Palestine Polytechnic University

College of Engineering and Technology

Mechanical Engineering Department

Graduation Project

Design and Control of Variable Air Volume (VAV) Box in an Air Conditioning System

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Symbol	Definition
A	*Area* (m^2)
A_d	The cross sectional area of the duct (m^2)
$\sum AUi$	Area integrated fabric surface U-value ($W/m^{2\circ}C$)
$CLTD$	Cooling load temperature difference (°C)
c_p	Specific heat capacity at constant pressure (KJ/Kg.K)
c_v	Specific heat at constant volume (cal/Kg °C)
D_a	The armature damping (N.m.s/rad)
D_l	The load damping (N.m.s/rad)
D_m	The equivalent damping (N.m.s/rad)
DR	Daily range (°C)
E_1	Emissivity of surface of gray body
f	The attic roof factor and its value
h_c	The average convective heat-transfer coefficient, ($W/m^2.°C$)
h_i	The inside surface heat-transfer coefficient at the liquid-to-solid interface ($W/m^{2\circ}C$)
h_o	Outside surface heat-transfer coefficient at fluid-to-solid interface ($W/m^{2\circ}C$)
I_a	Armature current (A)
J_a	The armature inertia ($Kg.m^2$)
J_l	The load inertia ($Kg.m^2$)
J_m	The equivalent inertia ($Kg.m^2$)
K	Thermal conductivity (W/m°C)
K_A	respectively of the composite wall A The thermal conductivity of layers (W/m°C).
K_c	The color correction factor
KE	Kinetic Energy (W)

Symbol	*Definition*
K_r	The room temperature gain to fluctuations in process output
K_t	I_a and T_m The constant of proportionality between
L_A	The thicknesses of layer A (m)
LM	The latitude correction factor and can be obtained from tables for roofs and wall
n_v	Ventilation air change rate (1/h)
Q_{in}	Air conditioning input energy (W)
Q_{tot}	The total heat transferred to the system from the surroundings (W)
q_c	Heat gain by conduction (W)
q_i	The rate of heat transfer (W)
q_h	Heat gain by convection (W)
q_o	the rate of heat-transfer at the outside surface of the composite wall (W/m²°C)
q_r	Radiation heat transfer coefficient (W).
R	Thermal resistance of layer (°C/W)
$T(t)$	Temperature as a function of time (°C)
ΔT	The temperature difference between outdoor and the design temperature (K)
T_i	The inner temperature (°C)
T_m	is the torque developed by the motor (N.m)
$T_{o,m}$	The mean outside temperature (°C)
T_{R1}	Absolute temperatures of surface 1 (K).
T_s	The surface temperature (°C)
T_{ss}	Settling Time (s)
τ_r	Time constant of the room transfer function (s)

Symbol	*Definition*
U	Overall heat-transfer coefficient of exterior wall or roof. (W/m² °C)
U_{in}	Internal Energy (W)
V	The air velocity (m/sec)
$\dot{v}$	The air flow rate (m³/sec)
υ_r	Room volume (m³)
W	Output work from the system (W)
x	Coordinate dimension along heat flow (m)
σ	Stefan-Boltzmann constant
δ	The thickness of layer (m)
θ	The angle of the damper (degree)
ω_n	The Natural Frequency (rad/s)
ζ	Damping Ratio
ρ	Air density (Kg/m³)

Dedication

To our parents who
spent nights and days doing their best
to give us the best....

To women, the sign of love, beauty, and loyalty

To every woman who decided to struggle
against life challenges

To all students and who
wish to look for
the future ...

To who love the knowledge and
looking for the new
in this world ...

To our beloved country Palestine...

To the souls of Palestinian Martyrs ...

To who believe just in
peace as a right for all nations...

To all of our friends...

Anwar Jweles
Eman Al_Ajlouni
Ghadah Tbakhi

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ABSTRACT

Heating Ventilating and Air Conditioning (HVAC) System is one of the systems that had widely spread among people in the recent decades. Their use is not anymore restricted to people in a certain field or industry, but they became a group of those systems satisfying basic requirements of life. This development created problems related to energy consumption and energy saving, which in turn opened a wide research area dealing with the solution of such problems. The energy conservation acquired by reducing amount of cooling energy and fan speed during the operation.

This project aims to find the optimal solution of the problem of energy losses in the HVAC systems, by introducing the design and control of a Variable Air Volume (VAV) system for a cooling system. The project started by having a look on the various types of the VAV systems found in the market, then choosing one of these systems, namely the VAV cooling system, to be designed and controlled. A mathematical model for the system was built based on the fundamentals of the energy balance. In addition, the main thermal analysis and control design considerations are being studied in order to obtain a more representative model for the system.

The whole system has been designed and simulated to be officinal to apply on a small prototype. This prototype consists of two rooms provided with a temperature sensor in each one, central cooling unit and three VAV Boxes, one VAV Box for each room and the third one for the excess air. All these components have been constructed to examine the changing in VAV Box opening according to the temperature variations in each room and the energy conservation that represented by the excess air flow.

In air conditioning systems many items are supposed to be taken under considerations, such as temperature, humidity, noise level, oders, and CO_2 amount in order to determine the quality of the air. In this project, the temperature variation is the only item discussed.

Chapter One

INTRODUCTION AND LITERATURE REVIEW

Chapter One

INTRODUCTION AND LITERATURE REVIEW

1.1 Overview

One of the modern Heating, Ventilating, and Air Conditioning (HVAC) systems is the Variable Air Volume (VAV) system. A (VAV) system is an air system that varies its supply air volume flow rate to match the variation of zone load during part-load operation to maintain a predetermined space parameter, usually air temperature, and to conserve fan power at reduced volume flow. A Constant Air Volume (CAV) system varies its supply air temperature to match the reduction of space load during part load operation to maintain a predetermined space air temperature, typical (VAV) system is shown in Fig1.1.

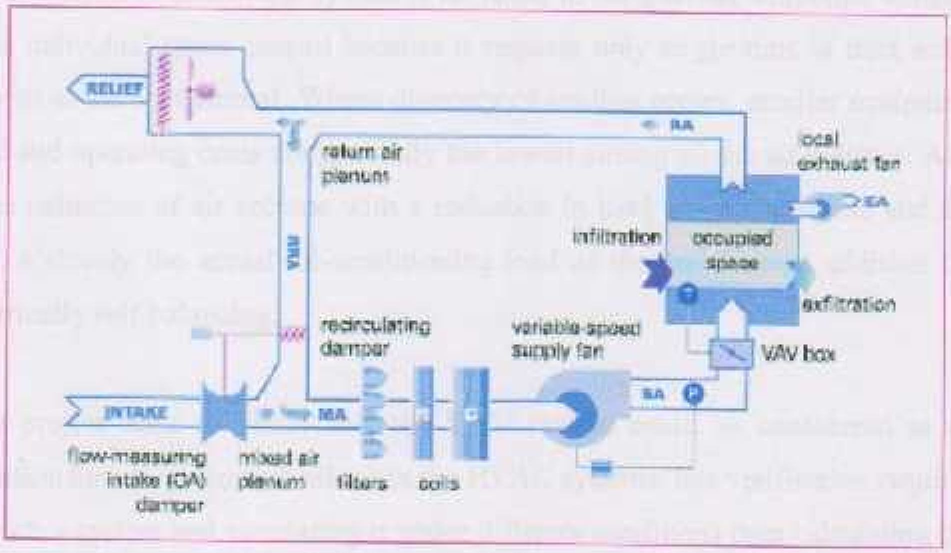


Figure 1.1 VAV System

Control of HVAC systems covers a wide range of products, functions, and sources of supply. Control can be defined as the starting and stopping or modulation of a process to regulate the condition being changed by the process. The application of controls starts with an understanding of the building and HVAC systems and the use of spaces to be conditioned and controlled. In addition the type of HVAC system plays an important role in control sequences. The basic control sequence can then be done by several types of control techniques such as pneumatic, electric, analog electronic, or electronic direct digital control (DDC) [4], [9].

1.2 Problem Definition

Nowadays, all science fields are directing towards reducing energy consumption and satisfying the luxury of human in all his occupants (home, office, hospital, shopping centers,...etc) which results in a wide spread of the HVAC systems. As a result VAV systems are introduced for their significant advantages including low first cost and low operating cost. The first cost of the system is far lower in comparison with other systems that provide individual space control because it requires only single runs of duct and a simple control at the air terminal. Where diversity of loading occurs, smaller equipment can be used and operating costs are generally the lowest among all the air systems. As a result of the reduction of air volume with a reduction in load, the refrigeration and fan power follow closely the actual air-conditioning load of the building. In addition the system is virtually self-balancing.

This project aims to verify that the VAV system could be considered as an optimal solution for energy conservations in the HVAC systems, this verification requires designing such a system and simulating it under different conditions then calculating the energy consumed and comparing the results with that which appear in the classical

HVAC systems. To validate the idea, a prototype will be built and different experiments will be carried in order to study the result of the developed system.

The cooling unit which will be used as a cooled air source is the central cooling unit of the specifications shown in Appendix (A). It is noticed that the fan speed is constant and can not be modulated although in the VAV system the position of the damper is modulated followed by the modulation of fan speed. With the case of this project, another representation of fan speed modulation will be established which will be explained in the nextchapters.

1.3 Project Description

The prototype which will be built is composed of two rooms of the same structural design, Fig 1.2 shows the layout of the two zones that will be built and conditioned, the building material includes two layers of gypsum board and an insulation material between the two layers, this is for the front and north walls, but the back wall is the original external wall of the workshop where the prototype will be established, and the southern is the internal wall. The two rooms are identical, and each is of the following dimensions:

Length = 1.5 m.
Width = 1.2 m.
Height = 2.15 m.

Each room has one door, the door dimensions are (0.70m * 1.70m), and the location of the door is in the front wall. The cooled air is supplied to the rooms by standard flexible ducts from the cooling unit, the diameter of the circular duct is 8 inches.

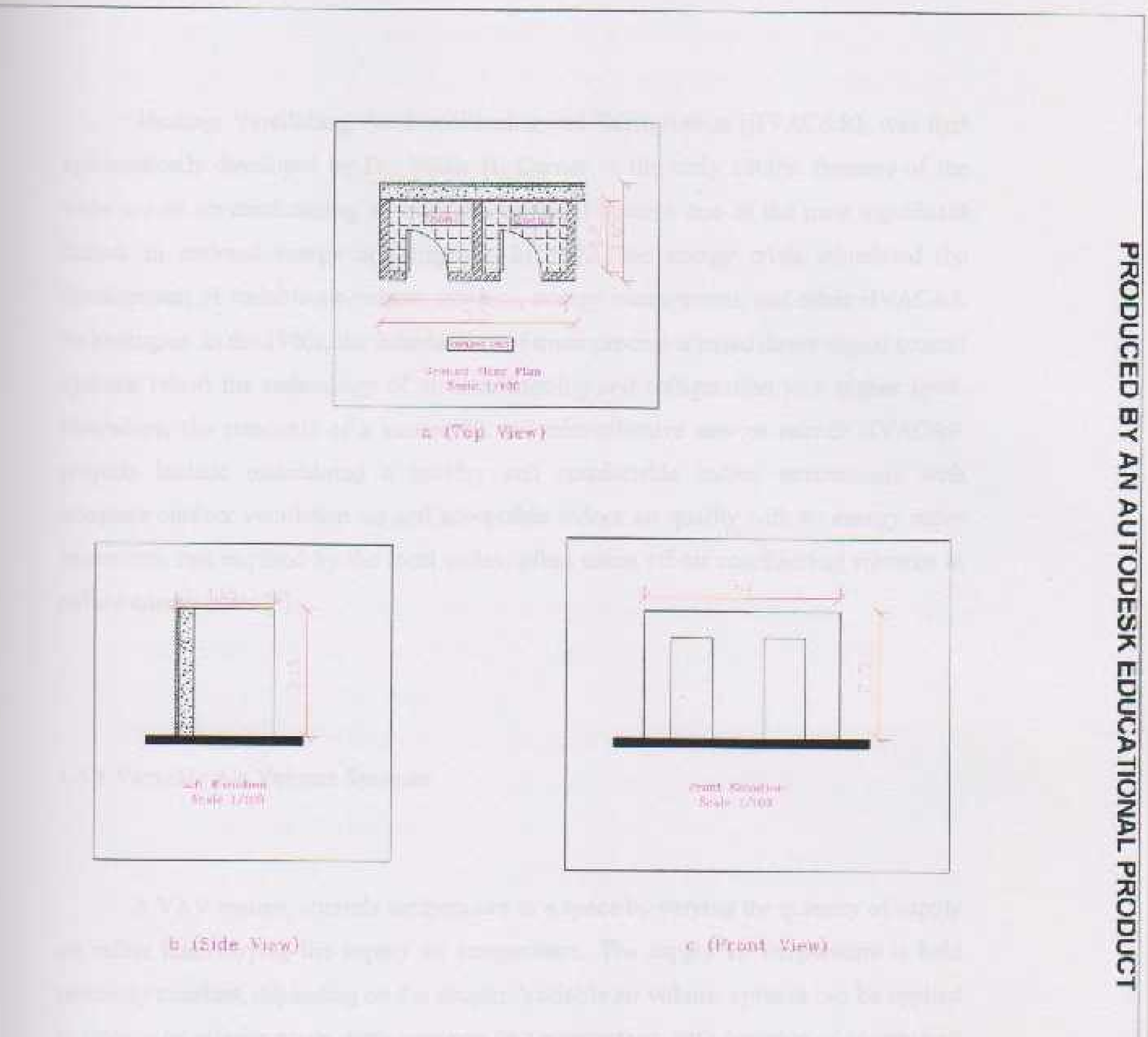


Figure 1.2: Zones Layout

1.4 Literature Review

Heating, Ventilating, Air Conditioning and Refrigeration (HVAC&R), was first systematically developed by Dr. Willis H. Carrier in the early 1900's. Because of the wide use of air conditioning all over the world, it became one of the most significant factors in national energy consumption. In 1973, the energy crisis stimulated the development of variable-air-volume systems, energy management, and other HVAC&R technologies. In the 1980s, the introduction of microprocessor based direct-digital control systems raised the technology of air conditioning and refrigeration to a higher level. Nowadays, the standards of a successful and cost-effective new or retrofit HVAC&R projects include maintaining a healthy and comfortable indoor environment with adequate outdoor ventilation air and acceptable indoor air quality with an energy index lower than that required by the local codes, often using off-air conditioning schemes to reduce energy costs [9].

1.4.1 Variable Air Volume Systems

A VAV system, controls temperature in a space by varying the quantity of supply air rather than varying the supply air temperature. The supply air temperature is held relatively constant, depending on the season. Variable air volume systems can be applied to interior or exterior zones, with common or separate fans, with common or separate air temperature control, and with or without auxiliary heating devices. The greatest energy saving associated with VAV occurs at the exterior zones, where variations in solar load and outside temperature allow the supply air quantity to be reduced. Humidity control is a potential problem with VAV. Particular care should be taken in areas where the sensible heat ratio (ratio of sensible heat to sensible plus latent heat to be removed) is low, such as in conference rooms [9].

The Variable Air Volume systems (VAV) reduce the volume flow rate of supply air at reduced loads instead of varying the supply air temperature as in constant-volume systems. These systems were introduced in the early 1950's and gained wide acceptance after the energy crisis of 1973 as a result of their lower energy consumption in comparison with constant-volume systems. With many variations, VAV systems are in common use for new high-rise office buildings today. Central air conditioning systems always will provide a more precisely controlled, healthy, and safe indoor environment for high-rise buildings, large commercial complexes, and precision manufacturing areas. Compared with a constant-volume system, a VAV system has mainly the following advantages:

- Reduced fan energy use during part-load operation when the supply volume flow rate is reduced.
- A slightly lower or nearly the same zone relative humidity when the supply volume flow rate is reduced during summer cooling mode part-load operation.
- More individual control zones.
- Reduction of the construction cost because of taking into consideration of the supply air volume flow diversity factor instead of the sum of zone peak loads.
- Capability of self-balancing of zone supply volume flow rates.
- Convenience during the relocation of the terminals and space diffusion devices during future expansion or retrofit [9].

1.4.2 Types of Variable Air Volume Systems

Most medium and large size buildings need multi-zone air systems. However, many indoor stadiums, convention centers, factories, residential buildings, and small retail stores employ single-zone air systems. Currently used variable-air-volume systems can be classified into the following types:

- VAV Cooling Systems.
- VAV Reheat Systems.
- Dual-duct VAV Systems.
- Fan-powered VAV Systems.

Nowadays, there is a technical basis for the Fault Detection & Diagnostics (FDD) software which was described in a July 2005 ASHRAE Journal article, "FDD for Air Handling Units (AHUs) and VAV Boxes". The article describes the basis for development of the AHUs Performance Assessment Rules and the VAV Control Chart algorithms [9].

1.4.2.1 VAV Cooling Systems

A VAV cooling system is a multi-zone air system with VAV boxes to modulate the cold supply air volume flow rate to match the variation of zone load in order to maintain a preset zone temperature, as shown by the air system that serves the interior zone in Fig 1.3, VAV cooling systems with VAV boxes only are widely used to serve the interior zone in commercial and industrial buildings and other different applications [9].

1.4.2.2 VAV Box

A VAV box or more specifically a cooling VAV box is a terminal device in which the supply volume flow rate is modulated by varying the opening of the air passage by means of a single-blade butterfly damper, multi-blade damper, or an air valve. A VAV box may have a single outlet or multiple round outlets. A single blade damper VAV box, as shown in Fig 1.4, has a simple construction and is simple to operate. A typical damper is of three conditions, closed, partially opened, or fully opened [9].

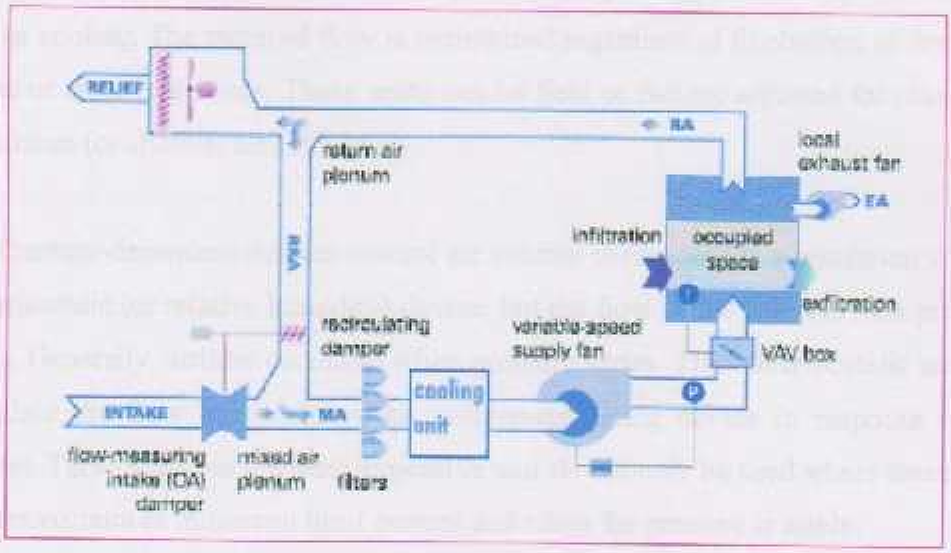


Figure 1.3: VAV Cooling System

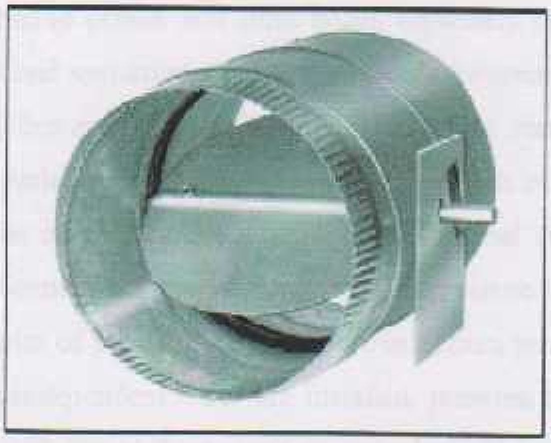


Figure 1.4: VAV Box with Single Blade Damper

1.5 Control of VAV Cooling Systems

Air volume can be controlled by duct-mounted terminal units serving a number of air outlets in a control zone or by units integral to each supply air outlet. Pressure-independent volume regulator units control flow in response to the thermostat's call for heating or cooling. The required flow is maintained regardless of fluctuation of the VAV unit inlet or system pressure. These units can be field or factory adjusted for maximum and minimum (or shutoff) air settings.

Pressure-dependent devices control air volume in response to a maximum volume unit thermostatic (or relative humidity) device, but the flow varies with the inlet pressure variation. Generally, airflow oscillates when pressure varies. These thermostatic units do not regulate the flow but position the volume-regulating device in response to the thermostat. These units are the least expensive and should only be used where there is no need for maximum or minimum limit control and when the pressure is stable.

The type of controls available for VAV units varies with the terminal device. Most use either pneumatic or electric controls and may be either self-powered or system air actuated. System-powered devices use air from the air supplied to the space to power the operator. Components for both control and regulation are usually contained in the terminal device. To conserve power and limit noise, especially in larger systems, fan-operating characteristics and system static pressure should be controlled. Many methods are available, including fan speed control, variable inlet fan control, fan bypass, fan discharge damper, and variable pitch fan control. The location of the pressure-sensing devices depends, to some extent, on the type of VAV terminal used. Where pressure-dependent units without controllers are used, the system pressure sensor should be near the static pressure midpoint of the duct run to ensure minimum pressure variation in the system. Where pressure-independent units are installed, pressure controllers may be at the end of the duct run with the highest static pressure loss. This sensing point ensures maximum fan power savings while maintaining the minimum required pressure at the last terminal. As the flow through the various parts of a large system varies, so do the static pressure losses. Some field adjustment is usually required to find the best location for the pressure sensor. In many systems, the initial location is two-thirds to three-fourths of the distance from the supply fan to the end of the main trunk duct. As the pressure at the system control point increases due to the closing of terminal units, the pressure controller signals the fan controller to position the fan volume control, which reduces flow and maintains constant pressure.

Many present-day systems measure flow rather than pressure and, with the development of economical DDC, each terminal box (if necessary) can be monitored and the supply and return air fans modulated to exactly match the demand [1], [4].

1.6 Multizone Systems

A multizone system supplies several zones from a single, centrally located air handling unit. The multizone system is similar to the dual-duct system. In operation, it has the same potential problem with high humidity levels. Multizone packaged equipment is usually limited to about 12 zones, while built-up systems can include as many zones as can be physically incorporated in the layout. A common variation on the multizone system is called the Texas multizone system or three-deck multizone system. In this system, the hot deck heating coil is removed from the air handler and replaced with an air resistance plate matching the pressure drop of the cooling coil. Individual heating coils are placed only in each perimeter zone duct [1].

1.6.1 Zoning Requirements

Loads vary over time due to changes in the weather, occupancy, activities, and solar exposure. Each space with a different exposure requires a different control zone to maintain constant temperature. Some areas with special requirements may need individual control or individual systems, independent of the rest of the building. Variations in indoor conditions, which are acceptable in one space, may be unacceptable in other areas of the same building. The extent of zoning, the degree of control required in each zone, and the space required for individual zones also narrow the system choices [1].

There are two types of zoning, exterior zoning and interior zoning.

A. Exterior zoning: Exterior zones are affected by varying weather conditions like wind, temperature, sun, and depending on the geographic area and season, require both heating and cooling. This variation gives the engineer considerable flexibility in choosing a system and enables the greatest advantages from VAV systems to be realized. The need for separate perimeter zone heating is determined by:

- Variety of the heating load (i.e., geographic location).
- Nature and orientation of the building envelope.
- Effects of downdraft at windows and the radiant effect of the cold glass surface (i.e., type of glass, area, height, and U-factor).
- Type of occupancy (i.e., sedentary versus transient).
- Operating costs (i.e., in buildings such as offices and schools that are unoccupied for considerable periods). Fan operating cost can be reduced by heating with perimeter radiation during unoccupied periods rather than operating the main supply fans or local unit fans. Separate perimeter heating can operate with any all-air system. However, its greatest

application has been in conjunction with VAV systems for cooling-only service [1].

B. Interior zoning: Interior spaces have relatively constant conditions because they are isolated from external influences. Usually interior spaces require cooling throughout the year. A VAV system has limited energy advantages for interior spaces, but it does provide simple temperature control. Interior spaces with a roof exposure, however, may require similar treatment to perimeter spaces requiring heat [1].

Chapter Two

MATHEMATICAL MODELING

Chapter Two

MATHEMATICAL MODELING

2.1 Overview

This chapter focuses on developing a complete model to represent the VAV Cooling System. Modeling is the process of converting the physical system into mathematical equations to explain its dynamic behavior. In other words a mathematical model of a process as defined by Denn is a system of equations whose solution, given specific input data, is representative of the response of the process to a corresponding set of inputs. Mathematical modeling process starts with understanding the physical system, then the mathematical equations must be derived based on the fundamental theories or laws such as the conservations of mass, energy, and momentum. The mathematical modeling procedure can be summarized by the following steps:

1. Clearly defining the engineering problem and the objectives to be achieved.
2. Valid and logic assumptions are to be made in order to simplify the model.
3. Then the fundamental theories or laws are applied to derive a complete model [5].

2.2 System Mathematical Modeling

As previously mentioned this project aims to design and control the VAV box in an air conditioning system which is composed of two typical rooms. The idea is to control the temperature of each room independently; in addition, each room is supplied with air through a VAV box. Thus, the VAV box (actuated damper) and the room are two subsystems that are representing the overall system or the plant. Fig 2.1 shows the block representation of these subsystems, this representation is valid for the two zones, which means that there are two identical systems.

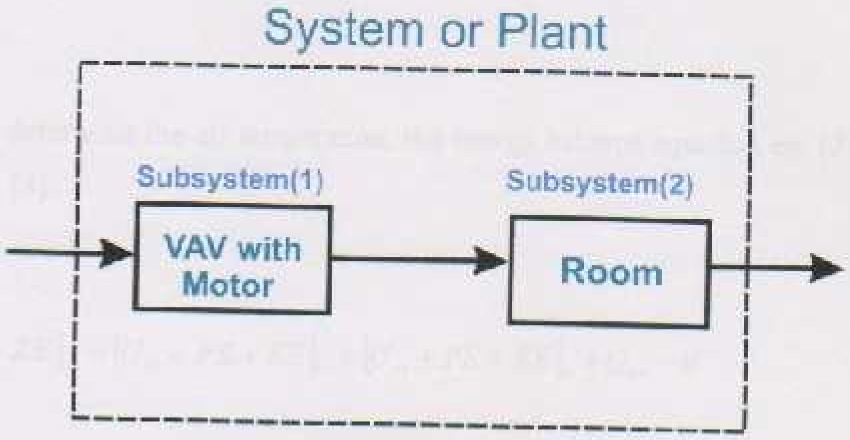


Figure 2.1: Plant Configuration

Now the procedure of deriving mathematical modeling is to be followed in order to convert each block shown in Fig 2.1 into its equivalent equations.

2.2.1 Room Mathematical Model

To derive the model, first it is important to specify the variable which has the dynamic behavior. For the simple room explained in this project, the goal is to control its air temperature, which means that the temperature is the time variable [5], [8].

The model of a simple room will be derived taking the following assumptions under consideration:

1. The room is thermally isolated from the surrounding.
2. Only the temperature of the room air is the controlled variable, this makes it possible to say that the room is Single Input Single Output (SISO) system.
3. Heat gain occurs through the walls, roof, ventilation and infiltration.
4. Effects of kinetic and potential energies are negligible.
5. It is assumed that the initial conditions are zeros, which means that the system has zero deviation at initial steady state.

In order to determine the air temperature, the energy balance equation eq. (2.1) is to be applied [5], [8].

$$[U_{in} + PE + KE]_A = [U_{in} + PE + KE]_i + [U_{in} + PE + KE]_o + Q_{tot} - W \tag{2.1}$$

Where;

KE : Kinetic Energy (W).

PE : Potential Energy (W).

Q_{tot} :The total heat transferred to the system from the surroundings (W).

U_{in} : Internal Energy (W).

W : Output work from the system (W).

$[U_{in} + PE + KE]_A$: Represents the Accumulation of Internal, Potential and Kinetic Energies.

$[U_{in} + PE + KE]_i$: Represents the Accumulation of Internal, Potential and Kinetic Energies in due to convention.

$[U_{in} + PE + KE]_o$: Represents the Accumulation of Internal, Potential and Kinetic Energies out due to convention [5].

Applying eq. (2.1) to the system of the simple room under consideration yields eq. (2.2):

$$\frac{dU_{in}}{dt} = 0 - 0 + Q_{tot} - W \tag{2.2}$$

All kinetic and potential energies go to zero since these energies are not variables in the case of the system of simple room. In addition, the system does not undergo any work production. Then eq. (2.2) reduces into eq. (2.3):

$$\frac{dU_{in}}{dt} = Q_{tot} \tag{2.3}$$

The rate of change of internal energy equals the total heat transferred Q_{tot}, where:

$$Q_{tot} = Q_{in} - Q_{gain} \tag{2.4}$$

Where;

Q_{in} : Air conditioning input energy (W).

Q_{gain} : The energy gained through walls, roof, and ventilation (W).

The energy gain Q_{gain} contains the thermal load from walls and roofs which can be expressed by $\sum AUi$, and the ventilation coefficient $\frac{n_v \upsilon_r}{3}$, this yields:

$$Q_{gain} = \left(\sum (AUi) + \frac{n_v \upsilon_r}{3} \right) T(t) \tag{2.5}$$

Where;

$T(t)$: Temperature as a function of time (°C).

$\sum AUi$: Area integrated fabric surface U-value (W/m²°C).

$\frac{n_v \upsilon_r}{3}$: Ventilation coefficient m^3 / h in which:

n_v : Ventilation air change rate per hour.

υ_r : Room volume (m³).

Substituting eq. (2.5) in (2.4) results in;

$$Q_{net} = Q_{in} - \left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right) T(t) \tag{2.6}$$

On the other hand, the change of internal energy is defined as:

$$\frac{dU_{in}}{dt} = \rho \upsilon_r c_v \frac{dT}{dt} \tag{2.7}$$

Where;

ρ : Air density (Kg/m³).

c_v : Specific heat at constant volume (cal/Kg °C).

It is assumed that ρ , υ_r, and c_v are constants with the variation of time t and temperature T, so a linear relationship between the internal energy and temperature expressed by eq. (2.7) is achieved.

Substituting eqs. (2.6) and (2.7) in to eq. (2.3) yields:

$$\rho \upsilon_r c_v \frac{dT}{dt} = Q_{in} - \left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right) T(t) \tag{2.8}$$

The transfer function of the air conditioned room is a representation of the relation between input and output, so it can be evaluated by dividing the output temperature over the input refrigeration energy all as a function of time. Now rearranging eq. (2.8):

$$\upsilon_r \rho c_v \frac{dT}{dt} + \left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right) T(t) = Q_{in}(t) \tag{2.9}$$

Eq. (2.9) is a first order differential equation, taking the Laplace transform, knowing that the initial conditions are zeros:

$$\upsilon_r \rho c_v T(s) s + \left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right) T(s) = Q_{in}(s) \tag{2.10}$$

The transfer function of the room $G_r(s)$ will be:

$$G_r(s) = \frac{1}{(\upsilon_r \rho c_v) s + \left(\sum (AU_i) + \frac{n_r \upsilon_r}{3} \right)} \tag{2.11}$$

To simplify the eq. (2.11), it can be written as:

$$G_r(s) = \frac{K_r}{(\tau_r s + 1)} \tag{2.12}$$

Where;

τ_r = Time constant of the room (s).

K_r = The room temperature gain to fluctuations in process output

$$\tau_r = \frac{\upsilon_r \rho c_p}{\left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right)} \tag{2.13}$$

$$K_r = \frac{1}{\left(\sum (AUi) + \frac{n_r \upsilon_r}{3} \right)} \tag{2.14}$$

[8].

2.2.2 Modeling of DC Motor

It is mentioned that the VAV Box has an actuator to rotate the damper to a specific position; this actuator may be a hydraulic motor, or a DC motor, the application of the room and the needed accuracy affect choosing the type of the actuator. In this project the DC motor has been selected. The construction of the DC motor is shown in Fig 2.2 (a); it is an electromechanical component which generates a displacement for the input voltage. Fig 2.2 (b) is a block representation to the DC motor showing the input voltage $E_a(s)$ and output angular displacement $\theta_m(s)$ all are expressed in s domain.

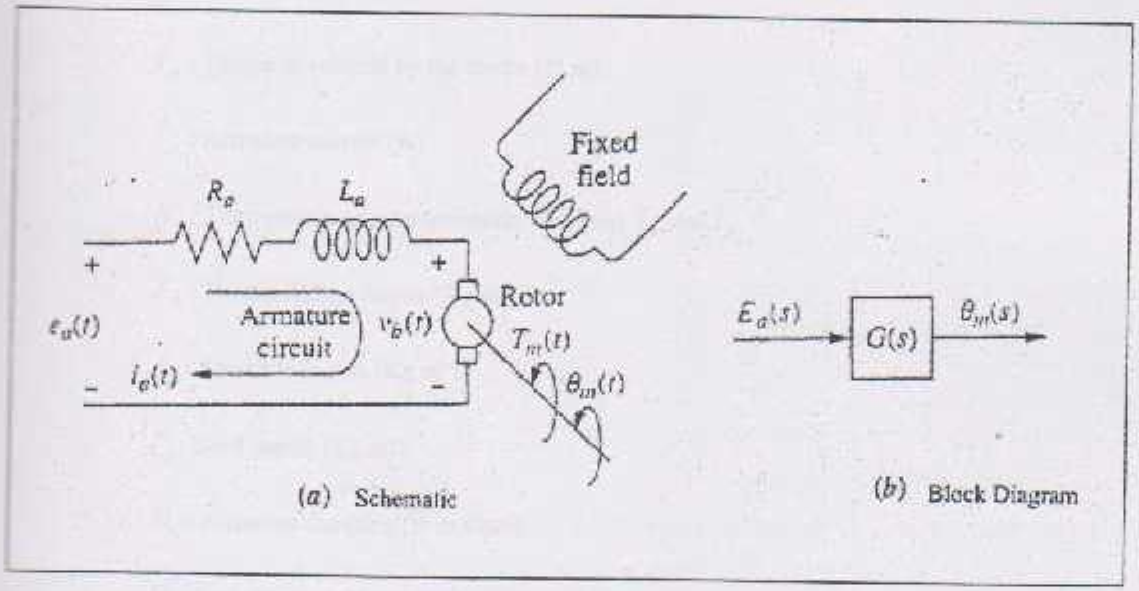


Figure 2.2: DC Motor

The transfer function of the DC motor is given by the following eq.

$$G_m(s)=\frac{K}{(s+a)s} \tag{2.15}$$

Where:

$$K = \frac{K_t}{R_a J_m} \tag{2.15.1}$$

$$\alpha = \frac{1}{J_m}\left(D_m + \frac{K_t K_b}{R_a} \right) \tag{2.15.2}$$

$$K_t = \frac{T_m}{I_a} \tag{2.15.3}$$

$$J_m = J_a + J_l \left(\frac{N_1}{N_2} \right)^2 \tag{2.15.4}$$

$$D_m = D_a + D_l \left(\frac{N_1}{N_2} \right)^2 \tag{2.15.5}$$

T_m : Torque developed by the motor (N.m).

I_a : Armature current (A).

K_t : The constant of proportionality between T_m and I_a.

J_m : The equivalent inertia (Kg.m²).

J_a : Armature inertia (Kg.m²).

J_l : Load inertia (Kg.m²).

D_a : Armature damping (N.m.s/rad).

D_l : load damping (N.m.s/rad).

D_m : Equivalent damping (N.m.s/rad).

As shown in eq. (2.15). If all the parameters contained in the eqs. (2.15.1) to (2.15.5) are measured, it is possible to substitute their values into eq. (2.15) then the transfer function of the motor will be obtained. But the fact that it is not possible to calculate these parameters since the motor is a closed system and the component parts are not easily identifiable. Therefore, the transfer function of the motor could be obtained experimentally [7].

Experiment Procedure: The DC Motor is given an input voltage of 12V, and the response of the motor speed was captured on the oscilloscope, the system is first order and the values of K and α were determined from the response settling time (T_{se}) which is the time required for the response to reach and steady with ±2 % of the steady-state value, and the final value [7].

$$T_{se} = \frac{4}{\alpha}$$

And from the final value theorem:

$$\text{Final Value} = \lim_{s \to 0} s\,\text{input}(s) G(s)$$

Now substituting the input voltage into final value theorem and considering the measured settling time yield:

$$\alpha = 20$$
$$K = 0.0466$$

The motor transfer function which is found experimentally is between the input voltage and output speed where the output speed is affected by the gear ratio, and then it must be multiplied by an integrator to be representative for the input voltage and output angular velocity, then:

$$G_m(s) = \frac{0.0466}{s(s+20)} \tag{2.16}$$

Chapter Three

THERMAL ANALYSIS AND DESIGN

Chapter Three

THAERMAL ANALYSIS AND DESIGN

3.1 Overview

In this chapter the main basis of heat transfer and thermodynamics are discussed, also, it includes simple explanation of the cooling load calculations methodology and the actual calculations applied in the project, and duct description for the system.

3.2 Principle of Heat Transfer

. Heat transfer is defined to be the transfer between two control volumes distinct with temperature gradient, such that heat will move from high temperature side to the lower temperature side. This science provided a significant tool for determining the rate of heat gain that the air conditioned envelopes experience, and so determining the required capacity for air conditioning to overcome this heat gain.

3.2.1 Conductive Heat Transfer

The conductive heat transfer is experienced by solid components of the building, such as walls, roofs, and doors. For one-dimensional steady-state heat conduction, Fourier's law gives the following relationship:

$$q_c = -KA\frac{dT}{dx} \tag{3.1}$$

Where;

q_c : Heat gain by conduction (W).

K : Thermal conductivity (W/m°C).

A : Cross-sectional area normal to heat flow (m²).

T : Temperature (°C).

x : Coordinate dimension along heat flow (m).

Eq. (3.1) shows that the rate of heat transferred is directly proportional to the temperature gradient dT/dx, the thermal conductivity K, and the cross-sectional area A. The minus sign indicates that the heat must flow in the direction of decreasing temperature; for steady-state heat conduction through a plane composite wall with perfect thermal contact between each layer, as shown in Fig 3.1, the rate of heat transfer through each section of the composite wall must be the same.

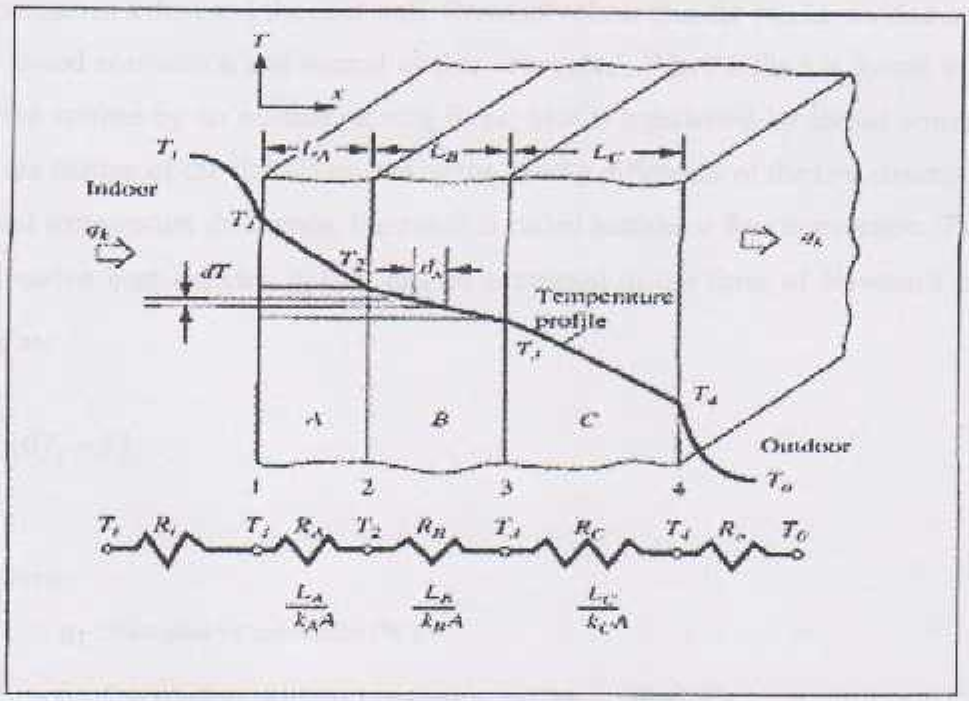


Figure 3.1: Steady-States One-Dimensional Heat Conduction through a Composite Wall.

From Fourier's law of conduction:

$$q_c = \frac{K_A A}{L_A}(T_1 - T_2) = \frac{K_B A}{L_B}(T_2 - T_3) = \frac{K_C A}{L_C}(T_3 - T_4) \tag{3.2}$$

Where;

L_A, L_B, L_C :The thicknesses of layers A, B, and C respectively of the composite wall (m).

T_i : The temperature at surface i where $i = 1, 2, 3, 4$ (°C).

K_A, K_B, K_C : The thermal conductivity of layers A, B, and C respectively of the composite wall (W/m°C).

3.2.2 Convection Heat Transfer

Convection is the second way for heat transfer that occurs in fluids, when a fluid contacts a surface at a different temperature, such as the heat transfer happening between the airstream in a duct and the duct wall. Convective heat transfer can be divided into two types: forced convection and natural or free convection. When a fluid is forced to move along the surface by an outside moving force, heat is transferred by forced convection. When the motion of the fluid is caused by the density difference of the two streams in the fluid and temperature difference, the result is called natural or free convection. The rate of convective heat transfer q_h (w), can be expressed in the form of Newton's law of cooling as:

$$q_h = h_c A(T_s - T) \tag{3.2}$$

Where;

q_h : Heat gain by convection (W).

h_c : The average convective heat-transfer coefficient, (W/m².°C).

T_s : The surface temperature (°C).

T : The temperature of fluid away from surface (°C).

The convective heat-transfer coefficient h_c is usually determined empirically.

3.2.3 Radiant Heat Transfer

In radiant heat transfer, heat is transferred in the form of electromagnetic waves traveling at the speed of light. The net rate of radiant transfer q_r (W), between a gray body at absolute temperature T_{R1} and a black surrounding enclosure at absolute temperature T_{R2} (for example, the approximate radiation exchange between occupant and surroundings in a conditioned space) can be calculated as:

$$q_r = \sigma A_1 E_1 \left(T_{R1}^4 - T_{R2}^4\right) \tag{3.3}$$

Where;

q_r : Radiation heat transfer coefficient (W).

$\sigma = 0.1714 * 10^{-8}$ (W/m^2.K^4) : Stefan-Boltzmann constant.

A_1 : Area of gray body (m^2).

E_1 : Emissivity of surface of gray body.

T_{R1}, T_{R2} : Absolute temperatures of surface 1 and 2 (K).

3.2.4 Overall Heat Transfer U_i

In actual application, many calculations of heat-transfer rate are combinations of conduction, convection, and radiation. Consider the composite wall shown in Fig 3.1; in addition to the conduction through the wall, convection and radiation occur at inside and outside surfaces 1 and 4 of the composite wall. At the inside surface of the composite wall, the rate of heat transfer q_i (W), consists of convective heat transfer between fluid, the air, and solid surface q_c (W) and the radiant heat transfer q_r (W), as follows:

$$q_i = q_c + q_r = h_c A_1 (T_i - T_1) + h_r A_1 (T_i - T_1) \tag{3.4}$$

Eq. (3.4) can be rewritten as:

$$q_i = h_i A_1 (T_i - T_1) \tag{3.5}$$

Where;

T_i : Indoor temperature (°C).

h_i : The inside surface heat-transfer coefficient at the liquid-to-solid interface (W/m²°C).

Similarly, at the outside surface of the composite wall, the rate of heat-transfer q_0 (W) is given by:

$$q_0 = h_0 A_4 (T_4 - T_0) \tag{3.6}$$

Where;

h_0 : Outside surface heat-transfer coefficient at fluid-to-solid interface (W/m²°C).

A_4 : Area of surface 4 (m²).

T_4 : Temperature of surface 4 (°C).

T_0 : Outdoor temperature (°C).

For one-dimensional steady-state heat transfer, the overall heat-transfer rate of the composite, it is considered that $q = q_i = q_o$.

Therefore:

$$Q_{in} = UA(T_i - T_o) \tag{3.7}$$

Where:

U : Overall heat-transfer coefficient, often called the U value (W/m²°C)

A : Surface area perpendicular to heat flow (m²).

Therefore, the overall heat transfer coefficient U is given as:

$$U = \frac{1}{1/h_i + L_A/K_A + L_B/K_B + L_C/K_C + 1/h_o} \tag{3.8}$$

[9].

3.3 Cooling Load Components

- Space Cooling Load is the rate at which heat must be removed from the space to maintain a constant space air temperature, cooling loads can be usually classified into two categories: external cooling loads and internal cooling loads.

A. External Cooling Loads: These loads are generated due to the heat gains in the conditioned space from external sources through the building envelope or building shell and the separation walls.

B. Internal Cooling Loads: These loads are generated due to the releasing sensible and latent heat from heat sources inside the conditioned space.

Cooling load calculation, which involves determining the building's entire heat gain component, is complex since there are many more factors involved in heat gain than in the heat loss, and heat gain varies sharply during the day.

The total cooling load of a structure involves:

1) Heat gain through walls and ceiling.
2) Heat gain through windows, doors.
3) Sensible and latent heat gain due to infiltration and ventilation.
4) Sensible and latent heat gain due to occupancy.
5) Heat gain due to equipment, lights, motors, computers ....etc [3].

3.4 Cooling Load Calculation

Cooling load calculations for air conditioning system design are mainly used to determine the volume flow rate of the conditioned air as the refrigeration load of the equipment in order to determine the HVAC proper system.

The required calculations for this project will be explained in the following sections.

3.4.1 Heat Gain through Walls and Roofs

The heat gain through an outside wall and roof is a combination of the direct solar radiation, conduction/convection heat gain and thermal lag effects. The combined effect is expressed in a factor called Cooling Load Temperature Difference (CLTD).

The heat gain through walls and roofs can be calculated by:

$$Q = UA(CLTD) \tag{3.9}$$

Where;

Q : Cooling load (W).

U : Overall heat-transfer coefficient of exterior wall or roof (W/m²°C).

A : Area of exterior wall, roof (m²).

$CLTD$: Cooling Load Temperature Difference (°C).

Also, $CLTD$ is given by the following eq:

$$CLTD = (cltd + LM)K_c + (25.05 - T_i) + (T_{o,m} - 29.4)f \quad (3.10)$$

Where:

LM : Is the latitude correction factor and can be obtained from the table for roofs and walls which is shown in Appendix (B).

K_c : Is the color correction factor, and it has a different values as shown in Table 3.1.

f : Is the attic roof factor and its values are shown in Table 3.2.

$T_{o,m}$: Mean outside temperature (°C).

Table 3.1: Color Correction Factors

Color	Color Correction Value (*Kc*)
Dark colored roof and walls	1.0
Permanently light colored roof	0.5
Permanent medium color wall	0.83
Permanent light color walls	0.65

Table 3.2: The Attic Roof Factor

Condition	Attic roof factor (f)
Flat roof and walls	1.0
Attic	0.75

The mean outside temperature is given by the following equation:

$$T_{o,m} = T_o - DR/2 \tag{3.11}$$

Where:

DR: Daily Range that equals to the difference between the average maximum and the average minimum temperatures for the warmest month of the summer season (°C).

Considering the rooms and applying energy equations to find the heat gain. Fig 3.2 is a schematic diagram in which the walls have a symbol to simplify the calculations.

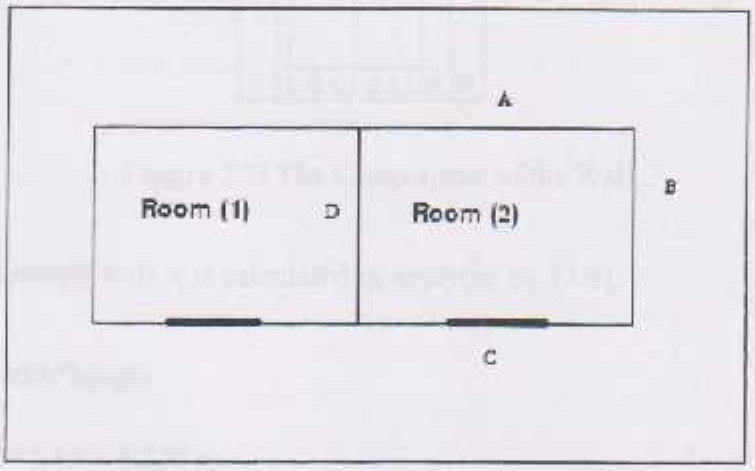


Figure 3.2: Rooms Schematic Diagram

A. Heat gain through walls:

Each of the two rooms has four walls each wall with different characteristics, and different construction material, so the calculation is to be made for the first room, and the other will be the same.

Wall (A):

Wall (A) is the original Lab's wall consists of regular building material as shown in the Fig 3.3.

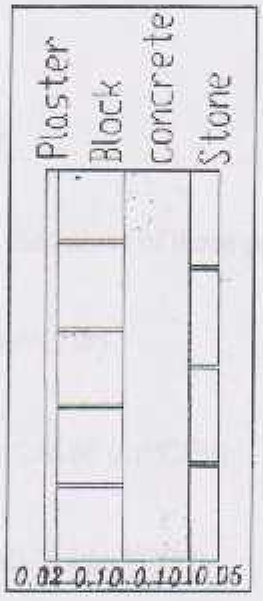


Figure 3.3: The Components of the Wall

The heat gain through wall A is calculated by applying eq. (3.8):

Area = width*height

= 1.5*2.15 = 3.225 m^2

Determining the value of U:

Applying eq. (3.8) to calculate U yields:

$$U = \frac{1}{\sum R_T} = \frac{1}{R_1 + R_2 + R_3 + R_4 + R_5 + R_6}$$

$$U = \frac{1}{\frac{1}{h_{in}} + \frac{\delta_2}{k_2} + \frac{\delta_3}{k_1} + \frac{\delta_4}{k_5} + \frac{\delta_5}{k_4} + \frac{1}{h_{out}}}$$

Where

δ : The thickness of layer (m).

R : Thermal Resistance of layer (°C/W).

From the tables in Appendix (B) the values of these parameters are as follows:

$R_1 = 1/h_{in} = 1/0.12 = 8.33$ (m²°C/W)

$R_2 = \Delta x/k_{plaster} = 0.02/1.20 = 0.0166$ (m²°C/W)

$R_3 = \Delta x/k_{block} = 0.10/0.14 = 0.714$ (m²°C/W)

$R_4 = \Delta x/k_{concrete} = 0.10/1.75 = 0.05714$ (m²°C/W)

$R_5 = \Delta x/k_{stone} = 0.05/2.20 = 0.0227$ (m²°C/W)

$R_6 = 1/h_{out} = 1/0.03 = 33.33$ (m²°C/W)

$R_{tot} = 42.47$ (m²°C/W)

$U = 1/R_{tot} = 0.235$ W/m²°C

Determining the value of *CLTD*:

Applying eq (3.10) results:

$$CLTD = (5 - 0.5)0.83 + (25.5 - 23) + (38.8 - 29.4)1$$

$$CLTD = 15.635\ °C$$

Substituting the value of heat gain and *CLTD* into eq. (3.11), and the result will be as the following:

$$Q = 3.225 * 0.235 * 15.635 = 11.849\ W$$

B. Heat gain through ceiling:

The ceiling is false ceiling in the ground floor

Area = 1.5*1.2 = 1.8 m².

$U = 1/R$ = 0.01/0.12 = 0.0833

$R_{in} = 1/h_{in}$ = 1/0.15 = 6.66

$R_{out} = 1/h_{out}$ = 1/0.07 = 14.2857

U = 0.047 W/m²C

ΔT = 13

$Q = AU(\Delta T)$ = 1.8*0.047*13

= 1.0998 W

Applying the same analysis to find the heat gain for each wall, but the differences occur are due to the different materials and thicknesses. Thus, the total heat gain through the walls and the ceiling (for the first room) = 61.471+ 1.0998 = 62.5708 W.

Table 3.3 summarizes the results of heat gains through walls and ceiling. As a result of the symmetry of the two rooms, these calculations will be also the same

Table 3.3: The Heat Gains Due to Walls and Ceiling of the Room

Zone(1/2)	Area (m^2)	*U* ($W/m^2.°C$)	*Ti-To* (°C)	*Q gain* (W)
A	3.225	0.235	15.635	11.849
B	2.58	0.3649	13	12.238
C	3.225	0.8917	13	37.384
Ceiling	1.8	0.047	13	1.0998
				62.5708

It is noticed that no calculations are made related to wall D, because this wall is assumed to be under the condition of $\Delta\ T=0$, this is due to the same designing temperature for the two rooms [3].

3.5 Ventilation

Buildings in which people live and work must be ventilated to replenish oxygen, dilute the concentration of carbon dioxide and water vapor, and minimize unpleasant odors. A certain amount of air movement or ventilation ordinarily is provided by air leakage through small cracks in the building's walls, especially around windows and doors. Engineers estimate that for adequate ventilation the air in a room should be changed completely from one and a half to three times each hour, or that about 280 to 850 liters (about 10 to 30 cu ft) of outside air per minute should be supplied for each occupant. Providing this amount of ventilation usually requires mechanical devices to augment the natural flow of air.

Simple ventilation devices include fans or blowers that are arranged either to exhaust the still air from the building or to force fresh air into the building, or both. Ventilating systems may be combined with heaters, filters, humidity controls, or cooling devices. Many systems include heat exchangers. These use outgoing air to heat or cool incoming air, thereby increasing the efficiency of the system by reducing the amount of energy needed to operate it [1].

In this project the natural ventilation is to be used by making certain space at each door of the zones, these spaces should be calculated to give three time air changing. Therefore, the ventilation coefficient which was explained in section 2.2.1 will be as the following:

$$\frac{n_r v_r}{3} = 3.5475 \text{ W/°C} \ [8].$$

3.6 Duct Design

Good duct design involves sizing the ducts, determining the pressure losses, calculating the noise levels, determining the out of balance pressures and optimizing this against the total cost of the system.

Design procedure:

1) Determine the air volume requirements and room size.

2) Select the tentative oulet type and location within room.

3) Determine the room's characteristic.

4) Select the appropriate outlet size.

5) Ensure that this outlet meets other imposed specifications such as noise and static pressure.

Duct design methods for HVAC systems and for exhaust systems conveying vapors, gases, and smoke are the equal friction method, the static regain method, and the T-method. Equal friction and static regain are non optimizing methods, while the T-method is a practical optimization method. To ensure that system designs are acoustically acceptable, noise generation should be analyzed and sound attenuators and/or acoustically lined duct provided where necessary [3], [6].

3.6.1 Equal Friction Method

In the equal friction method, ducts are sized for a constant pressure loss per unit length. When energy cost is high and installed ductwork cost is low, a low friction rate design is more economical. For low energy cost and high duct cost, a higher friction rate is more economical. After initial sizing, calculate the total pressure loss for all duct sections, and then resize sections to balance pressure losses at each junction.

3.6.2 Static Regain Method

The objective of the static regain method is to obtain the same static pressure at diverging flow junctions by changing downstream duct sizes.

VAV systems are supposed to be in balance at design loads but not part-load conditions, because there is no single critical path in VAV systems. The critical path is dynamic and continually changing as loads on a building change. In general, balancing dampers are not needed for systems designed by the static regain or T-method, because these design methods are self-balancing at design loads and VAV boxes compensate for inaccuracy in fitting data or data inaccuracy caused by close-coupled fittings (at design loads) and system pressure variation (at part loads). Balancing dampers, however, are required for systems designed using the non-self-balancing equal friction method. For systems designed using any method, dampers should not be installed in the inlets to VAV boxes. For any design method, VAV terminal units may have static pressures upstream higher than for which the box is rated, thus possibly introducing noise into occupied spaces. In these cases, control algorithms can poll the VAV boxes and drive the duct static pressure to the minimum set point required to keep at least one unit at starvation (open) at any given time. The upstream static pressure should always be kept at a minimum that is easy for the VAV box to control. Because there may be large differences in static pressure at riser takeoffs serving many floors from a single air handler, manual dampers should be provided at each floor takeoff so that testing, adjusting, and balancing (TAB) contractors can field-adjust them after construction. Alternatively, these takeoff dampers could also be dynamically controlled to adjust the downstream static pressure applied to the VAV boxes, while simultaneously driving the air handler to the lowest possible static pressure set point. Silencers downstream of VAV terminal units should not be necessary if the VAV box damper is operating at nearly open conditions. Their use in this location should be based on careful acoustical analysis, because silencers add total pressure to the system and therefore create more system noise by causing air handlers to operate at higher speeds for a given airflow [3], [6].

3.7 Damper Design

As previously mentioned in section 1.4.2.2, the VAV box supplies the room with a specific air flow which meets the thermal load of the room, this amount is varied according to the variation of room cooling loads, therefore, the opening of the damper will be modulated as a result of the signal given by the thermostat. Fig 3.4 shows the side view of the duct where the VAV Box is placed, the designer may assume that the maximum load requires the maximum opening of damper of 90° and the same about the minimum load which may requires 0°, but the fact that the maximum and minimum opening must not be assumed arbitrarily, because this depends on the conditioned space and the assumptions that are put to solve the problem, also the designer has to select these critical conditions (maximum load, minimum loads, and the disturbances that may occur) before designing the duct and so the damper, this is important for ensuring that the needed position will be verified at any load and within the safe conditions [9].

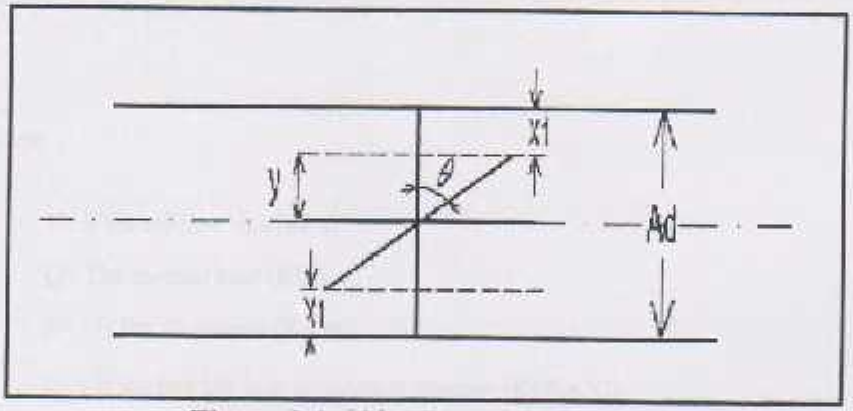


Figure 3.4: Side View of the Damper

The two zones are supposed to be of the same thermal conditions, thus the maximum thermal load of the two zones are the same, therefore, when the maximum thermal load is applied on the two zones at the same time and because the two secondary ducts are of the same diameter, the air flow of the main duct will be divided by 2. At the maximum load, the needed air flow requires a specific area that can be manipulated each time. And the air speed will be the same along the operation periods, since the air flow must be verified through the variation of the area which is provided by the modulation of damper position, this means that the speed must be constant, also the speed must be comfortable to the occupants, or generally suitable according to the application of the conditioned space itself [9].

3.7.1 Maximum and Minimum Opening of the Damper

It is mentioned that the opening of the damper is varied as a result of thermal load variation during the operation time. And there is a direct relationship between the thermal load and air flow which is given by eq. (3.14):

$$\dot{v} = \frac{Q}{\rho c_p \Delta T} \tag{3.14}$$

Where;

$\dot{V}$: is the air flow rate (m^3/s).

Q : The thermal load (KW).

ρ : Is the air density (Kg/m^3).

c_p : Is the specific heat at constant pressure (KJ/Kg.°C).

ΔT : Is the temperature difference between outdoor and the design temperature (°C).

The thermal load is due to the structure and assumed occupancy is 0.5 *KW*, also it was mentioned that this is the minimum possible thermal load. To measure the amount of air flow that meets this load, eq. (3.14) is applied and the result is of 0.032 m^3/s, and it is noticed that the air flow supplied by the main duct equals to 0.2 m^3/s* which will be distributed equally into two ducts of the same diameter, and because the fan speed is always constant, the maximum air flow which may enter the room is 0.1 m^3/s, this means that it is possible to meet the thermal load existed and also it is possible to raise the

---

* The maximum air flow rate of (0.2 m^3/s) has been measured using the flow meter.

thermal load to a certain value that requires the value of 0.1 m^3/s which could be considered as the maximum possible thermal load.

In this section, the maximum and minimum openings of the damper are to be measured according to the maximum and minimum possible thermal loads. In the Figure 3.4, the idea is to find the relationship between the required opening of the damper θ (needed area provided by the opening of the damper which is needed to meet the thermal load applied on the zone) and the airflow rate.

From the geometry shown in Fig 3.4:

$$X1 = \left(\frac{A_d}{2} - \frac{A_d}{2}\cos\theta\right) \tag{3.15}$$

Where;

$X1$: is the required area divided by 2.

A_d: The cross sectional area of the duct.

θ : The angle of the damper which provides with the required area to meet a specific thermal load.

From the fundamentals of fluid mechanics and thermodynamics:

$$\dot{v} = VA_d \tag{3.16}$$

Where;

V : is the air velocity (m/s).

A_d : is the cross sectional area of the pipe or duct (m^2).

And from eq. (3.16), the required area ($2X1$) can be expressed as in the eq. (3.17):

$$2X1 = \frac{\dot{v}}{V} \tag{3.17}$$

The air velocity V must be constant during the operation, in the case explained here, this speed equals to $6m/s$* and it will be the constant of proportionality between the air flow rate and the modulated area.

Substituting the air speed into eq. (3.17):

$$2X1 = \frac{\dot{v}}{6} \tag{3.18}$$

The supply duct has been selected to be of a standard diameter of 8", which means that the area of the duct A_d will be $0.0324\ m^2$.

Now, the relationship between θ and $\dot{v}$ will be verified by substituting eq. (3.18) into (3.15) with the value of A_d, rearranging the equation yields:

$$\theta = \cos^{-1}\left(1 - \frac{\dot{v}}{0.1944}\right) \tag{3.19}$$

To find the minimum and maximum opening of the dampers which are the limitations of

the damper opening, it is possible to substitute the maximum $\dot{v}_{max}$ into eq. (3.19), then the maximum angle will result.

---

* The air velocity of 6m/s has been measured using the speed meter.

$\dot{v}_{max}$ =0.1 m³/s, thus θ_{max} =60.01°.Which is the maximum opening of the damper.

And for the minimum opening:

$\dot{v}_{min}$ = 0.032 m³/s, thus θ_{min} =33.24 °.Which is the minimum opening of the damper.

3.7.2 Summary

In this chapter, the thermal analysis was determined following the basis explained, some of these calculations are necessary to be substituted into the mathematical modeling equations which were derived previously in chapter two and the others will be needed in next chapters. Table 3.4 summarizes the values of the parameters that have been measured besides the other constants of the air properties.

Table 3.4: The Design Parameters

Constant (Parameter)	Description	Value
ρ	Air Density	1.205 Kg/m^3
c_p	Specific Heat at Constant Pressure	1.005 $kJ\ /kg.C^{o}$
$\sum AU_i$	Area integrated fabric surface U-value	5.421049 $W/^{o}C$
ΔT	Temperature Difference Between outdoor and the design temperature	13 C^{o}
n_v	Ventilation Air Change Rate	3(*1/h*)
υ_r	Room Volume	3.5475 m^3
$\frac{n_v \upsilon_r}{3}$	Ventilation Coefficient	3.5475 m^3/h

Chapter Four

CONTROL SYSTEM DESIGN

Chapter Four
CONTROL SYSTEM DESIGN

4.1Overview

Control can be defined as the starting and stopping or modulation of process to regulate the condition being changed by the process. Control in a HVAC system is needed to regulate the amount of heating or cooling necessary to meet the required conditions.

4.2 Control Components in VAV System

Control of VAV System as shown in Fig 4.1 consists of the following components:

Setpoint: it is a desired temperature in summer mode needed in all situations of thermal load variations and it considered 23°C.

Microcontroller: contains a set of functions in order to take the necessary decision to control the process. These functions should implement all system parameters and variables.

Process: is any physical action that should be done by the damper (actuators) in order to supply the room with the suitable energy that should be compensating thermal load variations to achieve the desired output.

Sensor: a device that needed to read the temperature in each zone in order to provide a feedback for control system [2], [3], [4].

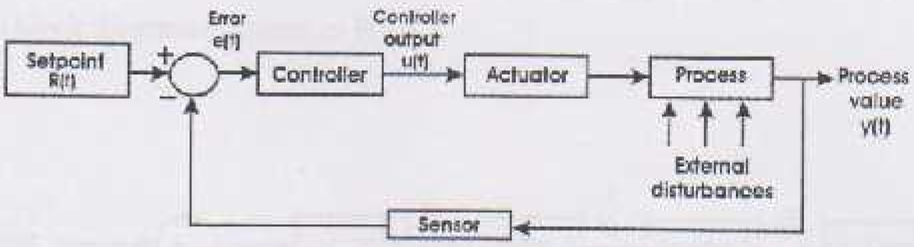


Figure 4.1: Feedback Control System

It should be noticed that this project aims to control the temperature of two rooms by modulating the amount of air flow enter the two rooms in order to reduce the energy consumed more than to control Settling Time; which is the time required for the response to reach and steady with ±2 % of the steady-state value, and other step response specifications.

As a result of modulating one of or both two dampers, the air flow in the main duct will be varied during the operation; this is an excess air which must be removed. Therefore, a third damper is required to allow that. Thus overall system is composed of three control loops, two for the temperature control of the two rooms and the third is to control the opening of exhausted air.

4.3 Room Temperature Control Loop

The two subsystems that was explained in section 2.2 are representative for the hardware of the room temperature control loop, the understanding of the input and output of each subsystem affects the way these subsystems will be related to each other, as shown previously in Fig 2.1, the output of the first subsystem (motor) is an angular velocity while the input to the second subsystem (room) is the input energy, thus the relationship between these two variables must be achieved in order to validate the control loop. These connections between subsystems are derived from the physical system itself. The temperature control loop block diagram is shown in Fig 4.2.

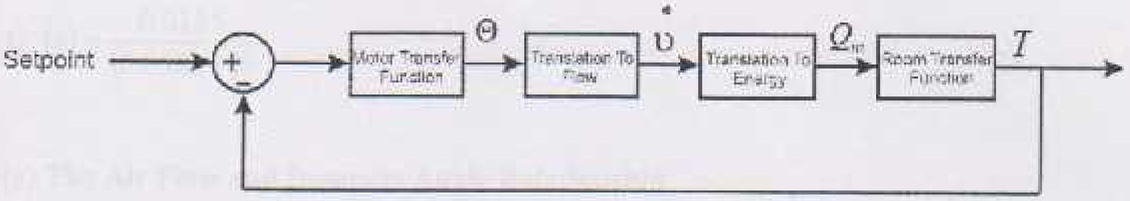


Figure 4.2: The Temperature Control Loop Block Diagram

4.4 Block Diagram Implementation

To implement each block shown in Fig 4.2, the equivalent mathematical model will be replaced by.

(a) Motor transfer function

Motor transfer function between the input voltage and the output angle was found experimentally in the Control System Lab:

$$G_m(s) = \frac{0.0466}{s\ (s+20)} \tag{4.1}$$

(b) Room transfer function

As previously explained in chapter 2, the room transfer function between the air conditioning energy and the room temperature is:

$$G_r(s)=\frac{K_r}{(\tau_r s+1)} \tag{4.2}$$

The values of K_r and τ_r has been evaluated in chapter three, then the transfer function of each room will be shown in eq. (4.3):

$$G_r(s)=\frac{0.0135}{(0.058s+1)} \tag{4.3}$$

(c) The Air Flow and Dampers Angle Relationship

The relationship between the air flow rate $\dot{\upsilon}$ that enters the room and the position of the damper can be obtained by rearranging eq. (3.19) which was discussed in chapter 3:

$$\dot{\upsilon}=0.1944(1-\cos\theta) \tag{4.4}$$

Eq. (4.4) is a nonlinear relationship, to linearize the equation; an Excel Sheet can derive the linear relationship between two variables by getting a set of values as shown in Table 4.1, applying for the values of Table 4.1, eq. (4.4) will be expressed in the form of eq. (4.5):

$$\theta=388.6\dot{\upsilon}+1.12 \tag{4.5}$$

Then:

$$\dot{\upsilon}=\frac{\theta-1.12}{388.6} \tag{4.6}$$

Table 4.1: Damper angle variation between minimum and maximum air flow rate

The relationship was founded as follows:

$\dot{\upsilon}\left(m^3/s\right)$	$A\left(m^2\times10^{-3}\right)$	θ°
0.0320	5.3	33.23
0.0330	5.5	33.88
0.0380	6.3	36.31
0.0410	6.8	37.8
0.0440	7.3	39.22
0.0480	8.0	41.14
0.0500	8.3	41.94
0.0540	9.0	43.76
0.0570	9.5	45.03
0.0600	10.0	46.26
0.0640	11.0	48.66
0.0667	11.1	48.89
0.0699	11.65	50.18
0.0730	12.16	51.34
0.0762	12.7	52.55
0.0794	13.2	53.66
0.0826	13.77	54.9
0.0858	14.3	56.04
0.0889	14.8	57.09
0.0921	15.35	58.25
0.0950	16.0	59.59
0.0997	16.6	60.81
0.1000	16.7	61.02

(d) The Air Flow and Input Energy Relationship

The energy enters the room is directly proportional to the air flow, which is given by eq. (4.7)

$$Q_{in} = \rho c_p \Delta T \dot{\upsilon} \tag{4.7}$$

Where;

$\dot{V}$: is the air flow rate (m^3/s).

Q_{in} : The input energy (KW).

ρ : Is the air density (Kg/m^3).

c_p : Is the specific heat at constant pressure (KJ/Kg.°C).

ΔT : Is the temperature difference between outdoor and the design temperature (°C).

Then, substituting the values of these parameters which are listed previously in Table 3.4, this yield:

$$Q_{in} = 15.647 \dot{\upsilon} \tag{4.8}$$

The detailed room temperature control loop is shown in Fig 4.3.

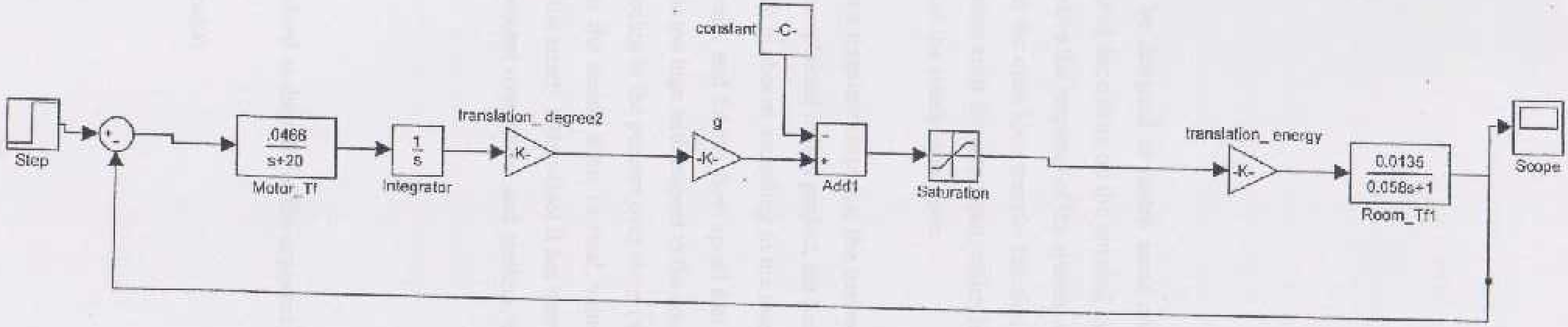


Figure 4.3: Room Temperature Control Loop

4.5 Control System Design

4.5.1 Design Assumption

A controller should be designed to match some needed specifications; these specifications are selected depending on the physical system behavior. The controller is designed to improve the response of the system, and since the system is type one which means that the open loop transfer function has one pole at the origin; it has a zero steady state error for step input, which leads to improve the transient response regardless to the steady state error.

To improve the system transient response; the desired specifications are selected first. For the system explained in this project, the Settling Time is about 40 second ($T_s = 40$), this time is chosen according to the room volume that may be considered a relatively small, and the air flow speed that was measured and given by *6m/s*, this air speed is too high with respect to the comfortable conditions in HVAC systems. And according to the percent over shoot (which is the amount that the waveform overshoots the steady state, or final, value at the peak time, expressed as a percentage of the steady state value) it has been selected to be less than 10%, these values of percent over shoot and steeling time yields $\zeta \geq 0.6$ and $\omega_n = 0.1667$.

Where:

ζ : Damping Ratio, that is defined as the ratio of the exponential decay frequency to the natural frequency

ω_n : The Natural Frequency (rad/s).

T_s : settling Time (s) [7].

4.5.2 Step Response Specifications of Room Temperature

As a first step of controller design, the stability of the system must be checked, the stability can be achieved from the poles of closed loop transfer function, and the system is stable if all the poles of closed loop transfer function are in left half of S_ plane. The following m. file is used to check the stability of the system.

Stability test of the temperature control loop in each room:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
g=1/388.6;
translation_energy=15647;
constant=1.12/388.6;
Room_Tf=tf([.0135],[0.058 1]);
m=Motor_Tf*translation_degree;
ml=(m*g)+constant;
mll=ml*tranlation_energy*Room_Tf;
Tf_Total=feedback(mll,1);
eig(Tf_Total)

 ans =

 -27.8525
 -19.8403
  -0.0453
```

It has been found that all the poles of closed loop transfer function are in negative (left half of S_ plane), then the system is stable. The next step, the uncompensated system specifications are found in order to design the controller that is to meet the needed specifications.

The step response specifications of uncompensated system:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
g=1/388.6;
translation_energy=15647;
constant=1.12/388.6;
Room_Tf=tf([.0135],[0.058 1]);
m=gear_ratio*Motor_Tf*translation_degree;
ml=(m*g)+constant;
mll=ml*translation_energy*Room_Tf;
Tf_Total=feedback(mll,1);
step(Tf_Total*23)
```

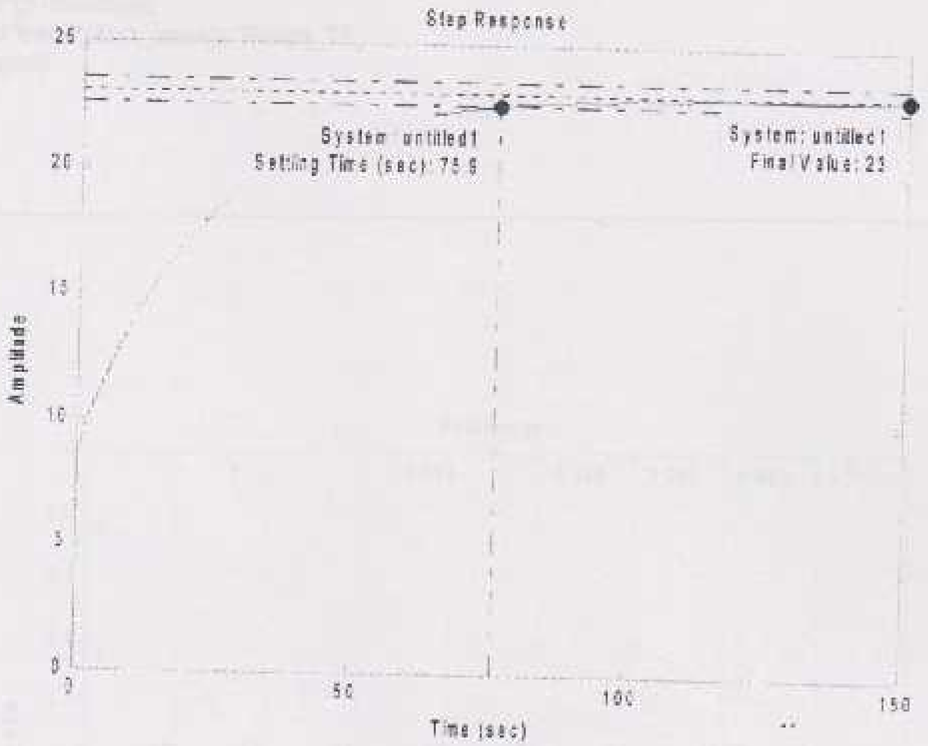


Figure 4.4: Step Response for Uncompensated System

As demonstrated in Fig 4.4 the steady state error for a step input of uncompensated system equals to zero, the settling time is 75.9 s, and over shoot that equals to zero

4.5.3 Control Design Via Root Locus

The root locus displayed both transient response and stability information. The root locus can be achieved by Matlab to get a general idea of the changes in transient response generated by changes in gain. Specific points on the root locus can be found accurately to give quantitative design information [7].

Root Locus of the temperature control loop in each room:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
g=1/388.6;
```

```
translation_energy=15647;
constant=1.12/388.6;
Room_Tf=tf([.0135],[0.058 1]);
m=Motor_Tf*translation_degree;
ml=(m*g)+constant;
mll=ml*translation_energy*Room_Tf;
rlocus(mll)
```

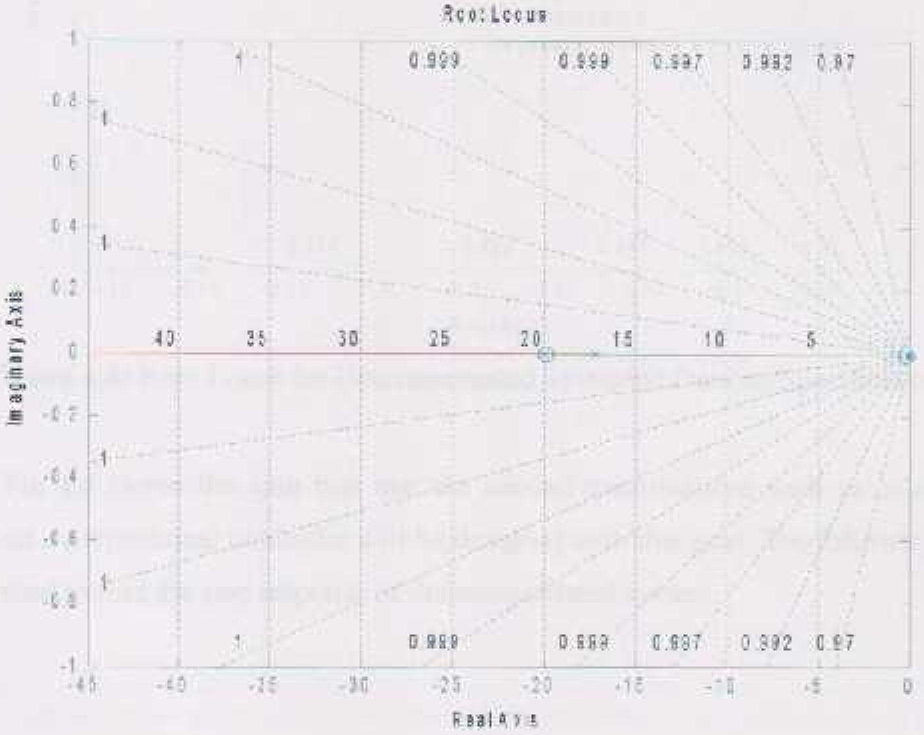


Figure 4.5: Root Locus for Uncompensated System

From the root locus as shown in Fig 4.5 it appears that ζ always equals to one, then the natural frequency ω_n to achieve settling time around 40 second with this value of ζ is calculated to have a value with 0.1rad/s.

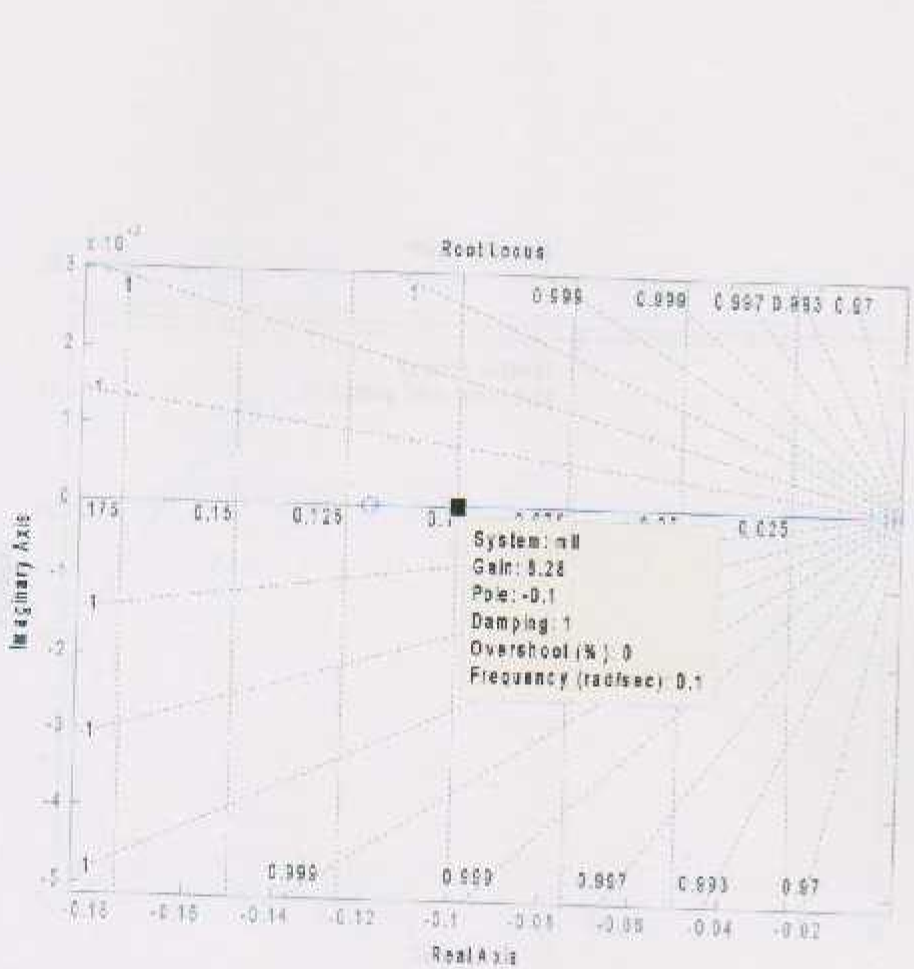


Figure 4.6: Root Locus for Uncompensated System at Desired Specifications

Fig 4.6 shows the gain that met the needed specifications with value equals to 8.28, then a proportional controller will be designed with this gain. The following m. file is presented to find the step response of the compensated system.

The step response specifications of compensated system with gain equals to 8:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
g=1/388.6;
translation_energy=15647;
constant=1.12/388.6;
Room_Tf=tf([.0135],[0.058 1]);
m=8*Motor_Tf*translation_degree;
ml=(m*g)+constant;
mll=ml*translation_energy*Room_Tf;
Tf_Total=feedback(mll,1);
step(Tf_Total*23)
```

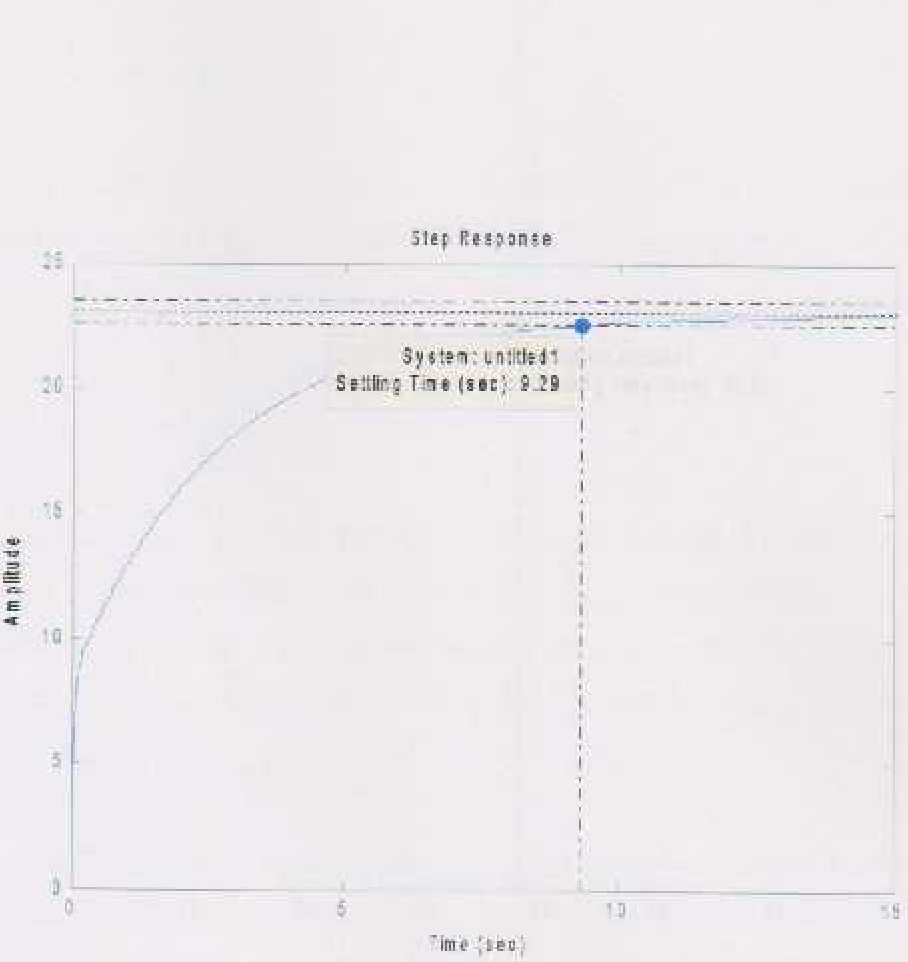


Figure 4.7: Step Response of Compensated System with Gain Equals to 8

The Fig 4.7 shows that the settling time of compensated system is equal 9.29 seconds, and this is very fast according to temperature variation and not acceptable , so that can be reduced the gain of Proportional controller to be 2.

The step response specifications of compensated system with gain equals to 2:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
g=1/388.6;
translation_energy=15647;
constant=1.12/388.6;
Room_Tf=tf([.0135],[0.058 1]);
m=2* Motor_Tf*translation_degree;
ml=(m*g)+constant;
mll=ml*translation_energy*Room_Tf;
Tf_Total=feedback(mll,1);
step(Tf_Total*23)
```

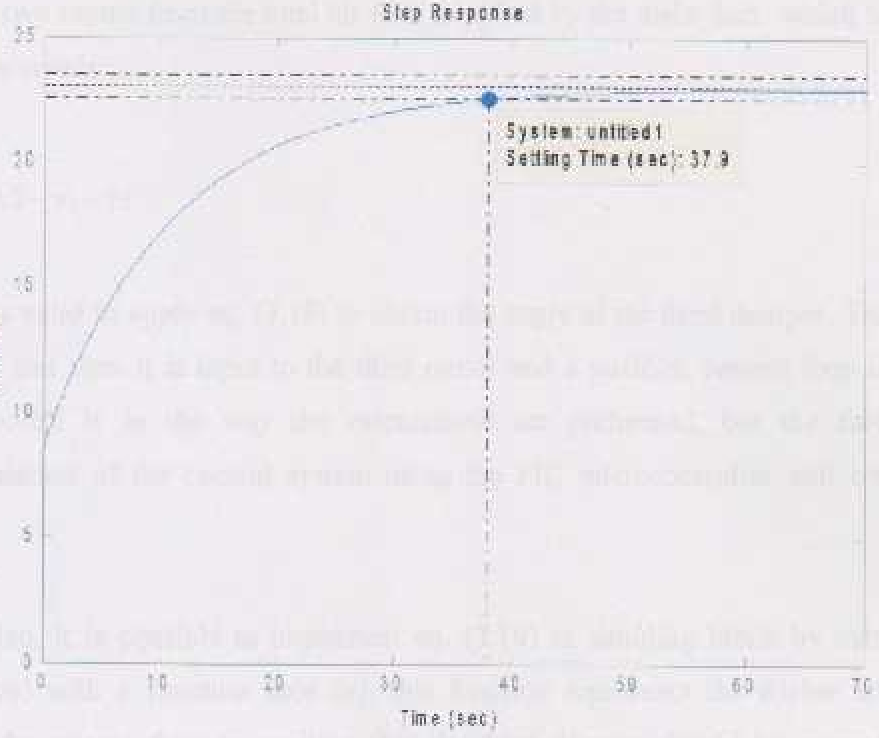


Figure 4.8: Step Response of Compensated System with Gain Equals to 2

Fig 4.8 displays the compensated system with gain = 2, and the settling time is achieved from simulation is around 40 seconds and that is the needed value.

4.6 The Control Loop of the Excess Air

This control loop is required to change the opening of the third damper that is modulated to allow the excess amount of air flow to leave the main duct. The idea is to measure the amount of the excess air flow rate in order to evaluate the suitable opening of the third damper.

The access air flow rate equals to the subtraction of the two air flow rates that enter the two rooms from the total air flow supplied by the main duct which is 0.2 m^3/s, or in other words:

$$\dot{v}_{excess} = 0.2 - \dot{v}_1 - \dot{v}_2 \qquad (4.9)$$

Then, it is valid to apply eq. (3.19) to obtain the angle of the third damper. This angle is measured and then it is input to the third motor and a position control loop is verified. The difficulty is in the way the calculations are performed, but the fact that the implementation of the control system using the PIC microcontroller will simplify the process.

Also, it is possible to implement eq. (3.19) in simuling block by using Matlab block (Fcn) with a function acos (u), this function represents the cosine inverse that translate the excess flow to position that the third damper should be opened, Fig 4.9 shows the simuling block of the overall system.

The controller of the third motor is designed via root locus technique, a proportional controller is required since the system is type one. The required specifications is now specified, the percent over shoot is less than 10% and settling time equals to 2-5 seconds ($\zeta \geq 0.6$ & $3.33 > \omega_n > 1$ rad/sec). The root locus is shown in Fig 4.10. The following m. files express the position control and the step response of the compensated system.

Root Locus of the excess flow damper position control loop:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
m1=Motor_Tf*translation_degree;
rlocus(m1)
```

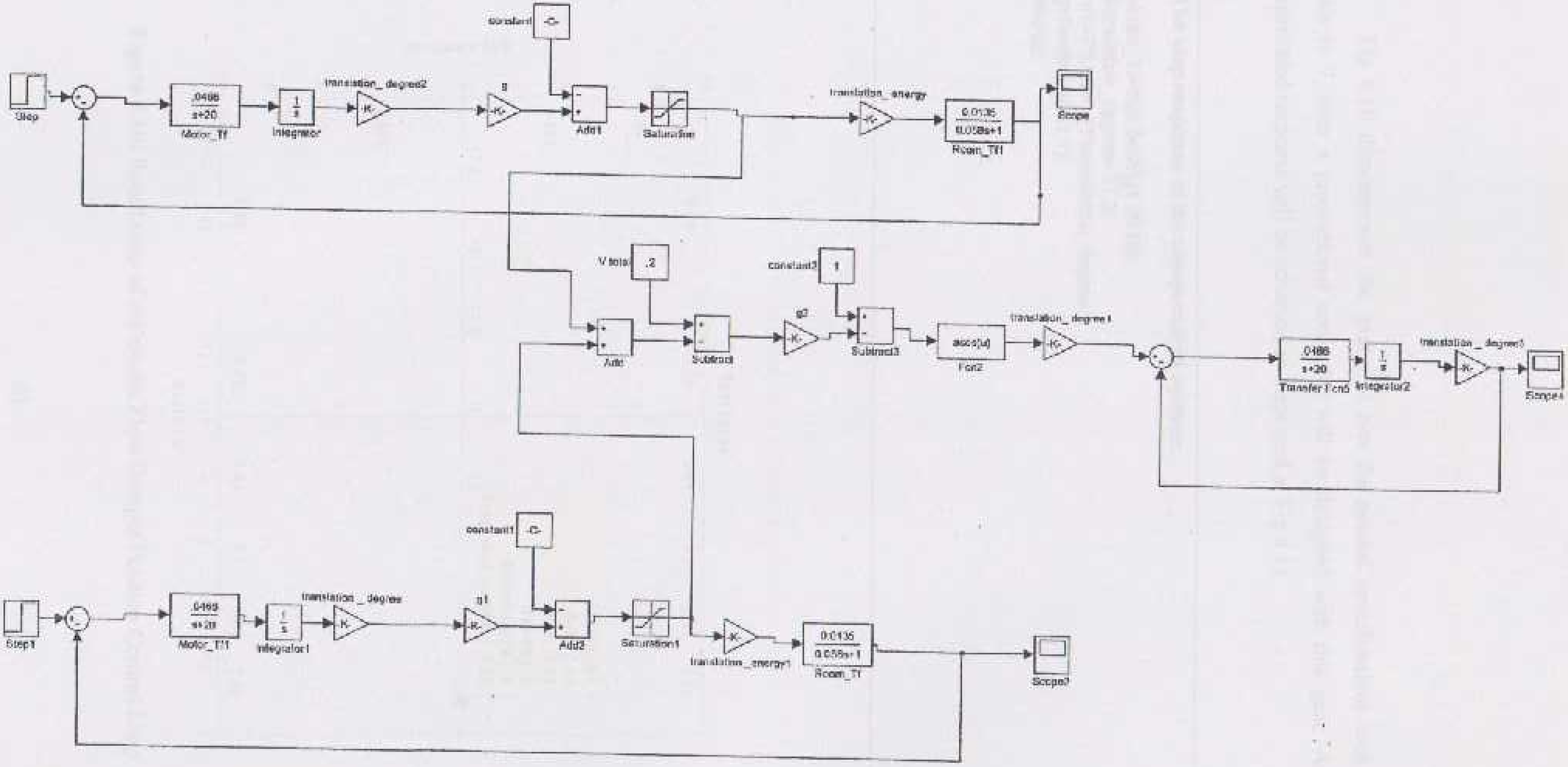


Figure 4.9: The Simulink Block of the Whole System

Fig 4.10 demonstrates the gain that met the needed specifications with value equals to 7, then a proportional controller will be designed with this gain. And the compensated response will be achieved as displayed in Fig 4.11.

The step response of the compensated system:

```
Motor_Tf=tf([0.0466],[1 20 0]);
translation_degree=57.3;
m1=7*Motor_Tf*translation_degree;
g=feedback(m1,1);
step(g)
```

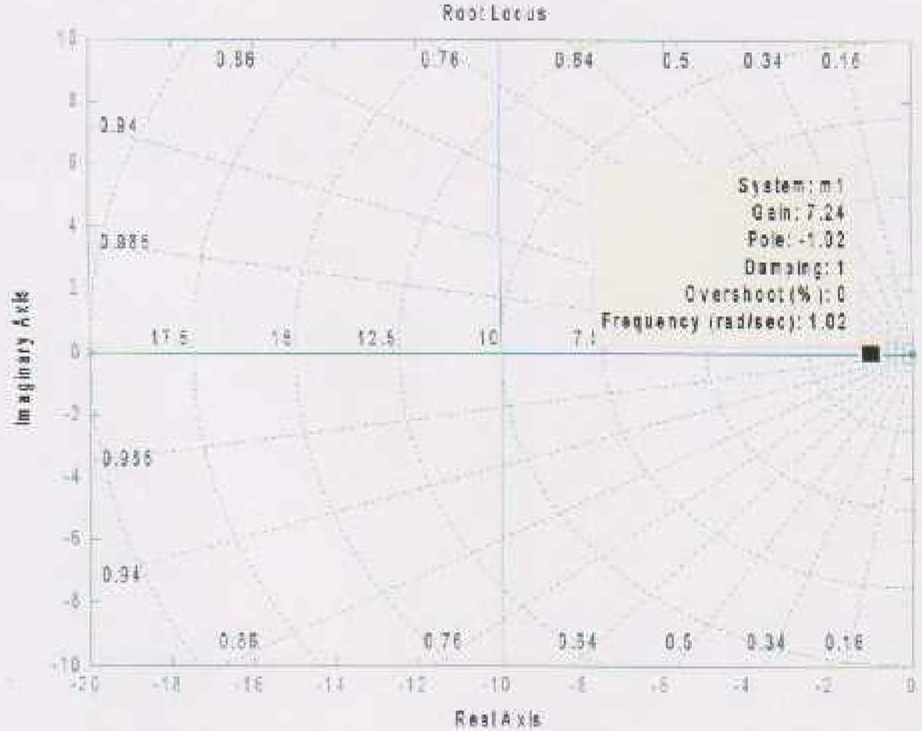


Figure 4.10: Root Locus of the excess Flow Damper Position Control Loop

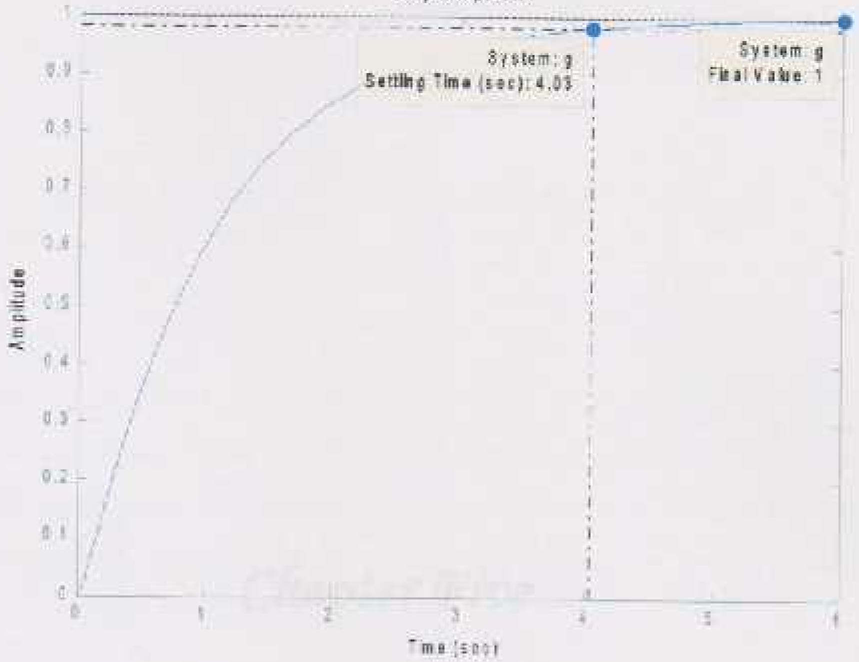


Figure 4.11: The Step Response of Compensated System

Chapter Five

HARDWARE AND IMPLEMENTATION

Chapter Five
HARDWARE AND IMPLEMENTATION

5.1 Overview

After modeling the system and gathering the required data, it is time to translate these efforts to a real system, and validate the assumptions and the mathematical model that resulted in the previous chapters. This chapter contains a general description for the system components (mechanical, electrical, and control system), in addition to the implementation of the PIC microcontroller.

5.2 Prototype Description

This project aims to design and control a VAV Box in an air conditioning system in order to prove that the VAV system is energy conserving. To examine the performance of a VAV system, a prototype has been built in the Air Conditioning and Refrigeration Workshop in PPU.

In the traditional case of designing HVAC systems, the design of the structure must be accomplished first, and according to the thermal load calculations the cooling unit is then selected. The prototype was located in the refrigeration workshop to get benefit from the cooling unit existing there, to decrease the whole project cost.

A. Cooling Unit

The cooling unit is the most important component in this project and the most expensive one, so the prototype was built in the refrigeration workshop in order to get benefit of the cooling unit existing there.

This cooling unit is of high quality, its specifications are shown in Appendix (A).the indoor part of the cooling unit which is located on the false ceiling in the workshop, and this part includes the evaporator coil which is exposed to an air stream that comes from a constant speed fan. In a VAV system, the speed of the fan is variable and changes according to the change in the duct's damper opening, the duct's damper changes its position according to the temperature sensor's reading, this reading indicates if the cooling load increases or decreases.

In this application, since the speed of the fan is constant, the air flow rate from the cooling unit is constant too. This means that another way for controlling the flow rate that enters the zones is needed.

Here, the idea was to add another VAV box at the terminal of an additional branch to exhaust the excess air flow rate from the system and this will be clarified later.

B. Building description

Building the prototype was for testing the designed VAV system and to make it possible to examine the system validity. This building was established after gathering certain information about heat transfer in order to build a convenient structure for experimenting the designed VAV.

Insulation is one of the most important issues that should be taken into consideration, so heat gain in the system could be applied according to the user need.

The structure was built of two layers of gypsum boards and insulated material called polystyrene in between, these building material were used to build two rooms each with dimensions of (1.2mx1.5mx2.15m).

A door was placed at the front wall of each room with (1.70mx0.7m) dimensions. Fig. 5.1 shows the rooms' final shape.

Figure 5.1: The Prototype

C. Duct System

Previously, the cooling unit was classified as the most important component in the Air Conditioning System, if it could be considered like the heart of the system, so the duct system could be considered as blood vessels. Duct system supplies the rooms with cooled air. The indoor unit which consists of an evaporator and a fan has six main outlets, two of them are closed, as shown in Fig 5.2, the others are used, and one of them was selected to be the main supplying duct for the system.

Figure 5.2: The Inner Cooling Unit

As is shown in Fig. 5.3 the main supply duct is connected to Y-fitting. Now, two terminals are resulted; the first one is taken to another Y-fitting to connect its terminals with two flexible ducts, and VAV boxes are fixed at the outlet of them in each room. The second was connected with third VAV box for getting rid of the excess air flow rate.

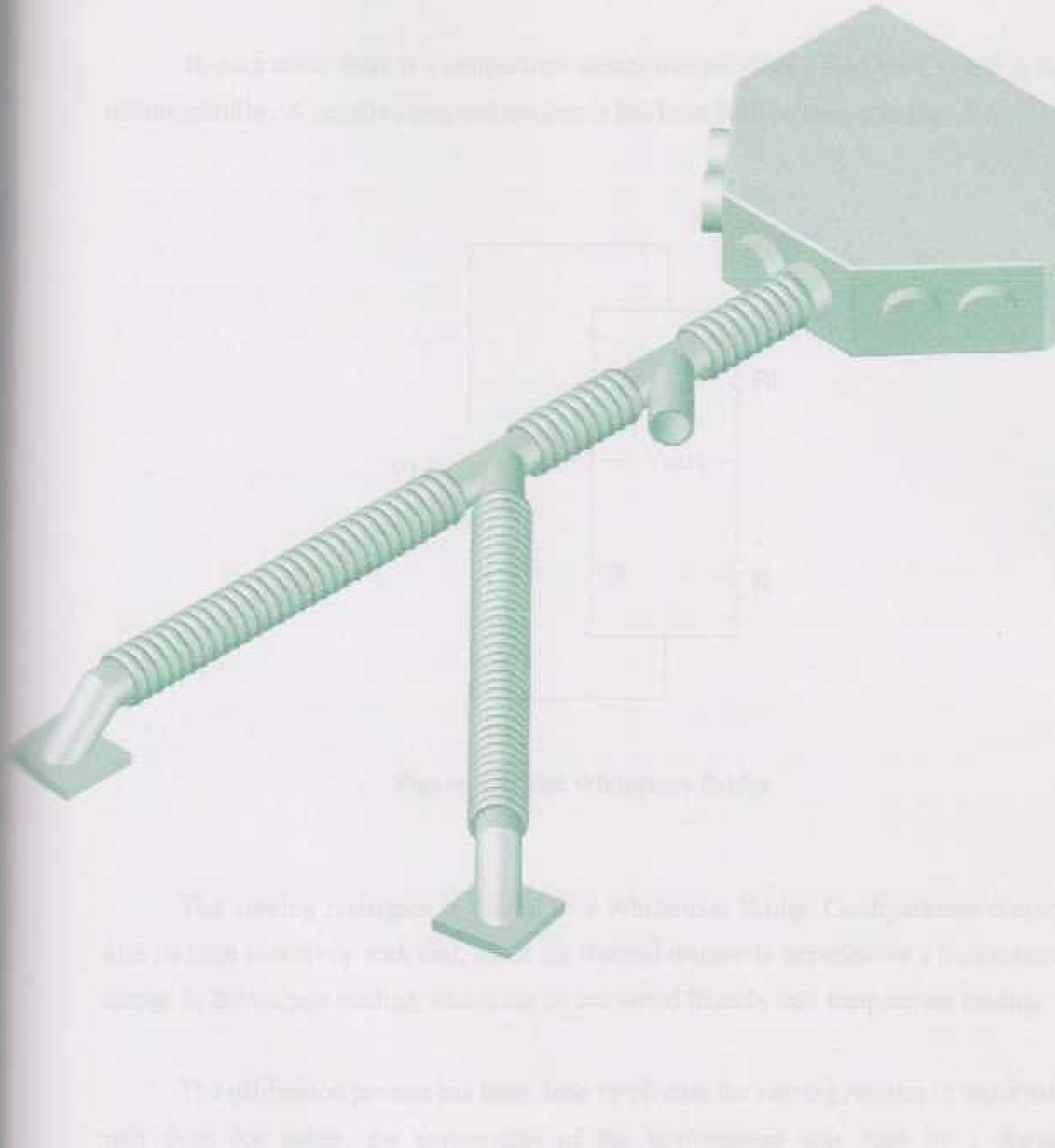


Figure 5.3: The Main Supply Duct with the Y Connections

D. Temperature Sensors

In each room there is a temperature sensor that provides a feed back signal to the microcontroller. A resistive temperature sensor has been built as shown in Fig 5.4.

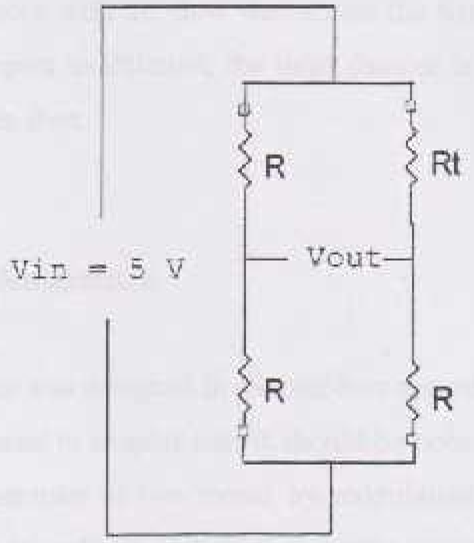


Figure 5.4: The Whitestone Bridge

The sensing resistance is placed in a Whitestone Bridge Configuration distinct with its high sensitivity such that, when the thermal resistor is experiences a temperature change in the voltage reading, which can be converted linearly into temperature reading.

The calibration process has been done by placing the sensing resistor in water just melt from ice cubes, the temperature of the environment was read by a digital thermometer, at 0.6 °C , and the resistance was obtained by the ohmmeter as 24 $K\Omega$, then the other resistors were selected of the same value, then the Bridge was constructed as shown in Fig 5.4.

E. Overall System Performance

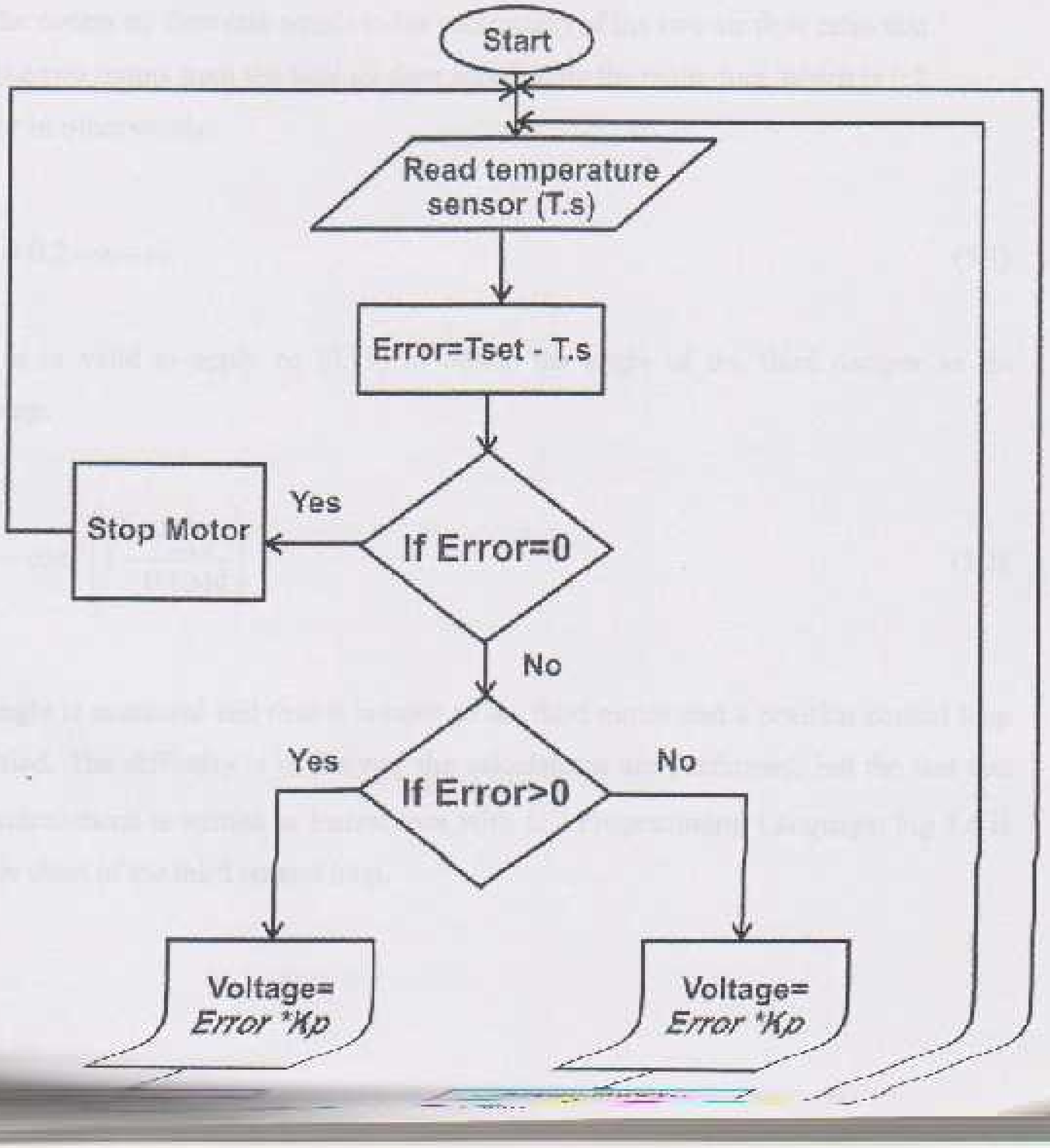
Inside each room the temperature sensor sends a signal which represents the temperature of the rooms' air to the controller, then the controller compares between the reading of the temperature sensor with the temperature setpoint, if the difference is resulted, then the controller sends a signal to the actuator of the damper to modulate its opening to supply the room with air flow that meets the temperature difference. As a result of the rooms' dampers modulation, the third damper is then varied to release the excess air outside the main duct.

5.3 Control System Implementation

The control system was designed in chapter four according to the required specifications. As mentioned in chapter four it should be noticed that this project aims to control the temperature of two rooms by modulating the amount of air flow entering the rooms in order to reduce the energy consumed more than to control Settling Time; which is the time required for the response to reach and steady with ±2 % of the steady-state value, and other step response specifications.

As a result of modulating either one or both the two dampers, the air flow in the main duct will be varied during the operation; this is an excess air which must be removed. Therefore, a third damper is required to allow that. Thus, the overall system is composed of three control loops, two for the temperature control of the two rooms and the third is to control the opening of exhausted air.

Now these control loops are implemented using the PIC Microcontroller 18F452, the data sheet of this PIC is shown in Appendix (C). And the programming language is C^{++}. Each control loop will follow a flow chart that represents the sequence of the system. The two rooms are identical and the control loop is also the same, Fig 5.5 shows the flow chart for each room temperature control.



The third control loop represents a position control of the motor, and the step input position is calculated through a set of equations as described in section 4.6. The access air flow rate equals to the subtraction of the two air flow rates that enter the two rooms from the total air flow supplied by the main duct which is 0.2 m^3/s, or in other words:

$$\dot{v}_{excess} = 0.2 - \dot{v}_1 - \dot{v}_2 \tag{5.1}$$

Then, it is valid to apply eq (3.19) to obtain the angle of the third damper as the following:

$$\theta_{excess} = \cos^{-1}\left(1 - \frac{v_{excess}}{0.1944}\right) \tag{5.2}$$

This angle is measured and then it is input to the third motor and a position control loop is verified. The difficulty is in the way the calculations are performed, but the fact that these calculations is written as instructions with C^{++} Programming Language. Fig 5.6 is the flow chart of the third control loop.

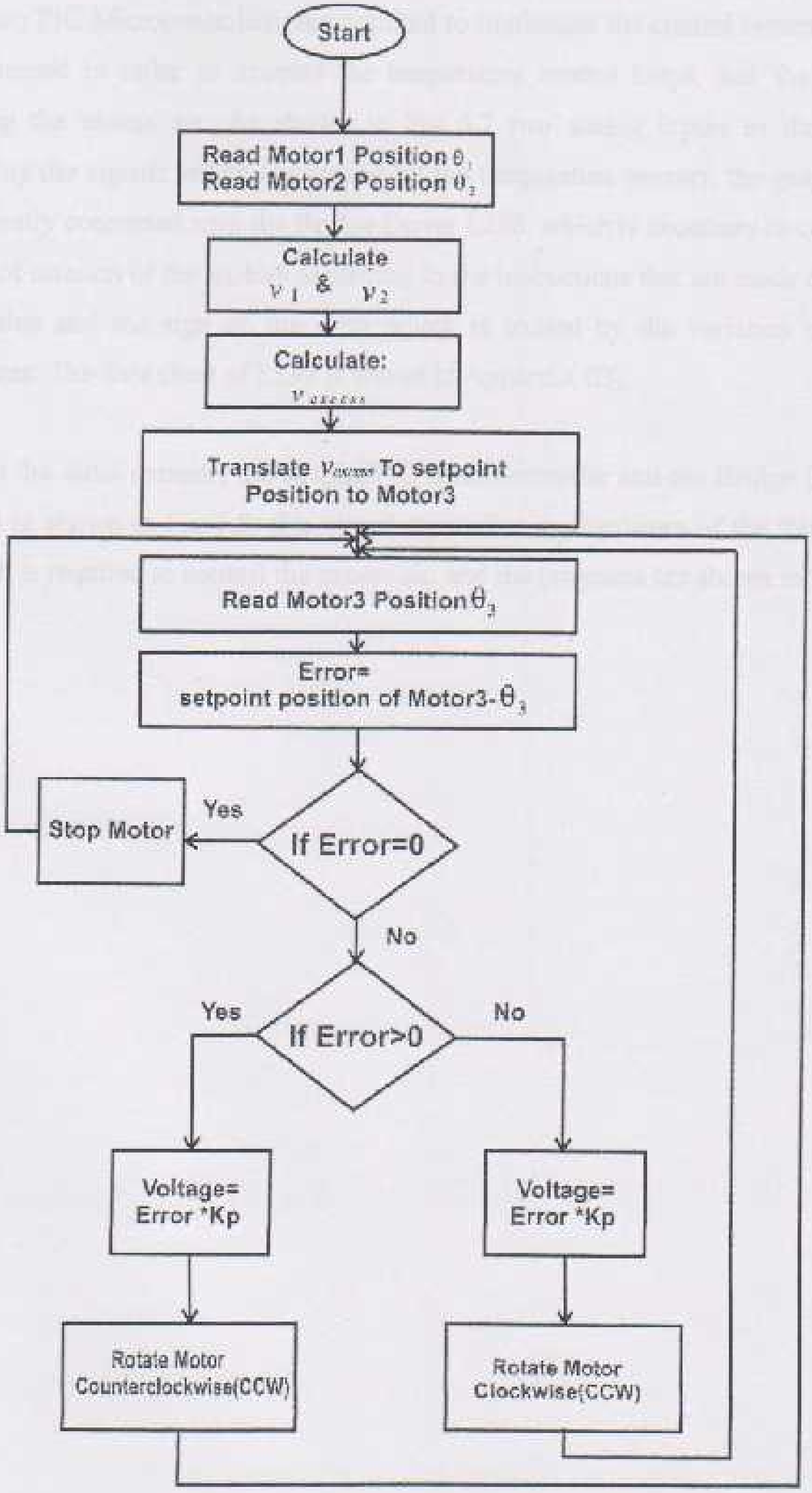


Figure 5.6: The Excess Air Flow Chart

Two PIC Microcontrollers are required to implement the control system; the first is programmed in order to execute the temperature control loops, and the other for controlling the excess air. As shown in Fig 5.7 two analog inputs to the PIC are representing the signals which are sensed by the temperature sensors, the output of the PIC is directly connected with the Bridge Driver L298, which is necessary to counter the direction of rotation of the motors according to the instructions that are made depending on the value and the sign of the error which is caused by the variation of rooms' temperatures. The data sheet of L298 is shown in Appendix (D).

On the same manner, the second PIC Microcontroller and the Bridge Driver are connected as shown in Fig 5.8; this circuit represents the hardware of the third control loop which is required to control the excess air, and the programs are shown in Appendix (E).

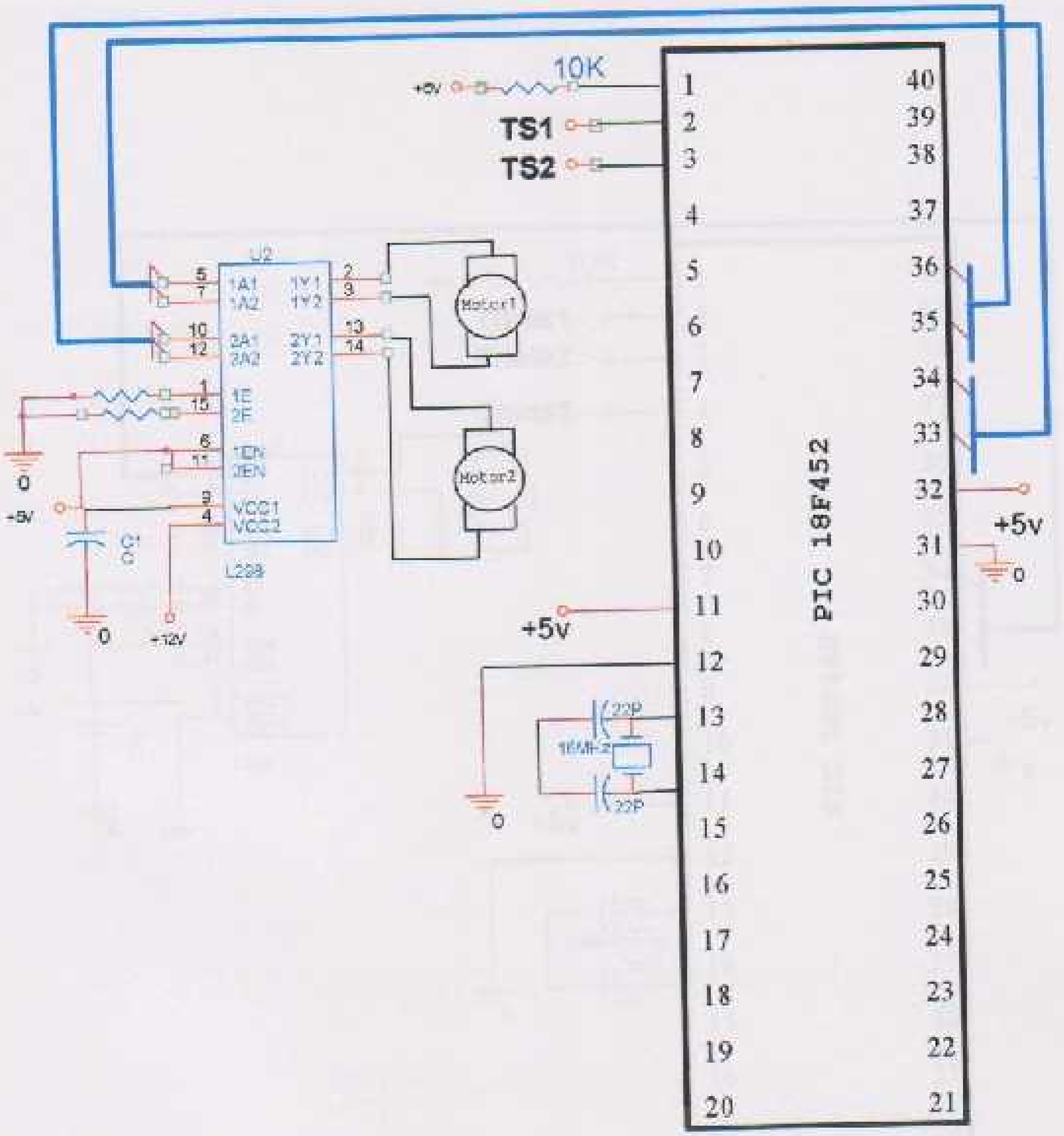


Figure 5.7: The Connection Circuit of the First PIC Microcontroller

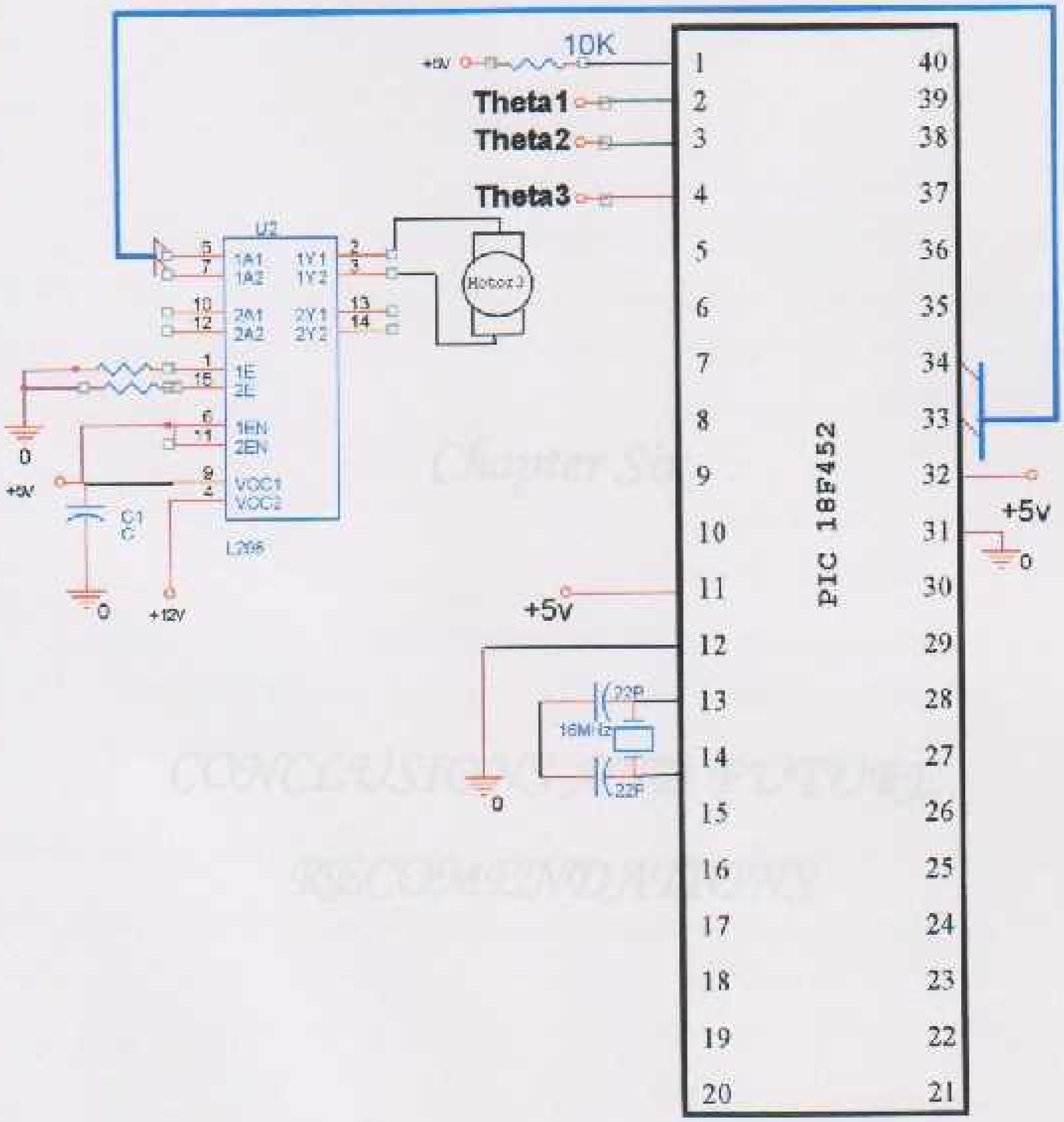


Figure 5.8: The Connection Circuit of the Second PIC Microcontroller

Chapter Six

CONCLUSIONS AND FUTURE RECOMENDATIONS

Chapter Six
CONCLUSIONS AND RECOMMENDATIONS

6.1 Overview

VAV system is developed to be the optimal solution for the human luxury and energy conservation. This chapter includes the actual results which validate the system design. And it also includes the recommendations for the future work.

6.2 System Testing

After establishing the overall system, experiments were carried out in order to examine the behavior of the system and to study the results that show whether the model is applicable or not.

In chapter three, eq.(3.19) was derived to reflect the relationship between the air flow rate ($\dot{v}$) and the position of the damper (θ) which was rearranged in eq.(4.4) as follows:

$$\dot{v} = 0.1944(1 - \cos\theta) \qquad (4.4)$$

When the dampers and flexible ducts were connected, the dampers of the two rooms were adjusted to a certain position of ($\theta = 45$ °) and the third terminal was adjusted for ($\theta = 56$ °). The air flow rates were measured using the flow meter and the results were (0.06 m^3/s) from the rooms' dampers and (0.08 m^3/s) out from the third terminal. By applying eq.(4.4) with ($\theta = 45$ °) gives (0.0569 m^3/s) and (0.0857 m^3/s) with ($\theta = 56$ °).

After constructing the whole system and connecting the control until, the temperature increased in order to be sensed, and then the dampers were opened to its maximum opening, which validates the system. On the other hand when the temperature decreased below to the setpoint, it was assumed that the damper must change its direction, and stop at a lesser opening than the previous one. And the fact that after decreasing the temperature, the damper changed its opening with a settling time more that the calculated That's because the sensor was responding slowly to temperature changes.

6.3 Energy Calculations

As mentioned previously, the energy conserved is represented by the excess flow out from the third damper, to calculate this energy the excess flow is measured and applying eq.(6.1) as the following:

$$q = v c_p (T_e - T_i)/\upsilon \qquad (6.1)$$

Where;

q : the energy conserved (W).

T_e : the exit air temperature (°C).

T_i : the inlet air temperature (°C).

υ : the specific volume of air which equals to 0.84 (m^3/Kg).

Applying eq.(6.1) with (0.08 m^3/s) air flow rate the resulted energy during an hour will be 3.790 (KW).

6.4 Conclusions

The achieved results lead to the following remarks:

1. The direct digital control is applicable in HVAC systems where the desired specifications could be achieved.
2. VAV system is energy conserving system.

6.5 Recommendations

Many enhancements can be added to this project as the following:

1. Expanding the control design to involve humidity, ventilation, CO_2, and other variables.
2. Replacing the constant speed fan by a variable speed fan and control it.
3. Connecting the control unit with the control panel of the chiller.
4. To insert the temperature set-point as a variable input to the microcontroller to make it more flexible.
5. To enhance the temperature sensor's' performance in order to enhance the settling time.

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APPENDICES

Appendix A

COOLING UNIT SPECIFICATIONS

PERFORMANCE TABLE

MODEL	ENTER WBT °F	ENTER CFM	TOTAL CAP. MBH	COMP. KW	LEAV WBT. °F	CAPACITY AT 95 (°F) AMBIENT TEMPERATURE / INDOOR ON COIL TEMP. (°F) D.B. 72 SEN. MBH	72 L. DBT °F	72 SHR	76 SEN. MBH	76 L. DBT °F	76 SHR	78 SEN. MBH	78 L. DBT °F	78 SHR	80 SEN. MBH	80 L. DBT °F	80 SHR
DGP 64 + CM 34	63	2000	64.354	5.61	-	-	-	-	-	-	-	-	-	-	-	-	-
		2400	66.744	5.73	52.80	43.330	55.3	0.649	48.875	56.0	0.732	52.571	56.5	0.788	56.267	57.0	0.843
		2800	68.454	5.84	54.00	48.142	57.3	0.703	55.092	58.0	0.805	59.726	58.5	0.874	64.368	59.0	0.940
	65	2000	66.923	5.74	51.90	35.860	55.7	0.536	40.083	54.4	0.599	42.891	54.9	0.641	45.200	55.4	0.682
		2400	69.330	5.89	54.90	39.514	56.8	0.570	45.905	57.3	0.690	48.754	58.0	0.703	52.450	58.5	0.757
		2800	71.231	5.99	56.90	43.478	58.8	0.610	50.328	59.5	0.707	55.011	60.0	0.772	59.644	60.5	0.837
	67	2000	70.372	5.94	55.50	33.280	55.1	0.473	37.495	55.8	0.533	40.381	56.3	0.575	43.109	56.8	0.613
		2400	73.044	6.07	56.80	36.075	58.2	0.494	41.622	59.0	0.570	43.310	59.5	0.620	49.015	59.9	0.671
		2800	75.103	6.17	58.80	39.080	60.2	0.520	46.031	61.0	0.617	50.664	61.4	0.679	55.298	61.9	0.736
	69	2000	72.156	6.03	54.50	31.979	55.8	0.443	36.191	56.5	0.502	38.993	57.0	0.540	41.808	57.5	0.579
		2400	75.003	6.14	57.70	34.355	58.6	0.458	39.088	59.7	0.532	43.556	60.1	0.581	47.743	60.6	0.631
		2800	77.139	6.26	59.70	36.979	61.0	0.478	43.829	61.7	0.568	48.453	62.2	0.628	53.098	62.7	0.688
	71	2000	75.926	6.30	56.40	29.384	57.2	0.387	33.997	57.9	0.442	36.550	58.4	0.479	39.334	58.9	0.516
		2400	78.891	6.34	59.60	30.240	60.1	0.391	36.387	61.1	0.461	40.083	61.6	0.508	43.780	62.1	0.555
		2800	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

PERFORMANCE TABLE

CAPACITY AT 95 (°F) AMBIENT TEMPERATURE

MODEL	ENTER. WBT °F	ENTER. CFM	TOTAL CAP. MBH	COMP. kW	LEAV. WBT. °F	INDOOR ON COIL TEMP. (°F) D.B. 73 SEN. MBH	73 L. DBT °F	73 SHR	76 SEN. MBH	76 L. DBT °F	76 SHR	78 SEN. MBH	78 L. DBT °F	78 SHR	80 SEN. MBH	80 L. DBT °F	80 SHR
DSP 64 + CMD 20	63	1260	55.324	5.26	-	-	-	-	-	-	-	-	-	-	-	-	-
		1700	59.333	5.41	-	-	-	-	-	-	-	-	-	-	-	-	-
		2080	61.553	5.31	-	-	-	-	-	-	-	-	-	-	-	-	-
	65	1260	57.345	5.34	-	-	-	-	-	-	-	-	-	-	-	-	-
		1700	61.404	5.50	-	-	-	-	-	-	-	-	-	-	-	-	-
		2080	63.751	5.59	-	-	-	-	-	-	-	-	-	-	-	-	-
	67	1260	60.007	5.44	-	-	-	-	-	-	-	-	-	-	-	-	-
		1700	64.229	5.61	-	-	-	-	-	-	-	-	-	-	-	-	-
		2080	66.969	5.75	56.90	32.819	58.6	0.490	37.797	59.4	0.564	41.116	59.9	0.614	44.435	60.3	0.664
	68	1260	61.263	5.49	-	-	-	-	-	-	-	-	-	-	-	-	-
		1700	65.852	5.68	54.50	28.951	55.9	0.440	32.653	56.7	0.496	35.122	57.2	0.533	37.590	57.8	0.571
		2080	68.580	5.85	57.90	31.210	59.3	0.455	36.187	60.1	0.528	39.505	60.6	0.576	42.824	61.2	0.624
	70	1260	63.947	5.60	-	-	-	-	-	-	-	-	-	-	-	-	-
		1700	65.955	5.82	56.50	36.549	57.3	0.385	30.252	58.1	0.305	32.720	58.7	0.475	35.188	59.2	0.540
		2080	71.990	6.02	59.50	28.006	60.7	0.389	32.984	61.5	0.458	36.303	62.0	0.504	39.677	62.6	0.550
DSP 64 + CM 20	63	1600	59.681	5.43	-	-	-	-	-	-	-	-	-	-	-	-	-
		2000	62.351	5.54	-	-	-	-	-	-	-	-	-	-	-	-	-
		2400	64.361	5.62	-	-	-	-	-	-	-	-	-	-	-	-	-
	65	1600	61.741	5.51	-	-	-	-	-	-	-	-	-	-	-	-	-
		2000	64.605	5.62	-	-	-	-	-	-	-	-	-	-	-	-	-
		2400	66.929	5.75	56.00	38.921	58.2	0.582	44.666	59.0	0.667	48.496	59.5	0.725	52.327	60.1	0.782
	67	1600	64.602	5.67	-	-	-	-	-	-	-	-	-	-	-	-	-
		2000	67.900	5.81	55.30	32.383	56.9	0.477	36.782	57.7	0.542	39.714	58.2	0.585	42.647	58.8	0.628
		2400	70.381	5.94	57.90	35.276	59.6	0.501	41.023	60.4	0.583	44.854	60.9	0.637	48.685	61.4	0.692
	68	1600	66.302	5.69	51.50	28.867	52.8	0.435	32.004	53.7	0.483	34.096	54.2	0.514	36.187	54.7	0.546
		2000	69.623	5.90	56.20	31.008	57.6	0.445	35.406	58.4	0.509	38.339	58.9	0.551	41.271	59.5	0.593
		2400	72.166	6.03	58.90	33.445	60.3	0.463	39.192	61.1	0.543	43.024	61.6	0.596	46.855	62.1	0.649
	70	1600	68.408	5.90	53.40	26.853	55.3	0.387	29.990	55.1	0.432	32.081	55.6	0.462	34.173	56.1	0.497
		2000	73.135	6.07	58.10	28.222	59.0	0.386	32.621	59.8	0.446	35.555	60.3	0.486	38.488	60.8	0.526
		2400	75.935	6.20	60.80	29.790	61.6	0.392	35.537	62.5	0.468	39.368	63.0	0.518	43.200	63.5	0.569

Appendix B
CONSTANTS FOR COOLING LOAD CALCULATIONS

TABLE 3.3 Thermal Properties of Selected Materials

	Density, lb/ft^3	Thermal conductivity, (Btu/h·ft·°F)	Specific heat, (Btu/lb·°F)	Emissivity	
Aluminum (alloy 1100)	171	128	0.214	0.09	
Asbestos: insulation	120	0.092	0.20	0.93	
Asphalt	132	0.43	0.22		
Brick, building	123	0.4	0.2	0.93	
Brass (65% Cu, 35% Zn)	519	69	0.09	0.033	Highly polished
Concrete (stone)	144	0.54	0.156		
Copper (electrolytic)	556	227	0.092	0.072	Shiny
Glass: crown (soda-lime)	154	0.59	0.18	0.94	Smooth
Glass wool	3.25	0.022	0.157		
Gypsum	78	0.25	0.259	0.903	Smooth plate
Ice (32°F)	57.5	1.3	0.487	0.95	
Iron: cast	450	27.6	0.12	0.435	Freshly turned
Mineral fiberboard:					
acoustic tile, wet-molded	23	0.035	0.14		
wet-felted	21	0.031	0.19		
Paper	58	0.075	0.32	0.92	
Polystyrene, expanded, molded beads	1.25	0.021	0.29		
Polyurethane, cellular	1.5	0.013	0.38		
Plaster, cement and sand	132	0.43		0.91	Rough
Platinum	1340	39.9	0.032	0.054	Polished
Rubber: vulcanized, soft	68.6	0.08	0.48	0.86	Rough
hard	74.3	0.092		0.95	Glossy
Sand	94.6	0.19	0.191		
Steel (mild)	489	26.2	0.12		
Tin	455	37.5	0.056	0.06	Bright
Wood: fir, white	27	0.068	0.33		
oak, white	47	0.102	0.57	0.90	Planed
plywood, Douglas fir	34	0.07	0.29		
Wool: fabric	20.6	0.037			

Source: Adapted with permission from *ASHRAE Handbook 1989, Fundamentals.*

TABLE 6.2 CLTD for Calculating Sensible Cooling Loads from Sunlit Walls of North Latitude, °F

Facing	Solar time, h 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	Hours of maximum CLTD	Minimum CLTD	Maximum CLTD	Difference in CLTD
Group C walls: typical, outside 1-in. stucco, 2-in. insulation (5.7 lb/ft²), 4-in. concrete, 0.75-in. plaster or gypsum, inside U = 0.119 Btu/h·ft²·°F; mass, 63 lb/ft²																												
N	15	14	13	12	11	10	9	8	8	7	7	8	8	9	10	12	13	14	15	16	17	17	17	16	22	7	17	10
NE	19	17	16	14	13	11	10	10	11	13	15	17	19	20	21	22	22	23	23	23	23	22	21	20	20	10	23	13
E	22	21	19	17	15	14	12	12	14	16	19	22	25	27	29	29	30	30	30	29	28	27	26	24	18	12	30	18
SE	22	21	19	17	15	14	12	12	12	13	16	19	22	24	26	28	28	29	29	29	28	27	26	24	19	12	29	17
S	21	19	18	16	15	13	12	10	9	9	9	10	11	14	17	20	22	24	25	26	25	25	24	22	20	9	26	17
SW	29	27	25	22	20	18	16	15	13	12	11	11	11	13	15	18	22	26	29	32	33	33	32	31	22	11	33	22
W	31	29	27	25	22	20	18	16	14	13	12	12	12	13	14	16	20	24	29	32	35	35	35	33	22	12	35	23
NW	25	23	21	20	18	16	14	13	11	10	10	10	10	11	12	13	15	18	22	25	27	27	27	26	22	10	27	17
Group D walls: typical, outside 1-in. stucco, 4-in. concrete, 1- or 2-in. insulation (2 lb/ft²), 0.75-in. plaster or gypsum, inside U = 0.119–0.20 Btu/h·ft²·°F; mass, 63 lb/ft²																												
N	15	13	12	10	9	7	6	6	6	6	6	7	8	10	12	13	15	17	18	19	19	19	18	16	21	6	19	13
NE	17	15	13	11	10	8	7	8	10	14	17	20	22	23	23	24	24	25	25	24	23	22	20	18	19	7	25	18
E	19	17	15	13	11	9	8	9	12	17	22	27	30	32	33	33	32	32	31	30	28	26	24	22	16	8	33	25
SE	20	17	15	13	11	10	8	8	10	13	17	22	26	29	31	32	32	32	31	30	28	26	24	22	17	8	32	24
S	19	17	15	13	11	9	8	7	6	6	7	9	12	16	20	24	27	29	29	29	27	26	24	22	19	6	29	23
SW	28	25	22	19	16	14	12	10	9	8	8	8	10	12	16	21	27	32	36	38	38	37	34	31	21	8	38	30
W	31	27	24	21	18	15	13	11	10	9	9	9	10	11	14	18	24	30	36	40	41	40	38	34	21	9	41	32
NW	25	22	19	17	14	12	10	9	8	7	7	8	9	10	12	14	18	22	27	31	32	32	30	27	22	7	32	25
Group G walls: typical, outside 1-in. stucco, airspace: 1-, 2-, or 3-in. insulation (2 lb/ft²), 0.75-in. plaster or gypsum, inside U = 0.081–0.78 Btu/h·ft²·°F; mass, 16 lb/ft²																												
N	3	2	1	0	−1	2	7	8	9	12	15	18	21	23	24	24	25	26	22	15	11	9	7	5	18	−1	26	27
NE	3	2	1	0	−1	9	27	36	39	35	30	26	26	27	27	26	25	22	18	14	11	9	7	5	9	−1	39	40
E	4	2	1	0	−1	11	31	47	54	55	50	40	33	31	30	29	27	24	19	15	12	10	8	6	10	−1	55	56
SE	4	2	1	0	−1	5	18	32	42	49	51	48	42	36	32	30	27	24	19	15	12	10	8	6	11	−1	51	52
S	4	2	1	0	−1	0	1	5	12	22	31	39	45	46	45	39	31	25	20	15	12	10	8	5	14	−1	46	47
SW	5	4	3	1	0	0	2	5	8	12	16	26	38	50	59	63	61	52	37	24	17	13	10	8	16	0	63	63
W	6	5	3	2	1	1	2	5	8	11	15	19	27	41	56	67	72	67	48	29	20	15	11	8	17	1	72	71
NW	5	3	2	1	0	0	2	5	8	11	15	18	21	27	37	47	55	55	41	25	17	13	10	7	18	0	55	55

Direct applications and adjustments are stated in the text.

Source: Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

TABLE 6.4 CLTD for Calculating Sensible Cooling Loads from Flat Roofs, °F

Description of construction	Weight, lb/ft²	U-value, Btu/h·ft²·°F	Solar time, h: 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	Hours of maximum CLTD	Minimum CLTD	Maximum CLTD	Difference in CLTD
Without suspended ceiling																														
2.5-in. wood with *2-in. insulation*	13	0.093	30	26	23	19	16	13	10	9	8	5	13	17	23	29	36	41	46	49	51	50	47	43	39	35	19	8	51	43
4-in. wood with 2-in. insulation	18	0.078	38	36	33	30	28	25	22	20	18	17	16	17	18	21	24	28	32	36	39	41	43	43	42	40	22	16	43	27
With suspended ceiling																														
1-in. wood with 2-in. insulation	10	0.083	25	20	16	13	10	7	5	5	7	12	18	25	33	41	48	53	57	57	56	52	46	40	34	29	18	5	57	52
2.5-in. wood with 1-in. insulation	15	0.096	34	31	29	26	23	21	18	16	15	15	16	18	21	25	30	34	38	41	43	44	44	42	40	37	21	15	44	29
8-in. lightweight concrete	33	0.093	39	36	33	29	26	23	20	18	15	14	14	15	17	20	25	29	34	38	42	45	46	45	44	42	21	14	46	32
4-in. heavyweight concrete with 2-in. insulation	54	0.090	30	29	27	26	24	22	21	20	20	21	22	24	27	29	32	34	36	38	38	38	37	36	34	33	19	20	38	18
2.5-in. wood with 2-in. insulation	15	0.072	35	33	30	28	26	24	22	20	18	16	18	20	22	25	28	32	35	38	40	41	41	40	39	37	21	18	41	23
Roof terrace system	77	0.082	30	29	28	27	26	25	24	23	22	22	22	23	23	25	26	28	29	31	32	33	33	33	33	32	22	33	22	11
6-in. heavyweight concrete with 2-in. insulation	77	0.088	29	28	27	26	25	24	23	22	21	21	22	23	25	26	28	30	32	33	34	34	34	33	32	31	20	21	34	13
4-in. wood with 2-in. insulation	19 20	0.082 0.064	35	34	33	32	31	29	27	26	24	23	22	21	22	22	24	25	27	30	32	34	35	36	37	36	23	21	37	16

Conditions of direct application and adjustments are stated in the text.
Source: *Abridged with permission from ASHRAE Handbook 1989, Fundamentals.*

TABLE 6.6 July Solar Cooling Load for Sunlit Glass, 40° North Latitude

Glass face	Solar time, h																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
												Zone type A												
N	0	0	0	0	1	25	27	28	32	35	38	40	40	39	36	31	31	36	12	6	3	1	1	0
NE	0	0	0	0	2	85	129	134	112	75	55	48	44	40	37	32	26	18	7	3	2	1	0	0
E	0	0	0	0	2	93	157	185	183	154	106	67	53	45	39	33	26	18	7	3	2	1	0	0
SE	0	0	0	0	1	47	95	131	150	150	131	97	63	49	41	34	27	18	7	3	2	1	0	0
S	0	0	0	0	0	9	17	25	41	64	85	97	96	84	63	42	31	20	8	4	2	1	0	0
SW	0	0	0	0	0	9	17	24	30	35	39	64	101	133	151	152	130	95	38	17	8	4	2	1
W	1	0	0	0	0	9	17	24	30	35	38	40	65	114	158	187	192	156	57	27	13	6	3	2
NW	1	0	0	0	0	9	17	24	30	35	38	40	40	50	84	121	143	130	46	22	11	5	3	1
Horiz.	0	0	0	0	0	24	69	120	169	211	241	257	259	245	217	176	125	70	29	14	7	3	2	1
												Zone type B												
N	2	2	1	1	1	22	23	24	28	32	35	37	38	37	35	32	31	35	16	10	7	5	4	3
NE	2	1	1	1	2	73	109	116	101	73	58	52	48	45	41	36	30	23	13	9	6	5	3	3
E	2	2	1	1	2	80	133	159	162	148	105	74	63	55	48	41	34	25	15	10	7	5	4	3
SE	2	2	1	1	1	40	81	112	131	134	122	96	69	58	49	42	35	26	15	10	8	6	4	3
S	2	2	1	1	1	8	15	21	36	56	74	86	87	79	62	46	37	27	16	11	8	5	4	3
SW	6	5	4	3	2	6	16	22	27	31	36	58	89	117	135	138	126	94	46	31	21	15	11	8
W	8	6	5	4	3	9	16	22	27	31	35	37	59	101	139	166	175	147	66	40	30	21	15	11
NW	6	5	4	3	2	9	16	22	27	31	34	37	37	46	76	108	128	119	51	32	22	16	11	8
Horiz.	8	6	5	4	3	22	62	104	147	185	214	233	239	232	212	180	137	90	50	37	27	19	14	11
												Zone type C												
N	5	5	4	4	4	24	23	24	27	30	33	34	35	34	32	29	29	34	14	10	8	7	6	6
NE	7	6	6	5	6	75	106	107	88	61	49	47	45	43	40	35	31	25	16	13	11	10	9	8
E	9	8	8	7	8	83	130	148	145	124	89	62	56	52	47	43	37	30	20	17	15	13	12	11
SE	9	8	7	6	6	45	82	107	121	121	105	82	59	51	47	42	36	29	19	16	14	13	11	10
S	7	7	6	5	5	12	18	25	36	54	70	79	79	70	54	40	33	26	16	13	12	10	9	8
SW	14	12	11	10	9	13	21	26	29	33	36	57	86	110	124	129	111	80	37	28	25	20	17	15
W	17	15	13	12	11	17	22	27	31	34	36	37	59	98	132	153	156	128	50	38	28	24	21	19
NW	12	11	10	9	8	14	20	25	29	32	34	36	36	44	73	102	118	107	39	30	21	17	15	13
Horiz.	24	21	19	17	16	34	68	107	144	175	199	212	215	207	189	160	123	83	53	44	38	34	30	27
												Zone type D												
N	8	7	6	6	6	21	21	21	24	27	29	31	32	31	30	28	29	32	17	14	12	11	10	9
NE	11	10	8	8	7	63	87	90	77	58	48	45	46	44	42	39	35	29	22	19	17	15	14	12
E	15	13	12	11	11	70	107	123	124	110	85	65	60	57	53	48	43	37	28	25	22	20	18	16
SE	14	13	11	10	10	39	68	90	102	104	95	78	60	55	51	47	42	35	27	24	21	19	17	16
S	11	10	9	8	7	12	17	21	32	46	59	67	69	63	52	41	36	30	22	19	17	15	14	12
SW	21	19	17	15	14	18	22	25	28	31	34	51	74	94	106	109	100	78	45	37	30	29	26	23
W	25	23	20	18	17	21	24	28	30	33	34	36	53	84	112	130	136	116	57	46	38	35	31	28
NW	18	16	15	13	12	17	20	24	27	30	32	35	34	43	64	87	101	94	42	35	28	25	22	20
Horiz.	35	33	30	27	24	28	64	95	124	150	171	185	191	188	176	156	128	96	72	63	56	50	45	41

Notes:

1. Values are in Btu/h·ft².
2. Apply data directly to standard double-strength glass with no inside shade.
3. Data apply to 21st day of July.
4. For other types of glass and internal shade, use shading coefficients as multiplier. For externally shaded glass, use north orientation.

Source: ASHRAE Handbook 1997, Fundamentals. Reprinted with permission.

Appendix C

DATA SHEET OF PIC MICROCONTROLLER

PIC18FXX2

Pin Diagrams (Cont.'d)

DIP

Note: Pin compatible with 40-pin PIC16C7X devices.

DIP, SOIC

* RB3 is the alternate pin for the CCP2 pin multiplexing.

TABLE 1-3: PIC18F4X2 PINOUT I/O DESCRIPTIONS

Pin Name	Pin Number			Pin Type	Buffer Type	Description
	DIP	PLCC	TQFP			
$\overline{MCLR}$/VPP	1	2	18			Master Clear (input) or high voltage ICSP programming enable pin.
$\overline{MCLR}$				I	ST	Master Clear (Reset) input. This pin is an active low RESET to the device.
VPP				I	ST	High voltage ICSP programming enable pin.
NC	—			—	—	These pins should be left unconnected.
OSC1/CLKI	13	14	30			Oscillator crystal or external clock input.
OSC1				I	ST	Oscillator crystal input or external clock source input. ST buffer when configured in RC mode, CMOS otherwise.
CLKI				I	CMOS	External clock source input. Always associated with pin function OSC1. (See related OSC1/CLKI, OSC2/CLKO pins.)
OSC2/CLKO/RA6	14	15	31			Oscillator crystal or clock output.
OSC2				O	—	Oscillator crystal output. Connects to crystal or resonator in Crystal Oscillator mode.
CLKO				O	—	In RC mode, OSC2 pin outputs CLKO, which has 1/4 the frequency of OSC1 and denotes the instruction cycle rate.
RA6				I/O	TTL	General Purpose I/O pin.
						PORTA is a bi-directional I/O port.
RA0/AN0	2	3	19			
RA0				I/O	TTL	Digital I/O.
AN0				I	Analog	Analog input 0.
RA1/AN1	3	4	20			
RA1				I/O	TTL	Digital I/O.
AN1				I	Analog	Analog input 1.
RA2/AN2/VREF-	4	5	21			
RA2				I/O	TTL	Digital I/O.
AN2				I	Analog	Analog input 2.
VREF-				I	Analog	A/D Reference Voltage (Low) input.
RA3/AN3/VREF+	5	6	22			
RA3				I/O	TTL	Digital I/O.
AN3				I	Analog	Analog input 3.
VREF+				I	Analog	A/D Reference Voltage (High) input.
RA4/T0CKI	6	7	23			
RA4				I/O	ST/OD	Digital I/O. Open drain when configured as output.
T0CKI				I	ST	Timer0 external clock input.
RA5/AN4/$\overline{SS}$/LVDIN	7	8	24			
RA5				I/O	TTL	Digital I/O.
AN4				I	Analog	Analog input 4.
$\overline{SS}$				I	ST	SPI Slave Select input.
LVDIN				I	Analog	Low Voltage Detect Input.
RA6						(See the OSC2/CLKO/RA6 pin.)

Legend: TTL = TTL compatible input
ST = Schmitt Trigger input with CMOS levels
O = Output
OD = Open Drain (no P diode to VDD)
CMOS = CMOS compatible input or output
I = Input
P = Power

TABLE 1-3: PIC18F4X2 PINOUT I/O DESCRIPTIONS (CONTINUED)

Pin Name	Pin Number			Pin Type	Buffer Type	Description
	DIP	PLCC	TQFP			
						PORTB is a bi-directional I/O port. PORTB can be software programmed for internal weak pull-ups on all inputs.
RB0/INT0	33	36	8			
RB0				I/O	TTL	Digital I/O.
INT0				I	ST	External Interrupt 0.
RB1/INT1	34	37	9			
RB1				I/O	TTL	
INT1				I	ST	External Interrupt 1.
RB2/INT2	35	38	10			
RB2				I/O	TTL	Digital I/O.
INT2				I	ST	External Interrupt 2.
RB3/CCP2	36	39	11			
RB3				I/O	TTL	Digital I/O.
CCP2				I/O	ST	Capture2 input, Compare2 output, PWM2 output.
RB4	37	41	14	I/O	TTL	Digital I/O. Interrupt-on-change pin.
RB5/PGM	38	42	15			
RB5				I/O	TTL	Digital I/O. Interrupt-on-change pin.
PGM				I/O	ST	Low Voltage ICSP programming enable pin.
RB6/PGC	39	43	16			
RB6				I/O	TTL	Digital I/O. Interrupt-on-change pin.
PGC				I/O	ST	In-Circuit Debugger and ICSP programming clock pin.
RB7/PGD	40	44	17			
RB7				I/O	TTL	Digital I/O. Interrupt-on-change pin.
PGD				I/O	ST	In-Circuit Debugger and ICSP programming data pin.

Legend: TTL = TTL compatible input
ST = Schmitt Trigger input with CMOS levels
O = Output
OD = Open Drain (no P diode to VDD)
CMOS = CMOS compatible input or output
I = Input
P = Power

TABLE 1-3: PIC18F4X2 PINOUT I/O DESCRIPTIONS (CONTINUED)

Pin Name	Pin Number			Pin Type	Buffer Type	Description
	DIP	PLCC	TQFP			
						PORTC is a bi-directional I/O port.
RC0/T1OSO/T1CKI	15	16	32			
RC0				I/O	ST	Digital I/O.
T1OSO				O	—	Timer1 oscillator output.
T1CKI				I	ST	Timer1/Timer3 external clock input.
RC1/T1OSI/CCP2	16	18	35			
RC1				I/O	ST	Digital I/O.
T1OSI				I	CMOS	Timer1 oscillator input.
CCP2				I/O	ST	Capture2 input, Compare2 output, PWM2 output.
RC2/CCP1	17	19	36			
RC2				I/O	ST	Digital I/O.
CCP1				I/O	ST	Capture1 input/Compare1 output/PWM1 output.
RC3/SCK/SCL	18	20	37			
RC3				I/O	ST	Digital I/O.
SCK				I/O	ST	Synchronous serial clock input/output for SPI mode.
SCL				I/O	ST	Synchronous serial clock input/output for I²C mode.
RC4/SDI/SDA	23	25	42			
RC4				I/O	ST	Digital I/O.
SDI				I	ST	SPI Data In.
SDA				I/O	ST	I²C Data I/O.
RC5/SDO	24	26	43			
RC5				I/O	ST	Digital I/O.
SDO				O	—	SPI Data Out.
RC6/TX/CK	25	27	44			
RC6				I/O	ST	Digital I/O.
TX				O	—	USART Asynchronous Transmit.
CK				I/O	ST	USART Synchronous Clock (see related RX/DT).
RC7/RX/DT	26	29	1			
RC7				I/O	ST	Digital I/O.
RX				I	ST	USART Asynchronous Receive.
DT				I/O	ST	USART Synchronous Data (see related TX/CK).

Legend: TTL = TTL compatible input
ST = Schmitt Trigger input with CMOS levels
O = Output
OD = Open Drain (no P diode to VDD)
CMOS = CMOS compatible input or output
I = Input
P = Power

TABLE 1-3: PIC18F4X2 PINOUT I/O DESCRIPTIONS (CONTINUED)

Pin Name	Pin Number			Pin Type	Buffer Type	Description
	DIP	PLCC	TQFP			
						PORTD is a bi-directional I/O port, or a Parallel Slave Port (PSP) for interfacing to a microprocessor port. These pins have TTL input buffers when PSP module is enabled.
RD0/PSP0	19	21	38	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD1/PSP1	20	22	39	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD2/PSP2	21	23	40	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD3/PSP3	22	24	41	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD4/PSP4	27	30	2	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD5/PSP5	28	31	3	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD6/PSP6	29	32	4	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
RD7/PSP7	30	33	5	I/O	ST TTL	Digital I/O. Parallel Slave Port Data.
						PORTE is a bi-directional I/O port.
RE0/$\overline{RD}$/AN5	8	9	25	I/O		
RE0 $\overline{RD}$					ST TTL	Digital I/O. Read control for parallel slave port (see also $\overline{WR}$ and $\overline{CS}$ pins).
AN5					Analog	Analog input 5.
RE1/$\overline{WR}$/AN6	9	10	26	I/O		
RE1 $\overline{WR}$					ST TTL	Digital I/O. Write control for parallel slave port (see $\overline{CS}$ and $\overline{RD}$ pins).
AN6					Analog	Analog input 6.
RE2/$\overline{CS}$/AN7	10	11	27	I/O		
RE2 $\overline{CS}$					ST TTL	Digital I/O. Chip Select control for parallel slave port (see related $\overline{RD}$ and $\overline{WR}$).
AN7					Analog	Analog input 7.
V_{SS}	12, 31	13, 34	6, 29	P	—	Ground reference for logic and I/O pins.
V_{DD}	11, 32	12, 35	7, 28	P	—	Positive supply for logic and I/O pins.

Legend: TTL = TTL compatible input
ST = Schmitt Trigger input with CMOS levels
O = Output
OD = Open Drain (no P diode to V_{DD})
CMOS = CMOS compatible input or output
I = Input
P = Power

Appendix D

DATA SHEET OF L298

L298

DUAL FULL-BRIDGE DRIVER

- OPERATING SUPPLY VOLTAGE UP TO 46 V
- TOTAL DC CURRENT UP TO 4 A
- LOW SATURATION VOLTAGE
- OVERTEMPERATURE PROTECTION
- LOGICAL "0" INPUT VOLTAGE UP TO 1.5 V (HIGH NOISE IMMUNITY)

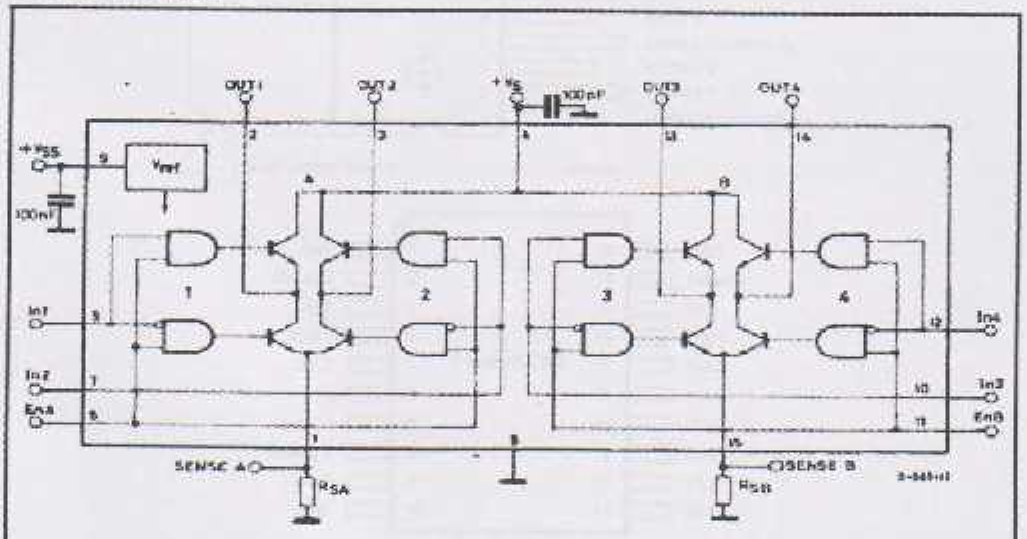
Multiwatt15 PowerSO20

ORDERING NUMBERS : L298N (Multiwatt Vert.)
L298HN (Multiwatt Horiz.)
L298P (PowerSO20)

DESCRIPTION

The L298 is an integrated monolithic circuit in a 15-lead Multiwatt and PowerSO20 packages. It is a high voltage, high current dual full-bridge driver designed to accept standard TTL logic levels and drive inductive loads such as relays, solenoids, DC and stepping motors. Two enable inputs are provided to enable or disable the device independently of the input signals. The emitters of the lower transistors of each bridge are connected together and the corresponding external terminal can be used for the connection of an external sensing resistor. An additional supply input is provided so that the logic works at a lower voltage.

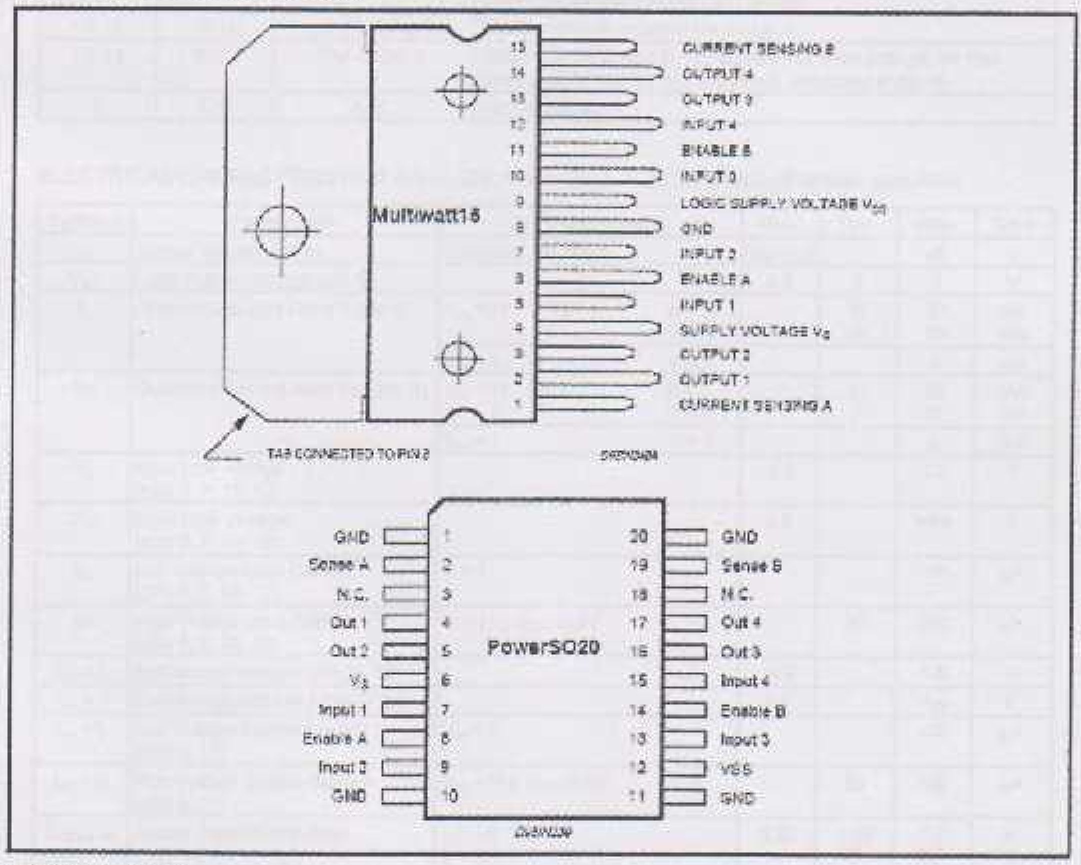
BLOCK DIAGRAM



ABSOLUTE MAXIMUM RATINGS

Symbol	Parameter	Value	Unit
V_S	Power Supply	50	V
V_{SS}	Logic Supply Voltage	7	V
V_I, V_{en}	Input and Enable Voltage	–0.3 to 7	V
I_O	Peak Output Current (each Channel) – Non Repetitive (t = 100μs) –Repetitive (80% on –20% off; t_{on} = 10ms) –DC Operation	 3 2.5 2	 A A A
V_{sens}	Sensing Voltage	–1 to 2.3	V
P_{tot}	Total Power Dissipation (T_{case} = 75°C)	25	W
T_{op}	Junction Operating Temperature	–25 to 130	°C
T_{stg}, T_j	Storage and Junction Temperature	–40 to 150	°C

PIN CONNECTIONS (top view)



THERMAL DATA

Symbol	Parameter		PowerSO20	Multiwatt15	Unit
$R_{th\ j\text{-}case}$	Thermal Resistance Junction-case	Max.	–	3	°C/W
$R_{th\ j\text{-}amb}$	Thermal Resistance Junction-ambient	Max.	13 (*)	35	°C/W

(*) Mounted on aluminum substrate

PIN FUNCTIONS (refer to the block diagram)

MW.15	PowerSO	Name	Function
1;15	2;19	Sense A; Sense B	Between this pin and ground is connected the sense resistor to control the current of the load.
2;3	4;5	Out 1; Out 2	Outputs of the Bridge A; the current that flows through the load connected between these two pins is monitored at pin 1.
4	6	V_S	Supply Voltage for the Power Output Stages. A non-inductive 100nF capacitor must be connected between this pin and ground.
5;7	7;9	Input 1; Input 2	TTL Compatible Inputs of the Bridge A.
6;11	8;14	Enable A; Enable B	TTL Compatible Enable Input: the L state disables the bridge A (enable A) and/or the bridge B (enable B).
8	1,10,11,20	GND	Ground.
9	12	VSS	Supply Voltage for the Logic Blocks. A100nF capacitor must be connected between this pin and ground.
10; 12	13;15	Input 3; Input 4	TTL Compatible Inputs of the Bridge B.
13; 14	16;17	Out 3; Out 4	Outputs of the Bridge B. The current that flows through the load connected between these two pins is monitored at pin 15.
–	3;18	N.C.	Not Connected

ELECTRICAL CHARACTERISTICS (V_S = 42V; V_{SS} = 5V, T_j = 25°C; unless otherwise specified)

Symbol	Parameter	Test Conditions		Min.	Typ.	Max.	Unit
V_S	Supply Voltage (pin 4)	Operative Condition		V_{IH} +2.5		46	V
V_{SS}	Logic Supply Voltage (pin 9)			4.5	5	7	V
I_S	Quiescent Supply Current (pin 4)	V_{en} = H; I_L = 0	V_i = L V_i = H		13 50	22 70	mA mA
		V_{en} = L	V_i = X			4	mA
I_{SS}	Quiescent Current from V_{SS} (pin 9)	V_{en} = H; I_L = 0	V_i = L V_i = H		24 7	36 12	mA mA
		V_{en} = L	V_i = X			6	mA
V_{iL}	Input Low Voltage (pins 5, 7, 10, 12)			–0.3		1.5	V
V_{iH}	Input High Voltage (pins 5, 7, 10, 12)			2.3		VSS	V
I_{iL}	Low Voltage Input Current (pins 5, 7, 10, 12)	V_i = L				–10	µA
I_{iH}	High Voltage Input Current (pins 5, 7, 10, 12)	V_i = H ≤ V_{SS} –0.6V			30	100	µA
V_{en} = L	Enable Low Voltage (pins 6, 11)			–0.3		1.5	V
V_{en} = H	Enable High Voltage (pins 6, 11)			2.3		V_{SS}	V
I_{en} = L	Low Voltage Enable Current (pins 6, 11)	V_{en} = L				–10	µA
I_{en} = H	High Voltage Enable Current (pins 6, 11)	V_{en} = H ≤ V_{SS} –0.6V			30	100	µA
$V_{CEsat(H)}$	Source Saturation Voltage	I_L = 1A I_L = 2A		0.95	1.35 2	1.7 2.7	V V
$V_{CEsat(L)}$	Sink Saturation Voltage	I_L = 1A (5) I_L = 2A (5)		0.85	1.2 1.7	1.6 2.3	V V
V_{CEsat}	Total Drop	I_L = 1A (5) I_L = 2A (5)		1.80		3.2 4.9	V V
V_{sens}	Sensing Voltage (pins 1, 15)			–1 (1)		2	V

Appendix E

Programming

Program 1

```
#include <p18f452.h>
#include "timers.h"
#include "pwm.h"
#include "adc.h"
#include "motor.h"
#pragma config OSC = HS
#pragma config WDT = OFF
#pragma config LVP = OFF

void main(void)
{
int kp1=2,kp2=2;
float Tmes1,Tmes2,error1,error2,Tset1=23.0,Tset2=23.0,speed1=0.0,speed2=0.0;
ADCON1=0x0D;
TRISA=TRISA & 0b00000011;
OpenADC(ADC_FOSC_64 & ADC_RIGHT_JUST & ADC_8ANA_0REF,
ADC_INT_OFF );
TRISB=TRISB & 0b00000000;
PORTB=0x00;
Motor_int();
while(1) {

//==========================Reading sensors ================================/

        SetChanADC(ADC_CH0);
  ConvertADC();
  while(BusyADC()==1);
  Tmes1=ReadADC();
 SetChanADC(ADC_CH1);
  ConvertADC();
  while(BusyADC()==1);
  Tmes2=ReadADC();

//=========================================Error===========================/

Tmes1=Tmes1*0.1286;//convert number to temp.
error1=Tset1-Tmes1;
Tmes2=Tmes2*0.1286;
error2=Tset2 - Tmes2;
```

```
if ( error1<0.009 && error1>=0 )
        {
        speed1=0;                       //stop the motor 1
        motor1(speed1,'c');
        }
if ( error2<0.009 && error2>=0 )
        {
        speed2=0;                       //stop the motor 2
        motor2(speed2,'c');
        }
//===========closing motor1=========================================/

if ( error1 > 0 )
        {
        speed1= error1*kp1 ;                    //closing the motor 1
        motor1(speed1,'c');
        }
if ( error2 > 0 )
        {
        speed2= error2*kp2 ;                    //closing the motor 2
        motor2(speed2,'c');
        }
//==========================openingmotor1==========================/

if ( error1 < 0 )
        {
        speed1= error1*kp1 * -1;                        //closing the motor 1
        motor1(speed1,'o');
        }
if ( error2 < 0 )
        {
        speed2= error2*kp2 * -1;                        //closing the motor 2
        motor2(speed2,'o');
        }

}//while

}//main
```

Program 2

```
#include <p18f452.h>
#include "timers.h"
#include "pwm.h"
#include "adc.h"
#include "motor.h"
#include "math.h"
#pragma config OSC = HS
#pragma config WDT = OFF
#pragma config LVP = OFF

void main(void)
{
int kp3=7;
float flow=0.2, Psensor1, Psensor2, Psensor3,flow1, flow2, flowexcess, Pset3,error3,
speed;
ADCON1=0x0A;
TRISA=TRISA & 0b00000011;
OpenADC(ADC_FOSC_64 & ADC_RIGHT_JUST & ADC_8ANA_0REF,
ADC_INT_OFF );
TRISB=TRISB & 0b00000000;
Motor_int();
while(1) {

//=========================Reading sensors ========================/

SetChanADC(ADC_CH0);
   ConvertADC();
   while(BusyADC()==1);
   Psensor1=ReadADC();
  SetChanADC(ADC_CH1);
   ConvertADC();
   while(BusyADC()==1);
   Psensor2=ReadADC();
  SetChanADC(ADC_CH2);
   ConvertADC();
        while(BusyADC()==1);
        Psensor3=ReadADC();

//====================Converting positions into flows==================/

Psensor1=Psensor1*0.00523;
Psensor2=Psensor2*0.00523;
Psensor3=Psensor3*0.00523;
```

```
flow1=0.1944-0.1944*cos(Psensor1);
flow2=0.1944-0.1944*cos(Psensor2);
flowexcess=0.2-(flow1+flow2);


//==============================Error3 ==============================/

Pset3=acos(1-flowexcess/0.1944);//convert flowexcss into position setpoint.
error3=Pset3-Psensor3;

if (error3 >= 0 && error3 <= 0.009)
        {
        speed=0;            //stop motor 3
        motor1(speed,'c');
        }
//==============================closingmotor3==============================/

        if ( error3 > 0 )
                {
                speed= error3*kp3 ;                     //closing  the motor 3
                motor1(speed,'c');

                }

//==============================openingmotor3==============================/

        if ( error3 < 0 )
                {
                speed= error3*kp3 * -1;                         //closing  the motor 1
                motor1(speed,'o');
                }

}//while
}//main
```