TEMPERATURE RISE AND HEAT DISSIPATION SIMULATION OF HYDRAULIC ELEVATORS

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SUMMARY

This paper is to introduce a new method which deals with simulation of transient heat dissipation and dynamic temperature rise of elevator hydraulic systems. A computer software based upon the method proposed has been developed to predict the system thermodynamic behaviours in any instant of operation under variable load conditions, which is very significant to achieve system optimal design parameters. The simulation results obtained by this method is far more realistic compared to those calculated from the conventional method, which adopts empirical formulas and is widely used in the elevator industry. The predicted results of heat dissipation and fluid temperature rise obtained using the method proposed in this paper have been found good agreement with the measured results.

NOTATION

A	total heat dissipation area
A_1	inner surface area of component
A_2	outer surface area of component
C	specific heat
C_f	specific heat of fluid
C_i	specific heat of components or fluid
C_p	specific heat of pipe
F_i	heat dissipating area of components
G_i	weight of oil or components
h	lifting height
h_a	heat-transfer coefficient by convection on outside surface of pipe.
h_f	heat-transfer coefficient by convection on internal surface of pipe
H_1	rate of heat flow transferred from fluid to pipe wall
H_2	rate of heat flow transferred from pipe wall to atmosphere
H_z	rate of total loss converted into heat
k_1	average factor to calculate P_{nec}
k_2	coefficient used to calculate P_{ex}
K	heat conductivity
K_i	heat conductivity of component materials
m_f	rate of fluid mass flow

M_f mass of fluid
P_{ex} existing heat dissipation capacity of the system
P_{nec} necessary heat dissipation ability of the system
Q unit rate of heat generation
t system operation time
T temperature
T_{di} temperature difference between the component i and atmosphere
T_f fluid temperature
T_{in} inlet fluid temperature
T_r permissible temperature rise of oil
T_{w1} wall temperature at internal pipe surface
T_{w2} wall temperature at external pipe surface
T_m difference between average temperature of system and environment
T_i temperature at point i
w summation of all weights
W rate of power loss converted into heat
z motor starts per hour

Greek symbols

ε emissivity
σ Stefan-Boltzmann constant
ρ density

INTRODUCTION

Previous research has not provided an appropriate method to calculate dynamic temperature rise in hydraulic systems, especially at the unsteady state thermodynamic conditions. Since system designers are very uncertain about the thermodynamic conditions of the systems, the sizes of components such as coolers and reservoirs are often improperly selected and designed. Some unnecessary additional cost or early break down of components would adversely be resulted. If designers are able to analyze and simulate the system thermodynamic behaviours at the design stage, the system can be designed with optimal parameters.

For a hydraulic elevator designer, temperature simulation and heat dissipation seem to have more importance, since the hydraulic elevator is a comparatively large energy consuming engineering system, the majority of energy input into the system is converted into thermal energy which results in the rise of oil temperature. So the elevator designers need a method that can be used in any kind of load condition to predict thermodynamic behaviours of the prospect system. In the conventional methods, heat transfer process in components is assumed to be insignificant and the effects of many thermodynamic parameters of components, such as heat transfer coefficient by convection, thermal conductivity and emissivity of materials on the heat dissipation have not been taken into account, so the calculated system temperature results are quite inadequate. These methods would even fail to predict temperature results of a hydraulic system under unsteady state thermal conditions.

A more accurate theoretical model describing heat transfer of the hydraulic components during the system operation has been established in Reference [1]. A group of equations are derived by applying the First Law of thermodynamics and heat transfer theories to form the essence of this method. The effects of many coefficients on temperature rise and heat dissipation can be analyzed each in detail using this method. The application of this method in elevator hydraulic system is discussed in this paper, and some experiments have been carried out to validate simulated results.

CONVENTIONAL METHODS

Most elevator hydraulic system manufacturers use the following two empirical formulas to calculate the heat balance in the elevator hydraulic systems. The necessary heat dissipation of the practical system is calculated from:

$$P_{nec} = k_1 whz \tag{1}$$

The existing heat dissipation capacity of the system is obtained according:

$$P_{ex} = k_2 A T_r \tag{2}$$

It is noticed in the above two formulas that the heat generation is taken as an average value and the heat dissipation ability of the system is considered to be proportional to the temperature difference between the system and the environment. In fact, however, the system heat generation changes with its operation time, and the increase of system heat dissipation is not lineal with the increase of the temperature difference between the system and environment atmosphere. Furthermore, the coefficient k_2 can not reflect many factors which affect the heat dissipation of the system. Hence, the heat balance calculated using these formulas are quite inaccurate.

The most commonly used method to calculate heat balance and temperature rise in hydraulic systems is available in the literature [2], [3], [4]. In this method, the heat dissipation is calculated from:

$$H_z dt = \sum C_i G_i dT_{di} + \sum k_i F_i T_{di} \tag{3}$$

If the average initial temperature difference between system and surrounding atmosphere is T_{m0}, T_m can be solved as:

$$T_m = \frac{H_z}{\sum K_i F_i} (1 - e^{-\frac{\sum K_i F_i}{\sum C_i G_i}}) + T_{m0} e^{-\frac{\sum K_i F_i}{\sum C_i G_i}} \tag{4}$$

When t approaches infinity, the steady state temperature balance is reached and the maximum T_m remains constant:

$$T_{mmax}=\frac{H_z}{\sum C_i G_i} \tag{5}$$

The above equation would be correct to calculate the average system temperature if heat radiation were neglected. However, the experimental studies in Reference [5], [6] have concluded that heat radiation during hydraulic system operation under normal room temperature region plays a very important role in determining heat dissipation and can not be ignored. This means the maximum temperature rise given by the above equation must have been over estimated. Moreover, all the factors in Equation (3), (4) and (5) are taken as average values which mean that average temperature distribution has been assumed for each section of hydraulic system concerned. This would lead to unrealistic transient fluid temperature results, especially under variable load conditions.

METHOD PROPOSED

A much more accurate method has been developed in Reference [1], [5], [6], in which many factors including the difference between fluid and pipe wall temperatures, inner and outer pipe wall temperature difference and system radiation have been taken into account on the basis of hydraulic system thermodynamic analysis. A group of equations have been established to estimate heat dissipation as well as temperature rise in any section of hydraulic systems.

If it is considered that the temperature of parts of a hydraulic system differs substantially, it is necessary to treat the system as several discrete parts. Each part can be considered as an open thermal system with an unit temperature in that part, hence this unit fluid temperature can be calculated from

$$\frac{dT_f}{dt}=\frac{W-H_1+m_f C_f (T_{in}-T_f)}{M_f C_f} \tag{6}$$

where H_1 is given by

$$H_1=h_f A_1 (T_f-T_{w1}) \tag{7}$$

In order to calculate the heat transfer between the wall and the oil, H_1, the fundamental equation for heat conduction, which describes the heat distribution in the wall of components, is adopted. This equation is well known as:

$$\rho C\frac{\partial T}{\partial t}=K(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2})+\frac{\partial Q}{\partial T} \tag{8}$$

Its boundary condition in the outside is described as:

$$H_2=h_aA_2(T_{w2}-T_a)+\varepsilon\sigma A_2(T_{w2}^4-T_a^4) \tag{9}$$

Where the heat flow transferred from the pipe wall to its surrounding atmosphere by natural convection and radiation, H_2, is assessed using Newton Cooling Formula and the Stefan-Boltzmann Law.

The finite-difference method on the basis of control volume has been used to resolve this partial differential equation. Then the wall of the component may be taken to be consisting of a network of discrete points, each point represent a control volume as shown in Figure 1. The system operation time is also treated as a series of discrete small time intervals, and the derivations in this equation is replaced with the finite-differences to change this equation to a series of lineal equilibriums.

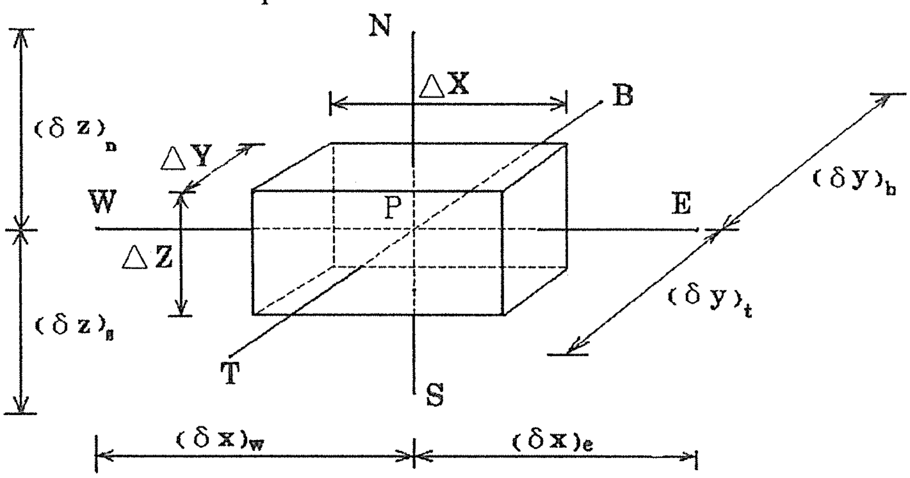


Figure 1 Control volume of P point

When a specific point, P, in the wall is considered at a specific time interval t_{i+1}, as illustrated in Figure 1, the lineal equation describing the heat balance in the P control volume can be derived as:

$$a_PT_P^{i+1}=a_ET_E^{i+1}+a_WT_W^{i+1}+a_NT_N^{i+1}+a_ST_S^{i+1}+a_TT_T^{i+1}+a_BT_B^{i+1}+b \tag{10}$$

where

$$a_E=\frac{K\Delta Y\Delta Z}{(\delta x)_E},\quad a_W=\frac{K\Delta Y\Delta Z}{(\delta x)_W},\quad a_B=\frac{K\Delta Z\Delta X}{(\delta y)_B},$$

$$a_T=\frac{K\Delta Z\Delta X}{(\delta y)_T},\quad A_N=\frac{K\Delta X\Delta Y}{(\delta z)_N},\quad a_S=\frac{K\Delta X\Delta Y}{(\delta z)_S}$$

$$a_P^0=\frac{\rho C\Delta X\Delta Y\Delta Z}{\Delta t}$$

$$a_P=a_E+a_W+a_N+a_S+a_T+a_B+a_P^0$$

$$b=Q\Delta X\Delta Y\Delta Z+a_P^0T_P^i$$

In the above equation, the coefficients of a_E, a_W, a_N, a_S, a_B, a_T represent the heat conductivity between P point and its six surrounding points, then $a_p{}^0T_p{}^i$ indicates the internal energy within control volume of P point. The constant b consists of this internal energy and the rate of heat generation or loss within this control volume concerned.

For the complete description of the heat transfer characteristics of wall of a component using the finite-difference equations, the relative boundary conditions must be defined to obtain sufficient equation number. For instance, if P point is adjacent to fluid in the direction of E as shown in Figure 1, the finite-difference equation of P can be derived from Equation (7) as:

$$a_P T_P^{i+1} = a_W T_N^{i+1} + a_N T_N^{i+1} + a_S T_S^{i+1} + a_T T_T^{i+1} + a_B T_B^{i+1} + a_E T_f^{i+1} + b \qquad (11)$$

where

$$a_E = h_f \Delta Y \Delta Z$$

When P is surrounded by atmosphere in the direction of W, the lineal equation is then derived from Equation (9) in the form of:

$$a_P T_P^{i+1} = a_E T_E^{i+1} + a_N T_N^{i+1} + a_S T_S^{i+1} + a_T T_T^{i+1} + a_B T_B^{i+1} + a_W T_a^{i+1} + b \qquad (12)$$

where

$$a_W = h_a \Delta Y \Delta Z, \quad b = a_P^0 + \varepsilon \sigma \Delta Y \Delta Z [\, (T_P^i)_4 - T_a^4)\,]$$

Equation (6) can be re-written in a finite-difference form:

$$\frac{T_f^{i+1} - T_f^i}{\Delta t} = \frac{W + m_f C_f (T_{in} - T_f^i) - \sum a_i (T_i^{i+1} - T_f^{i+1})}{M_f C_f} \qquad (13)$$

The above Equations (10), (11), (12) and (13) can be combined together in matrix form:

$$[A]\,[T] = [B] \qquad (14)$$

When this equation is resolved using numerical techniques, the temperature values of every discrete point at t_{i+1} as well as that of the fluid can be calculated as a result. In the same way, these temperature values at t_{i+1} can be used to calculate those of the t_{i+2} and so on. Therefore, the results including the dynamic fluid and wall temperatures and heat dissipation values of a component or any part of hydraulic systems can be obtained.

APPLICATION IN HYDRAULIC ELEVATOR

SIMULATION PROGRAMME A computer programme has been developed using MS-Fortran 5.0 to simulate temperature rise and heat dissipation in hydraulic elevator systems.

The structure of this programm is show in Figure 2. The programme has mainly two functions: one is to generate the matrix A and B of any temperature region of the system and the other is to resolve Equation (14) incorporated with these matrixes. This programme can be used to calculate the results including the dynamic fluid and wall temperatures and transient heat dissipation through the walls of pipelines, or those of reservoirs and cylinders respectively. Because the difference between the loop temperatures in the different part of the system is significant, the programme is designed to take the hydraulic system as a series of discrete open thermal systems with different uniform fluid temperature in each. Furthermore, this programme is designed to calculate the results of temperature, heat generation and dissipation under any type of variable load conditions, for which heat generation of a system has to be described as a function of time.

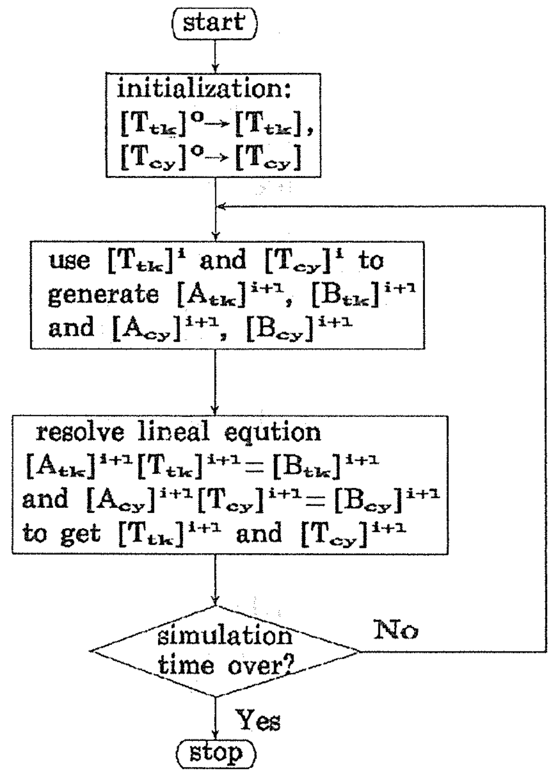


Figure 2 Structure of the simulation programme

EXPERIMENTAL SYSTEM The simplified circuit diagram of the practical hydraulic elevator system used to carry out the experiment is shown in Figure 3. A series of experiments have been carried out at the system to investigate its thermodynamic behaviours. This particular elevator system has a travel distance of 7.4 meters and supported by an indirect act cylinder. The static pressure of the system is 24.5 bar and its dynamic pressure 34.5 bar when no load is on. The flowrate of the pump in this system is 350 l/min. The load condition is changed by varying the starting number of the system. Since the pipe between the tank and the cylinder is rather short, this system can be considered only as two temperature regions, which means that the oil has two temperature values, one in the tank and the other in the

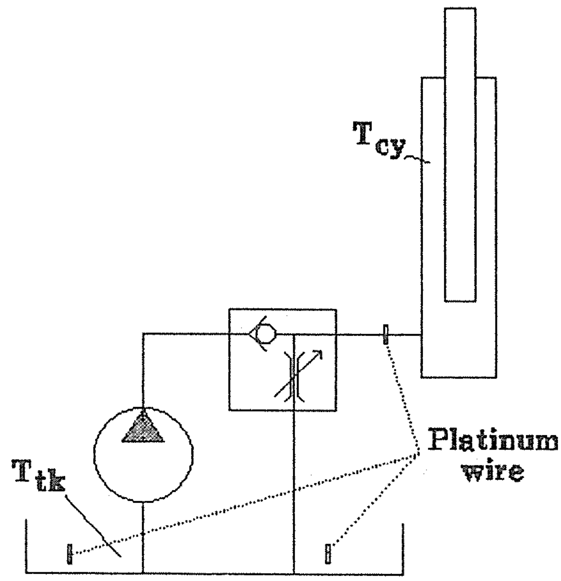


Figure 3 Simplified cercuit diagram of the experimental system

cylinder. The oil temperatures are measured with platinum thermal resistances. Two of the resistances are put into the reservoir to measure the reservoir temperature, while the oil temperature in the cylinder is taken by a thermal resistance placed in the outlet of the cylinder as the oil flows from cylinder back to the reservoir.

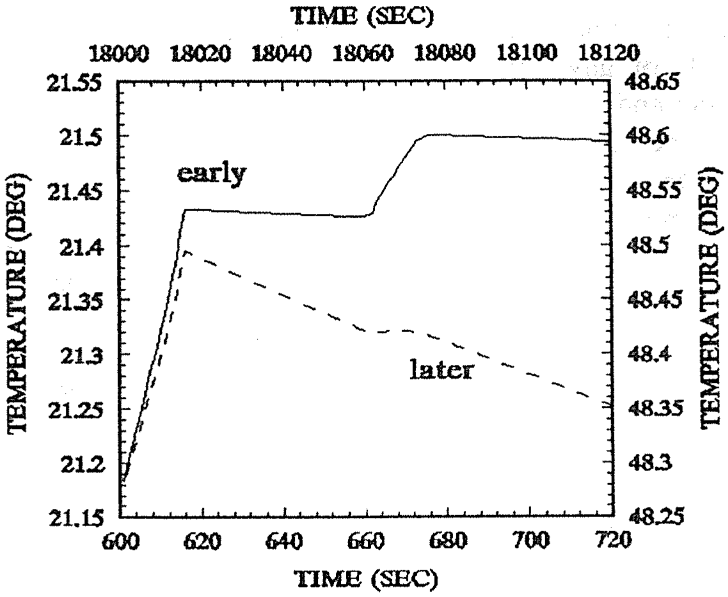
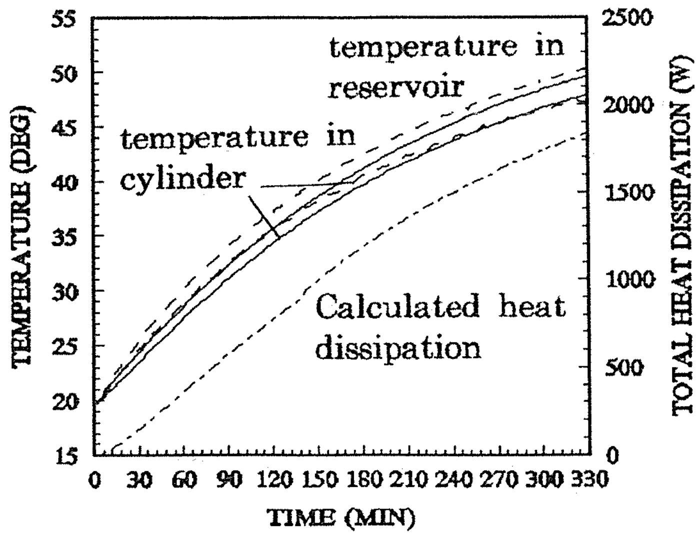


Figure 4 Oil temperature in reservoir in a working cycle

COMPARISON OF SIMULATED AND MEASURED RESULTS In the method proposed, system operation time is considered as a series of short intervals and the heat generation of the system is described as function of time, so the temperature in the system in any instant can be calculated. Figure 4 shows the oil temperature change in the reservoir in a system working cycle (cabin up and down once). The result in the early stage of the system operation shows that the reservoir temperature increases from 21.17 to 21.43 degree as the cabin travels from its bottom to top position. The temperature slightly drops during the stationary because the heat dissipation is quite small at the temperature level considered. When the cabin travels down, the temperature increases from 21.42 to 21.5 degree. On the other hand, in the later stage of system operation the temperature increases as the cabin travels up, and it drops from the 48.5 to 48.42 degree during the stationary as the heat dissipation is larger. The total temperature increase in one working cycle in early stage is much higher than the that of the later stage, which is caused by the variation of heat dissipation, although the heat



Fgure 5 Comparison of simulated and calculated results

generation is the same.

The comparison of simulated and measured temperature results of reservoir and cylinder, when the motor starts once every two minutes, is shown in Figure 5, and the total heat dissipation is also calculated. Good agreements have been found between the simulated and measured oil temperatures of the reservoir and cylinder. Since the temperature distribution in the wall of the components is calculated in the method proposed and also the computer programme, the heat dissipation from the outside of the component wall can be also predicted. This can be used to determine the cooling capacity if needed to maintain the system temperature at an accepted level so as to obtain a favorable working condition.

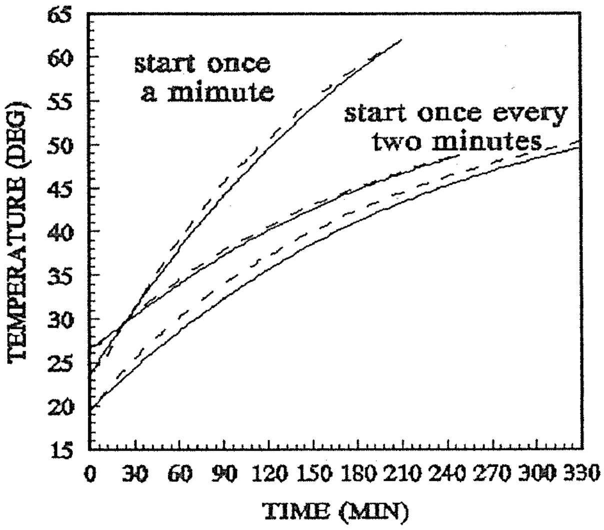


Figure 6 Comparison of simulated and measured results under different conditions

The comparisons of simulated and measured temperature results are shown in Figure 6 under different initial and load conditions. Close agreements between calculated and experimental results have been obtained for different initial and load conditions. In this way, the method and programme developed can be used to predict the temperature rise and heat dissipation of elevator hydraulic systems under a specific working environment places to obtain optimal design parameters.

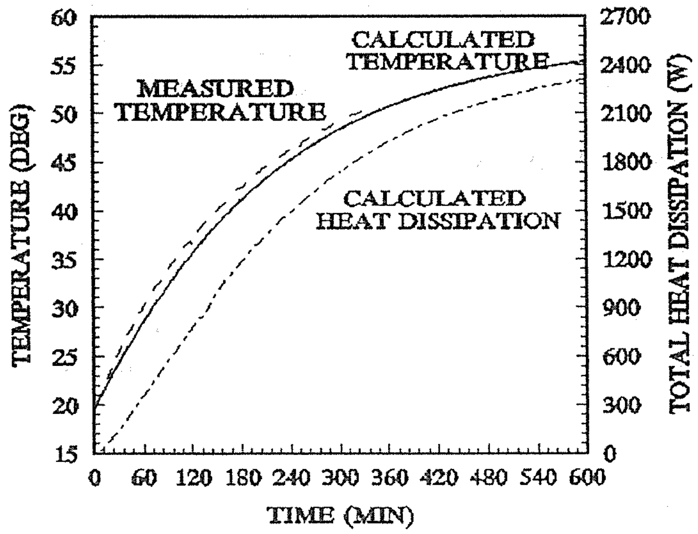


Figure 7 Comparison of simulated and measured results when no cooler needed

Figure 7 shows the calculated and experimental results when no cooler is needed and the motor starts twice every two minutes. In the case considered, the average heat generation of the system is about 2.4 Kw. The results show that the reservoir oil temperature will not exceed 55 degree, after the uninterruptedly ten hour system operation time, under the atmosphere temperature of 20 degree. The total average system heat dissipation increases to 2.3 Kw as the oil temperature reaches 55 degrees, which indicates that the heat balance is almost achieced, so no cooler is needed under such circumstances.

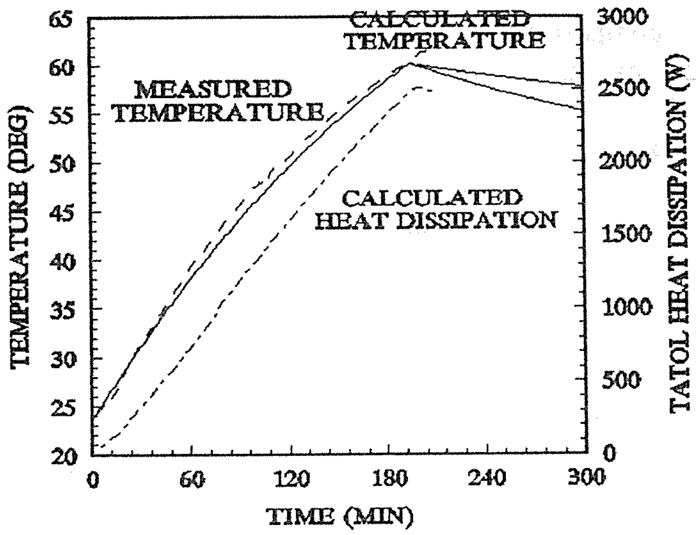


Figure 8 Comparison of simulated and measured results when a cooler is needed

In the system when the motor starts once a minute, the average total heat generation is doubled. The measured reservoir temperature in Figure 8 exceeds the acceptable temperature level of 60 degree after loading about 190 minutes. At the time, the average total heat dissipation is somewhat near 2.5 Kw, thus a cooler must be needed. As the cooling capacity of 2.5 Kw is considered in the simulation when the calculated temperarure goes above 60 degree, the temperature results falls back under 60 degree. If a 3.0 Kw cooler is turned on when the calculated temperature exceeds 60 degree, the reservoir temperature will fall back in a slightly faster speed.

CONCLUSION

A new method has been introduced in this paper to analyse temperature rise and heat dissipation in hydraulic elevator systems. On the basis of this method, a computer programme has been developed to calculate the heat dissipation and temperature rise in a practical hydraulic elevator system. Experiments have been carried out to validate the simulation results. Good agreements have been found between the calculated and measured results.

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