | 0,000132 |
| Run 9 | 0,05761 | 0,000126 |
| Run 10 | | |

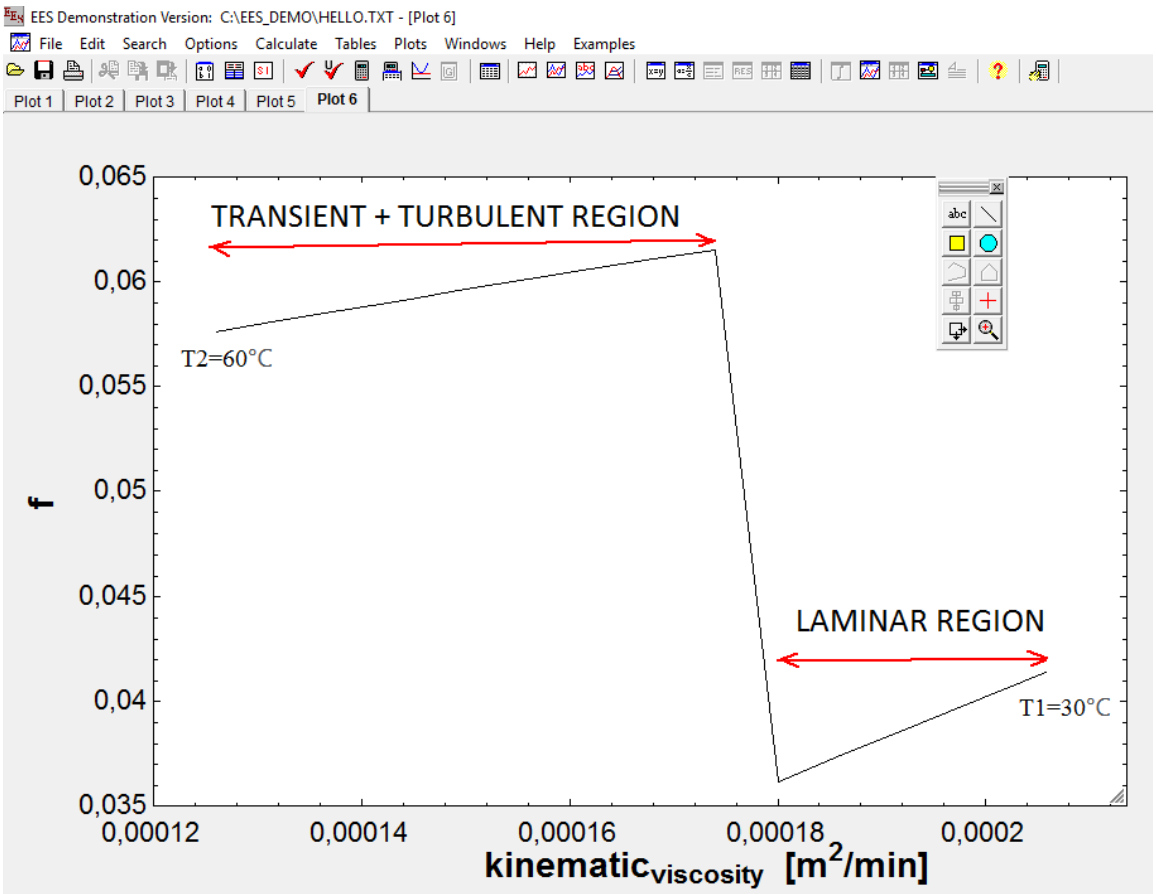
Figure 5

By collecting data for each region and plotting general graph:

EES Demonstration Version: C:\EES_DEMO\HELLO.TXT - [Parametric Table]

| 1..20 | f | kinematic$_{viscosity}$ [m2/min] |
|---|---|---|
| Run 1 | 0,04136 | 0,0002058 |
| Run 2 | 0,03979 | 0,000198 |
| Run 3 | 0,03858 | 0,000192 |
| Run 4 | 0,03738 | 0,000186 |
| Run 5 | 0,03617 | 0,00018 |
| Run 6 | 0,06154 | 0,000174 |
| Run 7 | 0,06107 | 0,000168 |
| Run 8 | 0,06059 | 0,000162 |
| Run 9 | 0,06011 | 0,000156 |
| Run 10 | 0,05962 | 0,00015 |
| Run 11 | 0,05913 | 0,000144 |
| Run 12 | 0,05863 | 0,000138 |
| Run 13 | 0,05812 | 0,000132 |
| Run 14 | 0,05761 | 0,000126 |

Figure 6




Graph 7

As it is observed from graph 7 in the beginning of process along laminar region initially friction factor is greater and as increasing temperature process goes on f is linearly decreases. However when flow becomes transient + turbulent then dramatic increasing happens in magnitude of f. In this case changing f value changes rate of head loss at constant flow rate:

$$h_f = \left(\frac{8fLQ^2}{gD^5\pi^2}\right) \qquad (20)$$

$h_f$ – head loss along exit pipe in L distance

As a result it is apparent that increasing temperature is advantageous for minimizing head loss so that if process is possible to operate in laminar flow it is best case to have minimal pressure loss. If fluid has passed transient+turbulent region there will be greater head loss than laminar region but while temperature continues rising in this region raised friction factor will decrease slightly again

## CONCLUSION

According to investigated theoretical and experimental result we can makefollowing inferences:

External gear pump is hydraulic model of spur gear (the same module) reducer.
Observations show in reducer the gears rotating in meshing position create heat.The reason of heat is friction and Hertzian contact stress (induced as a result of load and rotation on mutual contact area of gears ) and this excessive heat damages reducer so to remove that undesired heat from gears of reducer some viscosity adjusted lubricants such as Mobilux-EP2 and Mobilux-EP023 are used. Since in external gear pump gears are completely surrounded by fluid, developed heat are completely absorbed by pumped medium thus this " removed heat" appears in outlet pipe of gear pump. For this reason, in comparison with other typeof positive displacement pumps, in the same flow rate gear pumps create "hotter" fluids in exit port. From another aspect, as it is plotted in graph 2 more flow rates decreases exit temperature. Increased flow rate is achieved through increasing RPM, and increasing RPM creates greater Hertzian contact stress.Greater contact stress leads to greater developed heat in the teeth of gears so greater outlet temperature but here it must be taken into account that more flow rate causes distribution of generated heat among more fluid thus in this case less outlet temperature is observed. For this reason undesired temperature can be reduced by achieving greater flow rate through increased RPM.
Along with these temperature affected parameters and their variations were determined through Graph 1,2,3. Due to those graphs outlet temperature value can inform us about operating capacity of external gear pump such that in high outlet temperature observed pumps it is clear that this is the sign of low efficiency and amount of reduction in efficiency becomes heat in exit port thus lead to high temperature values.From the other aspect, while outlet temperature is constant highpressure values are sign of high total efficiency of pump because greater amount ofenergy converted from motor shaft to fluid with less energy losses indicates itself as excessive pressure in exit port. These concepts were proved both theoretically and practically.

Additionally if outlet temperature exceeds medium's vaporization temperature this can cause serious dramatic results. Not only in diesel oil example but also in easy vaporizing mediums used in gear pumps, it is observed that while beginningvaporization process the medium expands its volume so increased volume in closed pipe creates more internal pipe pressure and if the process is not controlledwell pipes get harm and undergo destructive stresses. The best example of this situation is encountered in boiler (HVAC system component) high temperature outlet piping so that to overcome developed pressure hydraulic accumulators is installed at the exit pipe to absorb internal resistance of fluid.The same case can happen in outlet pipe of external gear pump therefore this shows how it is important to control temperature level of fluid in desired level due to both economical (without utilizing absorbing devices) and technical reasons.External gear pumps are widely used to pump high viscosity mediums such as polymers soon. Depending on requirements of process keeping at the same viscosity level of those mediums can be significant so in that cases high temperature values are not desirable for affecting viscosity levels and qualities of pumped material thus mustbe controlled by adjusting inlet temperature and other parameters of system. It was also studied that in rising temperature interval it is possible to achieve lower energy losses (head loss). For this purpose laminar region is the best optionif industrial parameters satisfied for laminarity conditions in the comparison withtransient + turbulent.

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