

Development of Heat Transfer Relationship for Modeling Energy Use in a Room

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Abstract
In a room, the use of minimal energy and the provision of necessary comfort are the most desired results. To achieve this result, optimum control must be achieved in the heaters and coolers that consume the most energy. In this study, a heat transfer model was developed to be used in optimal control methods in order to provide energy saving in a an environmental room by keeping comfort conditions. This model has been examined on the basis of a simulation. A test chamber established for this was taken as basis and the behavior of the heat transfer relation based on the energy balance was determined according to the time. The dynamics of thermal effects are described by the variables and model parameters described herein. The results show that the heat transfer model established for a selected room is stable. When the natural response of the system is taken into consideration, it is seen that it is stable without growing and not oscillating. The heat transfer model presented herein may at least be used in achieving a satisfactory control task. Simulation studies demonstrate that this model can be applied in environmental spaces and show that the results are promising.
Keywords: Heat transfer modeling, Control room, Optimal control.

1. Introduction

The volumes to be checked, housing, workplace, greenhouses, sea and air vehicles. It is possible to provide the desired weather conditions in these places with different energy expenditures. The aim is to operate and control the system within the comfort conditions, where this energy expenditure is at a minimum.

In environmental spaces, with energy saving, conditioning of air to keep temperature, humidity and air movement at the most appropriate levels for human health and comfort or for industrial processing is an expected feature. It has become compulsory to provide the environment conditions as a necessity for a manufacturing or processing, as well as for the reasons such as human health, increase of working efficiency and feeling of comfort.

The room based on this study was established by (Akgüney, 1994) and is shown in Figure 1. It was developed by a three variable model (Kaya, 1978) that keeps variables of temperature, humidity and air velocity, which affect the comfort conditions, in the comfort zone depending on each other, for the problem of minimizing the energy need in a certain activity level and clothing conditions. In a study conducted by using this developed model (Parmaksızoğlu, 1979), the mathematical model established with some assumptions was solved and its stability was shown. An energy use function based on the terms of comfort conditions and an optimal control problem (Kaya, 1981) were developed and solution methods based on an example were shown. This work was developed by (Kaya, 1982) and some experimental results were obtained with the energy savings provided. Taking these studies into account (Akgüney, 1994), an experimental chamber was established as a real system with multiple inputs and outputs. (Akgüney, 1994) establishes a dynamic relation between variables. (Akgüney, 1994), the equations of the nonlinear mathematical model were numerically solved and the system behavior was examined. In this study, the results of the experiments and the results of the simulations were compared. The academicians, such as (Taşdelen, İ., Baba, A. F., Erdal, H., Kozucuoğlu, A. H., 1995-2005), carried out a number of studies on this experimental installation, which was later established by (Akgüney, 1994). A lot of similar works are still going on today, with the emphasis on the matter.

Here, a heat transfer model is presented, which can be used to implement modern control methods for minimum energy use in a room. Heat loss and energy expenditure accordingly are among the most important parameters.

2. Material and Method

As a real system, the experiment room was established under the conditions required by (Akgüny, 1994). This room was designed and built considering the (Kaya, 1978, 1981) model. Inside the control volume, a particleboard of 0.018 m thickness was used and the air ducts were made of galvanized sheet metal. The lower and upper surfaces of the zone and the channels are well insulated. On the other surfaces, however, an insulation that is proportional to the heat transfer coefficient of the chipboard could be achieved. Here, the use of additional insulation material. The surfaces were painted with suitable paint so that the particle board would not work as a moisture capacitor. Some features of the room are shown in Figure 1.

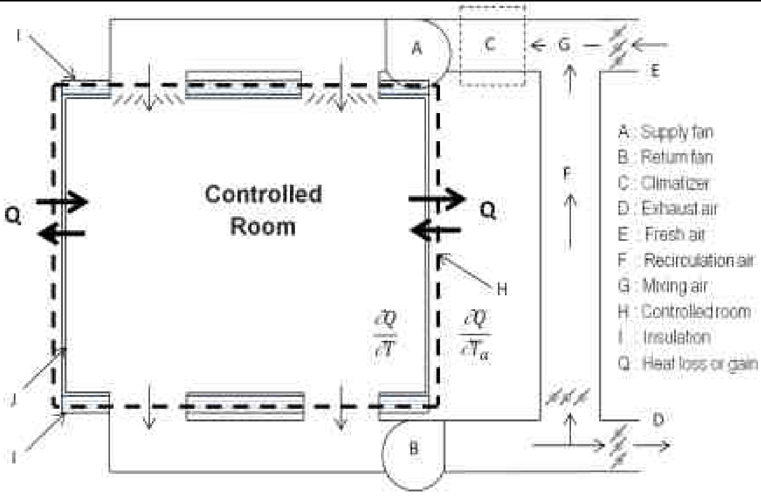

Figure 1. Room layout with heat transfer modeling

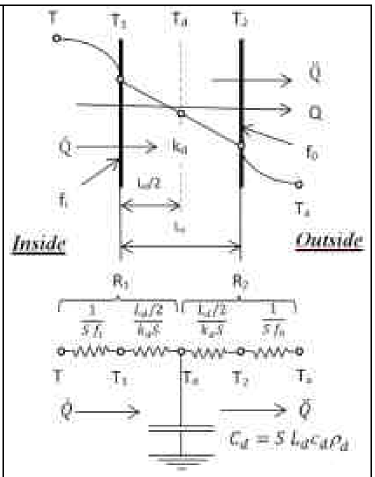

Figure 2. Heat transfer section for wall

Some values for the heat transfer model of this room are as follows.

Outer wall thickness	: 0.018 m
The density of the wall material	: 500 kg/m^3
The specific heat of the wall material	: 2.5 kJ/kg^{0}C
Heat transfer coefficient of wall material	: 0.5 kJ/mh^{0}C
Horizontal section	: 2 m^2
Volume	: 3.6 m^3
Unsealed surface area	: 10.8 m^2
Inner surface film coefficient	: 50 kJ/m^{2}h^{0}C
Outer surface film coefficient	: 30 kJ/m^{2}h^{0}C

Since the properties of the wall material are not standard values, approximate results are used. Practical values have been used since heat transfer coefficients (surface film coefficients) on the inner and outer surfaces depend on the speed, temperature and density of the air. The experiment room is located in an environmental laboratory area. In general, the internal surface film coefficient of buildings is assumed to be 7 kcal/m^{2}h^{0}C and for the external surface 20 kcal/m^{2}h^{0}C. In this system, 30 kJ/m^{2}h^{0}C is taken because the outer surface is in normal room conditions. On the inner surface, there is a faster air flow than the natural circulation. However, since there is a slower flow from the outside, it is accepted around 50 kJ/m^{2}h^{0}C.

The electrical representation of the heat conduction in the room walls is given in Figure 2. When examining the heat loss there, the temperature change between T_1 and T_2 can be considered linear, to simplify the process. In addition, the wall is examined in three parts, the two parts not absorbing heat and the other part absorbing heat. Non-absorbing elements, mass and L/2 thickness, the thermal resistances are equal and the heat transfer coefficients are k_d. The heat absorbing element is non-thickening mass m ρ_d v_d, where the specific heat is c_d. The energy balance of the room system in Figure 1 can be written as.

$$\left\{\begin{matrix}\text{Change}\\ \text{in air}\\ \text{energy}\\ (\rho V \frac{dh}{dt})\end{matrix}\right\} = \left\{\begin{matrix}\text{Heat energy}\\ \text{from the}\\ \text{heater}\\ (q)\end{matrix}\right\} + \left\{\begin{matrix}\text{The energy}\\ \text{of the}\\ \text{incoming air}\\ (\rho_e\, f\, h_e)\end{matrix}\right\} + \left\{\begin{matrix}\text{Humidifier}\\ \text{water}\\ \text{energy}\\ (m\, h_w)\end{matrix}\right\} - \left\{\begin{matrix}\text{The energy}\\ \text{of the}\\ \text{outgoing air}\\ (\rho\, d\, h)\end{matrix}\right\} - \left\{\begin{matrix}\text{Heat transfer}\\ \text{from}\\ \text{the walls}\\ (Q)\end{matrix}\right\}$$

3. Determination of The Amount of Heat to be Given

The energy balance value for the system after the control values of the air entering the system can be described as (Kaya, 1981).

$$h = h_e + \frac{q}{M} - \frac{Q}{M} \quad (1)$$

In the case of steady state heat loss for the system, (Mills, 1999) is written as follows.

$$Q = K\, S\, (T - T_a) \quad (2)$$

The enthalpy of the mixture air entering the system (Akgüney, 1994) is expressed as follows.

$$h_e = \frac{\rho\, r\, h + \rho_a (1-r) h_a}{\rho\, r + \rho_a (1-r)} \quad (3)$$

(1, 2 and 3) by using the equations,

$$h = \frac{\rho\, r\, h + \rho_a (1-r) h_a}{\rho\, r + \rho_a (1-r)} + \frac{q - S\,(T - T_a)}{M} \quad (4)$$

obtained. If necessary intermediate operations are performed,

$$h = \frac{\bar{T} + r\,(T_a - T)}{348.299\, A\, v\,(1-r)} [q - K\, S\,(T - T_a)] + h_a \quad (5)$$

be written like. The following expression is used for the enthalpy of moist air (mixture) containing 1 kg dry air and w kg water vapor.

$$h = c_{pa} T + w(h_g + c_{pw} T) \quad (6)$$

The specific humidity expression for the system can also be defined as follows (Akgüney, 1994).

$$w = 0.622\, \frac{\phi\, \alpha\, e^{\beta T}}{P - \phi\, \alpha\, e^{\beta T}} \quad (7)$$

If these statements are used,

$$q = \frac{348.289\, A\, v\,(1-r)}{\bar{T} + r\,(T_a - T)} [c_{pa}\,(T - T_a) + w\{c_{pw}\, T + h_g - \frac{0.622\, \lambda_a}{P - \lambda_a}(c_{pw}\, T_a + h_g)\}] + K\, S\,(T - T_a) \quad (8)$$

obtained. There are two separate heat transfers through conduction and convection from the wall of the room. Radyasyon ile ısı transferi ihmal edilir.

4. Development of Heat Transfer Expression

In Figure 2, the section of the electrical analogy expression for the wall is given. Bu kesit esas alınarak duvar için ısı transferi ifadesi geliştirilir.

4.1 Heat Transfer by Convection From Wall Inner Surface

$$\dot{Q}_t = S\, f_i\,(T - T_1) \quad (9)$$

4.2 Heat Transfer Through the Inner Wall

$$\dot{Q}_i = \frac{2\,S}{L_d}\, k_d\,(T_1 - T_d) \quad (10)$$

4.3 Heat Absorbe on the Wall

$$\dot{Q}_d = \rho_d\, S\, L_d c_d\, \frac{dT_d}{dt} = C_d\, \frac{dT_d}{dt} \quad (11)$$

Here, $C_d = \rho_d\, S\, L_d\, c_d$ be accepted as.

4.4 Heat Transfer by Conduction Around the Outer Wall

$$\ddot{Q}_i = \frac{2\,S}{L_d}\, k_d\,(T_d - T_2) \quad (12)$$

4.5 Heat Transfer by Convection on the Wall Surface

$$\ddot{Q}_t = S\, f_0\,(T_2 - T_a) \quad (13)$$

(9) and (10) are collected by equalizing the expressions according to the temperatures,

(9) using the numbered expression $T - T_1 = \frac{\dot{Q}}{S\, f_i}$, (10) using the numbered expression $T_1 - T_d = \frac{\dot{Q}\, L_d}{2\, S\, k_d}$ written and

$$(T - T_d) = \dot{Q}\,[\frac{1}{f_i} + \frac{L_d}{2\, k_d}]\, \frac{1}{S} \quad (14)$$

the end result is obtained. Here, $R_1 = [\frac{1}{f_i} + \frac{L_d}{2\, k_d}]\, \frac{1}{S}$ is described as. If the expressions (12) and (13) are equalized according to the temperatures and are collected again, (12) move from numbered identification $T_d - T_2 = \frac{\ddot{Q}\, L_d}{2\, S\, k_d}$, (13) move from numbered identification $T_2 - T_a = \frac{\dot{Q}}{S\, f_o}$ written and $Td - Ta = \frac{\ddot{Q}}{2\, S\, k_d} + \frac{\ddot{Q}}{S\, f_0}$ expression,

$$T_d - T_a = \ddot{Q}\, [\frac{1}{f_0} + \frac{L_d}{2\, k_d}]\, \frac{1}{S} \quad (15)$$

obtained. $R_2 = [\frac{1}{f_0} + \frac{L_d}{2\, k_d}]\, \frac{1}{S}$ is expressed as. (6) if the expression is rewritten,

$$\dot{Q}_i = \frac{1}{R_1}(T - T_d) \quad (16)$$

and (15) is also regulated,

$$\ddot{Q}_i = \frac{1}{R_2}(T_d - T_a) \quad (17)$$

obtained.

4.6 Heat Balance For The Wall

$$\dot{Q} = \dot{Q}_d + \ddot{Q}\,; \qquad \dot{Q} = \dot{Q}_t = \dot{Q}_i\,; \qquad \ddot{Q} = \ddot{Q}_t = \ddot{Q}_i \quad (18)$$

(11, 16 and 17) are substituted in the equation (18),

$$\frac{T-T_d}{R_i} = C_d \frac{dT_d}{dt} + \frac{T_d-T_a}{R_2} \quad (19)$$

and (17) the equation is rearranged,

$$T_d = T - R_i \dot{Q} \quad (20)$$

obtained. The time is derived from this,

$$\frac{dT_d}{dt} = \frac{dT}{dt} - R_1 \frac{d\dot{Q}}{dt} \quad (21)$$

(21) is used in the formula (19),

$$\frac{1}{R_1} (T - T_d) = C_d \frac{dT}{dt} - C_d R_1 \frac{d\dot{Q}}{dt} + \frac{T_d - T_1}{R_2} \quad (22)$$

and this expression $C_d R_1 \frac{d\dot{Q}}{dt}$ ıf equal to,

$C_d R_1 \frac{d\dot{Q}}{dt} = C_d \frac{dT}{dt} + \frac{T_d - T_a}{R_2} - \frac{1}{R_1} (T - T_d)$ obtained. If this statement is rearranged,

$$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} (T_d - T_a) - \frac{1}{C_d R_1^2} (T - T_d) \quad (23)$$

(20) expression (23) expression if used,

$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} \left[T - \dot{Q} R_1 - T_a\right] - \frac{1}{C_d R_1^2} [T - T + \dot{Q} R_1]$ and by performing intermediate operations, the expression of the change of heat transfer with respect to time is obtained as follows..

$$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} (T - T_a) - \frac{\dot{Q} R_1}{C_d R_1 R_2} - \frac{1}{C R_1^2} (\dot{Q} R_1)$$

$$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} (T - T_a) - \frac{\dot{Q}}{C_d R_2} - \frac{\dot{Q}}{C_d R_1}$$

$$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} (T - T_a) - \frac{\dot{Q}}{C_d} [\frac{1}{R_1} + \frac{1}{R_2}]$$

If these equations are regulated the heat loss takes the following final shape.

$$\frac{d\dot{Q}}{dt} = \frac{1}{R_1}\frac{dT}{dt} + \frac{1}{C_d R_1 R_2} [T - T_a] - \frac{\dot{Q}}{C_d} [\frac{R_1 + R_2}{R_1 R_2}] \quad (24)$$

Here, the equation (24) can be written in the t-domain as $\varphi = \frac{R_1 + R_2}{C_d R_1 R_2}$,

$$\frac{dQ(t)}{dt} = \frac{1}{R_1}\frac{dT(t)}{dt} + \frac{1}{C_d R_1 R_2} [T(t) - T_a(t)] - \varphi Q(t) \quad (25)$$

In this case, if the conversion of Laplace is taken,

$$[S\, Q(s) - Q(o)] = \frac{1}{R_1} [S\, T(s) - T(o)] + \frac{1}{C_d R_1 R_2} [T(s) - T_a(s)] - \varphi Q(s) \quad (26)$$

(26) can be rearranged by performing intermediate operations.

$$S\, Q(s) - Q(o) + \varphi Q(s) = \frac{1}{R_1} [S\, T(s) - T(o)] + \frac{1}{C_d R_1 R_2} [T(s) - T_a(s)].$$

If $\dot{Q}(o)$ is taken in the common parenthesis.

$$(S + \varphi)\, Q(s) - Q(o) = \frac{S\, T(s)}{R_1} - \frac{T(o)}{R_1} + \frac{T(s)}{C_d R_1 R_2} - \frac{T_a(s)}{C_d R_1 R_2}$$

$S + Q(s) = \frac{S\, T(s)}{R_1} - \frac{T_a}{R_1} + \frac{T(s)}{C_d R_1 R_2} - \frac{T_a(s)}{C_d R_1 R_2} + Q(o)$ and if edited,

$$Q(s) = \frac{S}{(S + \varphi) R_1} T(s) - \frac{T(o)}{(S + \varphi) R_1} + \frac{T(s)}{(S + \varphi) C_d R_1 R_2} - \frac{T_a(s)}{(S + \varphi) C_d R_1 R_2} + \frac{\dot{Q}(s)}{(S + \varphi)}$$

if the above equation is edited,

$$Q(s) = \frac{Q(s) - \frac{T(o)}{R_1}}{(S+\varphi)} + \frac{(S+\ \varphi - \varphi)}{(S+\varphi)R_1} T(s) + \frac{1}{C_d\, R_1\, R_2} \frac{T(s)}{(S+\varphi)} - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\varphi)}$$

$$Q(s) = \frac{Q(0) - \frac{T(0)}{R_1}}{(S+\varphi)} + \frac{1}{R_1} T(s) + \frac{\varphi}{(S+\varphi)R_1} T(s) + \frac{1}{C_d\, R_1\, R_2} \frac{T(s)}{(S+\varphi)} - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\varphi)}$$

$$Q(s) = \frac{Q(0) - \frac{T(0)}{R_1}}{(S+\varphi)} + \frac{T(s)}{R_1} + \frac{T(s)}{(S+\varphi)} \left(\frac{-\varphi}{R_1} + \frac{1}{C_d\, R_1\, R_2}\right) - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\varphi)}$$

In the above equation if the actual value in parentheses of φ is written,

$$Q(s) = \frac{Q(o) - \frac{T(o)}{R_1}}{(S+\ \varphi)} + \frac{T(s)}{R_1} + \frac{T(s)}{(S+\ \varphi)} \left(\frac{-R_2 - R_1}{C_d\, R_1^2\, R_2} + \frac{R_1}{C_d\, R_1^2\, R_2}\right) - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\ \varphi)}$$

$$Q(s) = \frac{Q(o) - \frac{T(o)}{R_1}}{(S+\ \varphi)} + \frac{T(s)}{R_1} + \frac{T(s)}{(S+\ \varphi)} \left(-\frac{R_2}{C_d\, R_1^2\, R_2}\right) - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\ \varphi)}$$

$$Q(s) = \frac{Q(0) - \frac{T(0)}{R_1}}{(S+\varphi)} + \frac{T(s)}{R_1} - \frac{1}{C_d\, R_1^2} \frac{T(s)}{(S+\varphi)} - \frac{1}{C_d\, R_1\, R_2} \frac{T_a(s)}{(S+\varphi)} \qquad (27)$$

We make the inverse Laplace transformation of number (27),

$$Q(t) = e^{-\varphi t}\, [Q(o) - \frac{T(o)}{R_1}] + \frac{T(t)}{R_1} - \frac{e^{-\varphi t}}{C_d\, R_1^2} \int_0^t e^{\varphi\tau}\, T(\tau)\, d\tau - \frac{e^{-\varphi t}}{C_d\, R_1\, R_2} \int_0^t e^{\varphi\tau}\, T_a(\tau)\, d\tau \qquad (28)$$

(28) is a derivative according to T,

$$\frac{\partial Q}{\partial T} = \frac{1}{R_1} - \frac{1}{C_d\, R_1^2} [1 - e^{-\varphi t}] \qquad (29)$$

(28) is a derivative according to T_a,

$$\frac{\partial Q}{\partial T_a} = -\frac{1}{\varphi\, C_d\, R_1\, R_2} [1 - e^{-\varphi t}] \qquad (30)$$

In the case of $\varphi = \frac{R_1 + R_2}{C_d\, R_1\, R_2}$ substituting the required values, φ = 1.7499 it is found as. With the use of this constant, the following result is obtained.

$$\frac{\partial Q}{\partial T} = 115.7412\, [1 - e^{-1.7499\, t}] \qquad (31)$$

$$\frac{\partial Q}{\partial T_a} = -115.7225\, [1 - e^{-1.7499\, t}] \qquad (32)$$

(29) and (30)'s, the variation with time has been examined. The graph of this behavior is drawn in Figures 3 and 4.

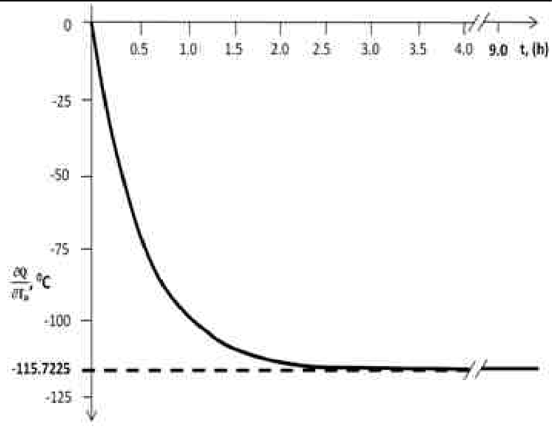

Figure 3. Graph of $\partial Q/\partial T_a$'s change with time

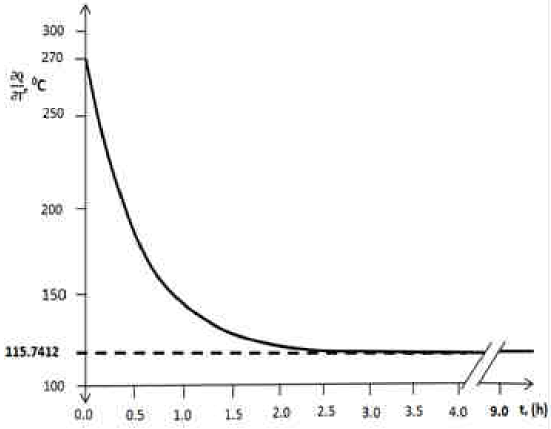

Figure 4. Graph of $\partial Q/\partial T$'s change with time

Table1. The calculated values of $\partial Q/\partial T$ ve $\partial Q/\partial T_a$'s variation with time

Number	t, Time (h)	$\partial Q/\partial T_a$, Temperature (^{0}C)	$\partial Q/\partial T$, Temperature (^{0}C)
1	0.0000	0.0000	270.0000
2	0.0001	- 0.0202	269.9971
3	0.0010	- 0.2025	269.7303
4	0.0100	- 2.0077	267.3239
5	0.1000	- 18.5780	245.2361
6	0.5000	- 67.4801	180.0490
7	1.0000	- 95.6110	142.5500
8	1.5000	-107.3384	126.9173
9	2.0000	-112.2273	120.4003
10	2.5000	-114.2655	117.6835
11	3.0000	-115.1150	116.5509
12	3.5000	-115.4682	116.0787
13	4.0000	-115.6169	115.8819
14	4.5000	-115.6784	115.7998
15	5.0000	-115.7041	115.7656
16	5.5000	-115.7148	115.7513
17	6.0000	-115.7193	115.7454
18	6.5000	-115.7211	115.7429
19	7.0000	-115.7219	115.7418
20	8.0000	-115.7224	115.7413
21	9.0000	-115.7224	**115.7412**
22	10.0000	**-115.7225**	115.7412
23	∞	-115.7225	115.7412

5. Conclusion and Discussion

In order to provide a certain level of activity and clothing, it is necessary to bring the comfort conditions to the desired values in the places with people. Energy use is inevitable for this. Temperature, humidity, air velocity and air mixing ratio are the most important variables in the problem of minimizing energy use. In modern control methods using computer / microprocessor, it is necessary to establish a heat transfer model for the walls of these rooms in order to provide both comfort conditions and energy saving in the air conditioning rooms. In this study, a heat transfer model was established for the experimental chamber, which was established as a multi-inlet and multi-outlet real physical system. While the model is being constructed, thermodynamic properties have been taken into account and simplified with acceptance.

In this model, the change of $\partial Q/\partial T_a$ and $\partial Q/\partial T$ is studied theoretically. The results are also given graphically. According to the results, this model was found to behave as a behavior behaving steadily. The

theoretical results of the model were found to be in full agreement with the behavior. The ratings on the graph are extremely positive. The model presented here seems to be possible to use in energy-driven buildings to achieve satisfactory control performance in order to use the necessary automatic control with modern control theories. The work in this area should be implemented in real buildings in order to obtain more accurate values.
This heat transfer model can be used in the following stages for optimum control problems.

1- For multi-input and multi-output rooms, it is used to establish a mathematical model expressing the dynamic relation between variables.
2- The nonlinear mathematical model is used in the numerical solution of the equation sets and in the study of the behavior of the system.
3- It is used to compare the test results with the simulation results.
4- It is used in control and room volume optimizations.
5- By staying within the limits of comfort, the least use of energy is used to optimize and solve the optimization problem.
6- Multivariable feedback is also used to check the state variables.
7- The performance criterion is improved and used to solve the optimal control problem.
8- It is used for optimum control and optimization techniques in all energy-consuming volumes.

6. Symbols

M	: Mass flow rate, (kg/h)
Q	: Heat loss, (kJ/h)
q	: Heat control variable, (kJ/h)
K	: Total heat transfer coefficient of the wall, (kJ/m^2h^0C)
S	: Wall heat loss surface area, (m^2)
$\dot{Q}_t$	: Heat transfer by convection on the inner wall surface, (kJ/h).
S	: Wall heat transfer surface area, (m^2).
f_i	: The film coefficient of the inner surface of the wall, (kJ/m^2h^0C).
T_1	: Internal surface temperature of the wall, (^0C).
$\dot{Q}_i$	: Heat transfer from the inner wall through conduction, (kJ/h).
L_d	: Total wall thickness, (m).
T_d	: The temperature in the middle of the wall, (^0C).
k_d	: Heat conduction coefficient of the wall, (kJ/m^2h^0C).
$\dot{Q}_d$	: Absorbed heat on the wall, (kJ/h).
C_d	: Average thermal capacity of the wall, ($kJ/^0C$).
c_d	: Average specific heat of the wall, (kJ/kg^0C).
ρ_d	: The average density of the wall material, (kg/m^3).
$\ddot{Q}_i$	: Heat transfer through the outside wall, (kJ/h).
T_2	: Wall, outside surface temperature, (^0C).
R_1, R_2	: Thermal resistances, (m^2h^0C/kJ).
$\dot{Q}$	: $T - T_d$ Heat transfer at the temperature difference between, (kJ/h).
$\ddot{Q}$	: $T_d - T_a$ Heat transfer at the temperature difference between, (kJ/h).
$\ddot{Q}_t$	: Heat transfer by convection from wall outer surface, (kJ/h).
f_0	: Outer surface film coefficient of wall, (kJ/m^2h^0C).

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