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Thermal Equilibrium of a Hydraulic Driving System

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Abstract: To put into evidence the consequence of the energetic losses that appears in a hydraulic driving systems and to evaluate how does the system performance and reliability are strongly affected by the temperature increase due-to the flowing fluid, in this study a thermal analysis is presented for improving the possibility of developing a practical and simplified method for establishing the optimum working temperature at any instant time. Focus is on computational methods that to be used for controlling the working temperature around the limit of admissible temperature, if the working temperature exceeds this limit, the fluid properties alteration will occur rapidly and a slow deterioration in the internal working parts of the system is expected, based on the failure rule rate that doubles for every 10°C of a temperature increase. Heat load duration is evaluated for both short and long operation periods, in which thermal equations are introduced to describe the conduction, convection and radiation modes of the heat transfer for the given mode of operation .The main conclusion of this study draws an important attention, that must be taken into account even during the first stages of designing such systems, in order to establish the optimum dimensions for the heat exchanger solution, as a design option when required for reducing the heat load for satisfying the needed working temperature and then keeping the system within the energy balance condition.

Key words: Energy balance, driving system, thermal equilibrium, working temperature

INTRODUCTION

The energetic losses, that appears in a hydraulic driving system due-to the flowing fluid are taken over by the fluid in a form of heat, which in turn determines a modification in the fluid temperature, from the surrounding temperature (at the moment when the system is started) to an operation temperature. Thermal equilibrium of the system, is reached at the moment when the generated heat is equal with the dissipated heat, the heat generation due-to the flowing fluid in the system will increase the working temperature, the dissipated heat as a result of a combination of conduction, convection and radiation from the system components, is represented by the existent difference between the surrounding temperature and the system temperature.

Generally it is admitted that the maximum working temperature of a given hydraulic driving system, should not exceeds the limit of 65°C, otherwise the temperature rise up to the established limit of 65°C, the fluid properties will alter rapidly depending on the amount of the temperature increase, knowing that the rule failure rate doubles for every 10°C of temperature increase[1] and then a significant deterioration of the fluid properties is observed, properties such as (density, viscosity, thermal conduction and specific heats).

The problem of keeping the working temperature within the range of the admissible limit of 65°C is a very important factor that determine the system performance and reliability, so it should be accounted even from the first stages of designing such system, also the time and mode of operation should be involved during the designing stages in order to obtain the temperature that suites the different working conditions within a thermal heat balance and then the real value of fluid viscosity must corresponds the value used in different computational and dimensioning methods. Knowing that the system performance and the working fluid properties are strongly affected due-to the temperature increase above the admissible working limit, that causes also, a slow deterioration of the internal working parts of the system, all of these aspects determine a maximum need for establishing an optimum working temperature that satisfy the energy balance at different working conditions, so the object of this study is to establish a usable procedure that determines the working temperature within a thermal equilibrium conditions, frequently two methods to determine this temperature are used[2-4].

The first method is based on the fact, that whole the lost energy is used to heat the fluid in the system, in away that the maximum temperature increase can be obtained, while the second method, takes into account that the

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dissipated heat through convection and radiation from the system components, although the second method is more accurate and exact for obtaining the temperature rise over the time than in the first method, but the difficulty of finding adequate values of the heat transfer coefficients is faced in the second method, as viewed as the problem of convection, because the heat transfer coefficients depends on the (fluid properties, surface geometry and on the flow conditions), all of these factors will causes the difficulty of obtaining an exactly adequate values of the heat transfer coefficients.

MATERIALS AND METHODS

In order to develop a suitable and simplified practical procedure for computing the temperature rise inside the given driving system at any instant time, both methods are presented in a numerical values by giving a real example as an application, in which both methods are used and then the obtained resulted values are compared in order to chose the optimum method that corresponds the given problem and ease determining the maximum admissible working temperature for achieving a thermal equilibrium, so both methods are presented below.

Method-1: The necessary amount of heat Q, as required to increase the fluid temperature with $\Delta\theta$, with a fluid volume V_u and a mass m, is given by:

$$Q = m \cdot C_Q \cdot \Delta\theta = \rho \cdot V_U \cdot C_Q \cdot \Delta\theta \qquad (1)$$

When we are required to determine the temperature rise as a product of the flowing fluid (q), that passes through a given component with a pressure drop ΔP, then equation (1), can be rearranged in a following form:

$$\Delta\theta = \frac{q \cdot \Delta p \cdot t}{\rho \cdot V_U \cdot C_Q} \qquad (2)$$

Where $Q = q.\Delta p.\Delta t$, is the heat generated during the time interval (t)?

Application: As an example, let taking a frequent universal case in a hydraulic driving systems in which the flow required by the driven part is zero, the system consists of a pumping group as shown in (Fig. 1) and we are required to determine the variation law of the fluid temperature in a time period, in situation that whole the supplied flow by a pump (P), passes through the safety valve (S_V) and then returns toward the main reservoir (R_Z).

Solution: In order to solve such type of problems and having whole the necessary data input as indicated on the

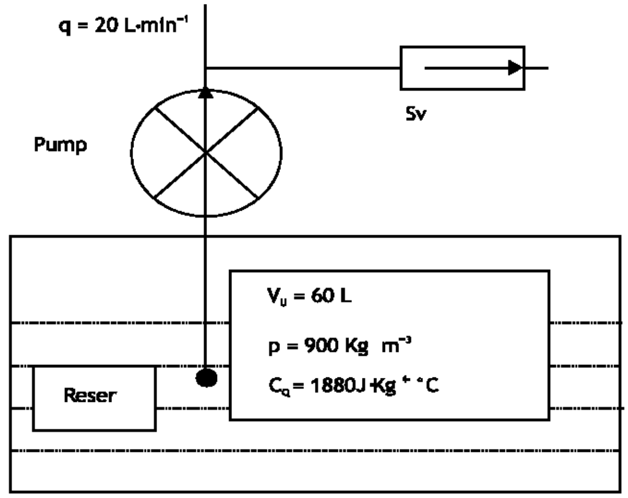

Fig. 1: Pumping group components

figure, so equation (2) is to be used for determining the dependency of:

$$\Delta\theta = \frac{q \cdot \Delta p \cdot t}{\rho \cdot V_u \cdot C_Q} = \frac{\frac{20.10^{-3}}{60} \cdot 180.10^5 \cdot t}{900 \cdot 60 \cdot 10^{-3} \cdot 180} = 0.059 \cdot t\ [°C]$$

This result indicates that, there are 3 minutes as a necessary time in which the pump can suck whole the amount of fluid from the system reservoir, then the fluid temperature will start to increase, as the time increases above the calculated time, so the fluid temperature during this time period of (3 minutes) is: $\Delta\theta = 0.059 \cdot 3 \cdot 60 = 10.62$°C.

By assuming that the fluid temperature at the moment in which the suction process starts is 20°C, we observe that only 13 working minutes are necessary for achieving the maximum admissible working limit of 65°C. The value of ($\Delta\theta$) that corresponds the fluid cycle of circulation can be expressed by substituting, $\frac{q \cdot t}{V_U}$ in equation (2), we obtain the following usable and simplified equation:

$$\Delta\theta = \frac{\Delta p}{\rho \cdot C_Q} \qquad (3)$$

By replacing the values of (ρ, CQ and Δp) in equation number (3), we will obtain the same result of 10, 62°C, which represent the fluid temperature rise for every flowing cycle from the main reservoir.

Method 2: The second method, takes into account the fact that an important amount of the heat is dissipated through convection and radiation from the surfaces of the system components, while the remained amount of heat is used for heating the fluid in the system, so the essence of this method is represented by the energy conservation law, that can be formulated as following:

$$\Delta N \cdot dt = \rho \, . \, V_U \, . \, C_Q \, . \, d\theta + \Sigma \, C_A \, . \, A \, . \, (\theta_P - \theta_a).dt \quad (4)$$

In which the component surfaces temperature (their walls temperature), represented as a function of the fluid temperature, so following expression can be used:

$$\theta - \theta_p = k \, . \, (\theta_p - \theta_a) \quad (5)$$

Where (k), is a coefficient that is determined experimentally and it depends on the fluid circulation mode inside the system components and on the air speed outside the system,

Denoting that: $$\theta_p = \frac{\theta - \theta_a}{k+1} + \theta_a \quad (6)$$

Replacing the expression (5) in equation (4), we will obtain the following equation:

$$\Delta N \, . \, dt = \rho \, . \, V_U \, . \, d\theta + \Sigma \, U \, . \, A \, . \, \Delta\theta \, . \, dt. \quad (7)$$

Where U, is the global (overall) heat transfer coefficient, [$j \cdot m^{-2} {}^\circ C$], in which some values of this coefficient are indicated in the following table.

Table 1: heat transfer coefficients

Coefficient	Conditions	Value
CQ	Fluid of a petroleum nature.	1880[$j \cdot m^{-2} {}^\circ C$]
U	1-Oil reservoir mode of ventilation:	[$j \cdot m^{-2} {}^\circ C$]
	• No ventilation	10 - 30
	• Natural ventilation	30 - 75
	• Forced ventilation	140 - 350
	2-Heat exchanger type:	-
	• Air-oil	140 – 350
	• Water-oil	450 – 550
	3-Steel pipe with a diameter of 50mm	6
C_A	When surface is exposed to:	[$j \cdot m^{-2} {}^\circ C$]
	1. Smooth air	11.4
	2. Ambient air	15.3
	3. Current of air flow	33.5
B	1- For weak oil circulation:	-
	Surface is exposed to smooth air flow	1-2
	Surface is exposed to strong air flow	0.5-1
	2-For good oil circulation:	-
	Surface is exposed to smooth air flow	0.5-1
	Surface is exposed to strong air flow	0.2 – 0.4

Denoting that: $$X = \frac{\sum U \cdot A}{\rho \cdot V_U \cdot C_Q} \text{ and } Y = \frac{\Delta N}{\rho \cdot V_Q \cdot C_Q} \quad (8)$$

By taking into account the fact that equation (6) becomes as:

$$\frac{d\Delta\theta}{dt} + X \cdot \Delta\theta = Y \quad (9)$$

This equation represents a simple differential equation (of the first order), with constant coefficients, whose solution gives the following new equation:

$$\Delta\theta = \frac{\Delta N}{\Sigma U \cdot A}\left(1 - e^{-X \cdot t}\right) + \Delta\theta_0 \cdot e^{-X \cdot t} \quad (10)$$

Where $\Delta\theta_0 = \theta_0 - \theta_a$, θ_0 being the initial temperature of the components surface. The developed expression (9) permits the calculation of the working temperature at every moment, while for a very long working period the temperature rise becomes as:

$$\Delta\theta_r = \frac{\Delta N}{\Sigma U \cdot A} \quad (11)$$

In fact that equation (10) is obtained by calculating, ($\lim_{t \to \infty} \Delta\theta$) in which, t = 1/x . So the difference between the computed temperature by using equation (9) and the other value as computed by using equation (10) is approx. about 5%, that is considered as the time value in which a thermal equilibrium is reached and then the regime temperature is computed by using equation number (10). An important conclusion that should be mentioned, is the fact that the necessary time for reaching a thermal equilibrium increases as the fluid volume increase inside the system, while the final temperature remains as the same, see equation (9). When the value of the working temperature exceeds the established admissible limit and for the same working conditions, in which whole the possible measures are taken to reduce the charge losses from the system, then the only solution that must be taken, will be by increasing the heat transfer surface area, such as to use the heat exchanger equipment.

Difficulties of realizing a thermal balance are by establishing the heat transfer coefficients and by evaluating the heat transfer surfaces for the given hydraulic driving system, so the main components that dissipate the largest amount of the developed heat by the system are (the heat exchanger, pipes and the system reservoir), in the same time the heat transfer coefficients can be founded and the computation of the heat transfer surfaces is not a difficult problem. Regarding the pumps, engines and the control commanded equipments of the hydraulic power, the heat transfer coefficients are almost non existent and for the first approximation in relation to the coefficients of (CQ and k), the values as indicated in Table 1, can be used for the information only when a quick evaluation of the temperature of the fluid inside the pump casing is required and for diagnosing some internal water-tightness problems.

Application 1: Let to take firstly the same example that was solved by using the first method, in which the required data is indicated in (Fig. 1), where the heat transfer coefficient is taken to be of (58 $J{\cdot}m^{-2\circ}C$) and the heat transfer area (A=1.4m2), then we are required to use the second method to determine the fluid temperature rise after (3 and 13) minutes from the moment of starting the system in operation.

Solution: In order to compute the dissipated heat, we assume that the initial temperature of the system components is the same as the surrounding temperature, so the dissipated heat will be:

$$\Delta N = q \cdot \Delta p = \frac{20 \cdot 10^{-3}}{60} \cdot 180 \cdot 10^5 = 6000 w$$ and:

$$\langle = \frac{\sum U \cdot A}{\rho \cdot V_U \cdot C_Q} = \frac{58 \cdot 1,4}{900 \cdot 60 \cdot 10^{-3} \cdot 1880} = 0.8 \cdot 10^{-3} \cdot s^{-\prime}$$

$$\Delta\theta_r = \frac{\Delta N}{\Sigma U \cdot A} = \frac{6000}{58 \cdot 1.4} = 73.89^\circ C$$

This obtained value of $\Delta\theta_r$, corresponds the temperature rise over the environmental temperature for reaching thermal equilibrium conditions. So we precede the calculation for both working periods as below:

For t=3 minutes, $X \cdot t = 0.8 \cdot 10^{-3} \cdot 3 \cdot 60 = 0.144$.
As assumed before that, $\theta_0 = \theta_a$, then the difference, $\Delta\theta_a = 0$ and by using equation (9) we will obtain that, $\Delta\theta_r = 73.89 \cdot (1\text{-}e^{-0.144}) = 9.9°C$.

This value represent the temperature rise as obtained by using the second method, while the first method that does not account the heat dissipation at the level of the system reservoir, the obtained value is: $\Delta\theta_r = 10.62°C$.
For t=13 minutes, $X{\cdot}t = 0.8{\cdot}10^{-1} \cdot 13.60 = 0.629$, so:
$\Delta\theta_r = 73.89\ .(1\text{-}e^{-0.629}) = 45.6°C$.

These results demonstrate the weak performance, regarding the heat transfer in reservoir with small dimensions.

Application 2: Let us to take the following example as a hydraulic driving system with the following data input.

Oil volume in the system conduits (pipes):
$Vuc = 0.06\ m^3$;
External surface area of the conduits: $Ac = 1.450\ m^2$;
For the system conduits, it is considered that:
$Uc = 4J{\cdot}m^{-2\circ}C$;
For a good ventilated system reservoir that has a capacity of 180 L, with a cubic shape and a side length of 0.8 m, it is considered that: $Urz = 4J{\cdot}m^{-2\circ}C$;

The environmental temperature is $\theta_a = 20°C$ and the initial temperature of the system is $\theta_0 = 30°C$.

Requests: Compute the necessary time for reaching thermal equilibrium and determine the value of the corresponding temperature.

Compute the necessary time for reaching the admissible temperature limit of 65°C, assuming continuous working conditions.

Determine the effective heat exchanging area that would keep the working temperature in the limit of 65°C.

Solution: The computational procedure will be made, by considering that only the system pipes and reservoir are responsible for dissipating the heat from the system, so we firstly compute the fluid volume from the system:

Vu =Vuc +Vurz, where (Vurz) is the quantity of oil in the reservoir, then
$Vu = 0.18 + 0.06 = 0.24\ m^3$ and the reservoir surface area will be:
$Arz = 66{\cdot}(0.8)2 = 3.84\ m^2$ and then:

$\Sigma ua = Uc \cdot Ac + Urz \cdot Arz = 4{\cdot}1.450 + 50{\cdot}3.84 = 197.8 J{\cdot}°C^{-1}$ and:

$$X = \frac{\sum U \cdot A}{\rho \cdot V_U \cdot C_Q} = \frac{197,8}{900 \cdot 0,24 \cdot 1880} = 4.87 \cdot 10^{-4} \cdot s^{-1}$$

, Then:

$$\Delta\theta_r = \frac{\Delta N_g}{\Sigma U \cdot A} = \frac{17.08 \cdot 10^{-3}}{197.8} = 86.35^\circ C$$

Concluding that a thermal equilibrium will be reached at (86.35+20) =106.35°C, that constitute an unaccepted value and the necessary time for the fluid to reach this temperature is:

$$t = \frac{1}{X} \cdot 3 = 3 \cdot \frac{1}{4.86 \cdot 10^{-4}} = 0.616 \cdot 10^4 s \approx 1.7h$$

The necessary time for the system to reach the maximum limit of working temperature of 65°C, in this case, the working temperature increase will be: $\Delta\theta_r = 65\text{-}20 = 45°C$ and the initial temperature of the component surfaces will be: $\Delta\theta_0 = 30\text{-}20 = 10°C$, so for calculating the necessary time in order to achieve the temperature of 65°C, we have to use equation (9). By making the replacement we will obtain:

$$45 = 86.35 \cdot (1\text{-}e^{-x \cdot t}) + 10 \cdot e^{-x \cdot t}$$ and

$$t = \frac{0.612}{4.87 \cdot 10^{-4}} = 1256.67 s = 20.9$$ minutes.

In case that the working period is longer than this period of 20.9 minutes, then the necessity of using a heat exchanger equipment is imposed, for preventing several problems that could appear due to the system over heating, that may result in exceeding the optimum working temperature (approx. 65°C).

For computing the effective heat transfer area for heat exchanger device, we must take into account the operating conditions for reaching thermal equilibrium. Thus, for $\Delta\theta r$ 45°C, as in equation (9) we get:

$$\sum ua = \frac{17.0810^3}{45} = 379.55 J.^\circ C^{-1}$$

In this case,

$\Sigma u.a = Uc \cdot Ac + Urz \cdot Arz + Us \cdot As$, Usas referring to the heat exchanger device that is obtained as: $Us. \cdot As = 379.55 - 197.8 = 181.75\ J \cdot {}^\circ C^{-1}$.

Choosing an air-oil heat exchanger type from Table 1, in which $Us = 2.00\ J.^\circ C^{-1}$

The heat exchanger area will be:

$$A_s = \frac{181.75}{200} = 0.91 m^2$$

So it is necessary to find a heat exchanger with heat transfer area to be at least equal to the computed value in order to maintain the system within a thermal balanced conditions (between the generated and dissipated heats)

Nomenclature

Q	Heat load	[J]	$\Delta\theta$	Temperature difference	[°C]
V_u	Fluid volume	[m^3]	Δp	Pressure drop	[Bar]
A	Area	[m^2]	ρ	Density	[$Kg \cdot m^{-3}$]
C_Q	Heat capacity	[$J \cdot kg^{-1} {}^\circ C$]	ΔN	Power of dissipation	[w]
m	Mass	[Kg]	V_{UC}	Oil volume in a conduit	[m^3]
q	Flow rate	[$L \cdot min^{-1}$]	A_C	Conduit (pipe) external surface area	[m^2]
t	Time	[s]	U_C	eat transfer coeff. For conduit	[$J \cdot m^{-2} {}^\circ C$]
P	Pressure	[Bar]	U_{RZ}	Heat transfer coeff. For reservoir	[$J \cdot m^{-2} {}^\circ C$]
C_A	Heat transfer coeff.	[$J \cdot m^{-2} {}^\circ C$]	A_{RZ}	Reservoir surface area	[m^2]
θ	Fluid temperature	[°C]	U_S	Heat transfer coeff. For the heat exchang.	[$J \cdot m^{-2} {}^\circ C$]
θ_p	Surface component temperature	[°C]	K	Experimental coeff.	[-]
θ_a	Surrounding temperature	[°C]	U	Overall heat transfer coeff.	[-]
θ_r	Working temperature	[°C]	A_S	Heat exchanger area	[m^2]
X, Y	Dimensionless	[-]			

In this study we tried to put into evidence the consequence of energy losses that appear in a hydraulic driving system over the system performance. The energy losses from the system generates an increase of the system working temperature, so if the temperature is too high, then the fluid properties will alter rapidly depending on the amount of a temperature increase, furthermore the following observations are clearly detected:

The fluid viscosity will decrease as a result of a temperature increase and then it causes a poor lubrication quality, which in turn increases the wear rate of the internal working parts.
The flow losses will increases, causing to diminish the system efficiency.
The durability and efficiency of the gaskets will be reduced.
The functional clearances will be modified and then getting to their cancellation or to cause a blocking of the mechanical parts.

The best way for reducing the working temperature consists of reducing the energy losses from the system. This arrangement has to be done even from the first stages of designing such a system, while in case that the system exist physically, then a thermal balance can be made by using the above presented methodology, where two solutions can be made by replacing the reservoir with a bigger one in order to suit the actual working conditions, or by introducing the heat exchanger equipment to the system.

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