ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF ELECTRICAL AND COMPUTER ENGINEERING

Non-Linear Adaptive Control of Hydropower Generator Cooling System

A Thesis Submitted to the School of Electrical and Computer Engineering of Addis Ababa Institute of Technology, School of Graduate Studies, Addis Ababa University in partial fulfillment of the Requirement for the Degree of

Masters of Science in Control Engineering
By

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Advisor: Dereje Shiferaw (Ph.D.)

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DECLARATION

I, the undersigned, declare that this thesis is my original work, and has not been presented for a degree in this or other universities, and all sources of materials used for this thesis work have been fully acknowledged.

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ABSTRACT

This thesis describes the use of nonlinear adaptive control to design the cooling system of hydropower generators. The increasing load demands are posing serious threats to reliable operation of power systems. These days, in the light of the attention given to reduction of carbon dioxide emissions and efficient usage of raw materials, high-efficiency electric generators are in the lime light over the world. As all losses generate heat it is important to have a cooling system that can maintain a sufficiently low operating temperature within all parts of the generator.

The objective of this research was to model, design and simulate a reliable and efficient cooling system for hydropower generators using nonlinear adaptive control. As the load on a generator varies with time, the generators heat dissipation also varies. This variation in the heat causes the temperature of the air surrounding the generator in the closed generator room to vary. In this thesis this variation of temperature is studied and heat transfer analysis method is used to determine the heat loss of the generators. Following that appropriate size of butterfly valve is chosen which is driven by a DC servomotor and a tubular heat exchanger is designed using dimensionless number analysis method which includes determining the Reynolds number, Prandtl number, and Nusselt number. Then the required amount of cooling water is supplied through the tubular heat exchanger according to the load on the generator.

In this research the cooling system is designed to keep the generator temperature below 50 ^{0}C in other words keeping the temperature of the surrounding air inside the generator room below 27 ^{0}C. Matlab Simulink software is applied for performance analysis of the designed cooling system. The simulation results show that the designed cooling system keeps the surrounding air at a temperature of 27 ^{0}C with a settling time of 0.6 seconds and without any overshoot when it exceeds this set point (27 ^{0}C). For this study we choose the Koka Hydropower Plant and the required data for the design is collected from there. The butterfly valve opening angle for adjusting the valve at the desired position in order to allow the required flow rate is related with the temperature of the generator room in a non-linear way specifically having an exponential relation. Thus an ANN controller is used which accounts for this process nonlinearity so that increasing model robustness and reducing the generator stoppage time due to cooling system failure. This results in increasing the efficiency of the hydropower plant as a whole.

Keywords: Nonlinear adaptive control, Reynolds number, Prandtl number, Nusselt number, ANN.

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LIST OF ABBREVIATIONS

ANN	Artificial Neural Network
IEA	International Energy Agency
NTU	Number of Transfer Units
HEPP	Hydro Electric Power Plant
DC	Direct Current
AC	Alternating Current
PPR	Pulses per Revolution
EELPA	Ethiopia Electric Light Power Authority
RTD	Resistance Temperature Detector

CHAPTER ONE

1. INTRODUCTION

1.1 Background

Energy is one of the most fundamental elements of our universe. It is inevitable for survival and indispensable for development activities to promote education, health, transportation and infrastructure for attaining a reasonable standard of living and is also a critical factor for economic development and employment. In the last decade, problems related to energy crisis such as oil crisis, climatic change, electrical demand and restrictions of whole sale markets have risen world-wide. These difficulties are continuously increasing, which suggest the need of technological alternatives to assure their solution. One of these technological alternatives is generating electricity as near as possible of the consumption site, using the renewable energy sources that do not cause environmental pollutions such as wind, solar, and hydro-electric power plants. Hydro-electric power is a form of renewable energy resource, which comes from the flowing water.

The first hydro-electric power systems were developed in the 1870's. According to the International Energy Agency (IEA), large-scale hydroelectric plants currently supply 23% of the world's electricity. Hydroelectric stations are particularly well suited to automatic control because of the relative simplicity of control of both the prime mover and the hydraulic energy which drives it. Generators are sophisticated machines used to convert the mechanical energy of a prime mover into electrical energy [5]. Electromagnetic and mechanical losses accompany this energy conversion process. One of the factors that determine the lifetime of a generator is temperature. As all losses generate heat it is important to have a cooling system that can maintain a sufficiently low operating temperature within all parts of the generator. To maintain performance and reliability, regular maintenance checks of a generator’s cooling system must be conducted. Lack of maintenance of the cooling system can cause the generator to lose power and perform below its designed power output, create excessive wear, degrade lubricating oil and at worst, result in a seizure of this critical unit. Site ambient operating temperatures can vary considerably and it is important to ensure that no over-heating (boil) occurs while the unit is operating, while at the same time protecting the engine during freezing conditions. The heat generated must have an adequate heat transfer system that is reliable and in good working order and the system must be pressurized in order to permit operation at higher temperatures.

The generator line can be provided with three different cooling methods depending on the ambient conditions; (1) open ventilated: where the ambient air is used directly for cooling, (2) air-to-water cooler: where the internal air in the generator is cooled by a top mounted water cooler, and (3) air-to-air cooler: where the internal air is cooled by the ambient air through a top mounted tube heat exchanger. Most hydro electric generating sets need cooling water for bearings, generator air coolers and governor oil coolers [6]. The basic requirement is for a supply of clean water at 3-10 m head in sufficient quantities. As a first approximation, generators require a cooling water flow of about one liter per second per MVA. The design temperature rise of the water is usually between 5 and 10 °C [6]. The electric resistances of the coils and the windings are temperature dependent. This means that in order to keep the generator at its highest efficiency, it should work in its normal design temperature range [4]. Having a complete picture of the losses, the ventilation flow and the temperatures inside the machine is a key prerequisite for an optimal design of cooling system [5]. Control of highly nonlinear processes such as cooling processes is usually a challenging task. Discontinuities in control actions upon switching may lead to instabilities and, therefore, achieving bump less transitions is always a concern [2]. Linear adaptive predictive controllers need to cope with nonlinearities by using high adaptation speeds, often leading to model vulnerability in the presence of aggressive perturbations [2]. Nonlinear model-based adaptive controller accounts for process nonlinearities. As a result adaptation speed can be dramatically reduced, therefore increasing model robustness.

1.2 Literature Review

In [1], the researchers' try to study the effect of temperature fluctuations of cooling water entering the cooling system of DEZ Hydropower plant, which is one of the oldest and most important Iran's hydropower plants which contains 8 rpm units of 65 MW which leads to 520 MW of total capacity, via analysis of energy (NTU method) and exergy. The exergy analysis has been done by definition of efficiency coefficient for exergy of cooling media (water) and heating media (air). From the research they found out that the radiator installed in the DEZ power plant's generator has an NTU of one at which heat removal of the radiator rises from 215.7 kW at 301 k to 289.7 kW at 289 k. As far as cooling media concerned, increase in temperature point of entering water flow to the radiator leads to fall in efficiency coefficient of fluid exergy. In this way, they concluded that the higher the temperature point at the inlet, the higher the efficiency coefficient of exergy is. Therefore, the higher the inlet fluid temperature is, the lower the heat removability of the fluid is, and vice versa.

The severe problem with the existing Cooling Water System of Marsyangdi Hydropower Plant which is in Nepal due to excessive silt which is coming with the flowing water in the open loop of Cooling Water system is tried to be overcome by introducing a close loop circuit along with an open loop circuit of cooling water supply which passes through a Tubular Heat Exchanger directly submerged into Draft Tube water where heat transfer could takes place [3]. They design a heat exchanger in the form of a cooling coil or a Shell and Tube type of heat exchanger (without Shell). Design of the Tubular Heat Exchanger is carried out using heat transfer correlation functions and convection laws. As a result they proposed that this new system would generate about 540,000 kWh of additional energy.

Central power systems and services suggest that a generator cooling system has mostly five primary components: radiator, thermostat and housing, cooling fan, hoses, and water pump. A radiator consists of top and bottom tanks and a core, installed on or close to the generator in a vertical mode. The radiator cooling fan forces ambient air over the core to remove coolant heat and maintain heat within generator parameters. Radiators can also have extra capacity for charge air-to-air cooling, generator oil coolers, auxiliary hydraulic oil coolers and cooled – exhaust gas recirculation. The thermostat is located under the thermostat housing to regulate coolant flow by opening up when the generator has reached its operating temperature. It is closed when the generator is cold to allow the generator to warm up more rapidly. The thermostat housing is located on the outlet side of the coolant flow from the generator. The majority of thermostat housings is an aluminum alloy, and will corrode away instead of the generator or cylinder head. This is known as a 'sacrificial component'. Most cooling fans are the fixed blade variety with one air-flow direction – with either "pull" or "push" direction. These fans provide maximum airflow at all times without regard for the actual ambient temperature or how hard the generator is working. A controller system constantly monitors the generator coolant temperature and adjusts the fan pitch in order to provide adequate cooling airflow which is dependent on the actual ambient temperature of the load. There are two radiator hoses - a top radiator hose is attached to the thermostat housing, which allows the heated coolant to enter the top or inlet side of the radiator. The bottom or lower radiator hose is connected between the outlet of the radiator and the inlet of the water pump. The radiator hoses and by-pass hoses are held in position by clamps attached to the generator block or radiator assembly. These can be spring clamps, wire wound clamps, or worm drive clamps. The water pump is bolted to the front of the generator block. The bottom radiator hose comes from the radiator and is connected to the water pump inlet. The water pump is usually driven by the generator via a drive belt. As the coolant leaves the outlet of the radiator, which has removed much of its heat, the water pump forces it through the water jackets by the action of the impeller in the pump.

As it is described in [6], as a first approximation, generators require a cooling water flow of about one liter per second per MVA. The design temperature rise of the water must be usually between 5 and 10 °C and the coolers must be designed to cope with a certain amount of fouling.

In this thesis nonlinear adaptive control is used to control the cooling system of a hydropower generator. The heat transfer mechanism between the generator surrounding air and the cooling water is analyzed using heat transfer laws and calculations are made based on them. Appropriate sized valves and tubes are designed and the necessary flow rates will be flown through the tube according to the temperature of the generator. Then finally, Artificial Neural Network will be used to control the cooling system ensuring that the generator is working on its normal operating temperatures.

1.3 Statement of the Problem

A common approach to cooling system of hydropower generators involves deploying a set of operation point range-specific controllers, whose actions are to be combined in a switching strategy. However, discontinuities in control actions upon switching lead to instabilities in the system.

And in using radiators as a cooler for the hydropower generators, the failures in the cooling system are not identified easily and mostly correction of the failure is done by try and error resulting in a less efficient cooling mechanism.

In addition, linear adaptive controllers need to cope with nonlinearities by using high adaptation speeds, often leading to model vulnerability in the presence of aggressive perturbations.

In this thesis, a nonlinear adaptive controller will be developed which accounts for process nonlinearities so that increasing the efficiency and reliability of the hydropower generator cooling system.

1.4 Objectives

1.4.1 General Objective

The objective of this thesis is to model, design and simulate a reliable and efficient cooling system for hydropower generators using Artificial Neural Network which is a nonlinear adaptive control.

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1.4.2 Specific Objectives

- To model the hydropower generators cooling system.
- To design a cooling system for hydropower generators.
- To design a reliable and efficient cooling system using Artificial Neural Network.
- To simulate and analyze the performance of the cooling control system in terms of steady state and transient response using MATLAB software.

1.5 Methodology

The methodology of this research employed to undertake the study starts with literature review. This includes reading of books, publications, and different kinds of papers and thesis works to gain basic information and ideas that will help in developing a well structured system. Based on this, heat transfer analysis method is chosen and identification of the major components of the proposed cooling system proceeds. Mathematical modeling of each component of the cooling system is illustrated then.

For the designing of the overall nonlinear adaptive controller, the Artificial Neural Network, for the hydropower generator cooling system, data collection procedure is undertaken at Koka HEPP. There the data is gathered from documented readings, structural drawings, and by having interviews with the employees. Using the collected data, the overall cooling system is designed. Simulation of the design is then done using MATLAB software.

The simulation results are represented graphically and based on this results analysis and interpretation of the outcomes of the research are explained. Finally conclusions are made based on the analysis and recommendation is presented for the designed non linear adaptive control of hydropower generator cooling system.

1.6 Organization of the Thesis

Chapter one introduces about the concept of cooling of hydropower generators and reviews the different literatures conducted on the cooling system of a hydropower generator. The methodology and the general and specific objectives of the thesis along with brief description of the work are also mentioned in this section of the paper. Chapter two deal with the mathematical modeling of the proposed cooling system. In this chapter the components used in the cooling system for the hydropower generator are identified and modeled accordingly. Chapter three presents the overall design procedures for the proposed cooling system. Heat transfer calculations along with identification of the proper size of the butterfly valve and DC servo specifications are presented in this session. Also All the designed parameters for the hydropower generator cooling system based on the data collected from the KOKA HEPP are described. Chapter four describes the simulation studies done by using Matlab Simulink software. Here the designed cooling system is analyzed based on the simulation results. Chapter five discusses the conclusions drawn and the recommendation on further work for designing a hydropower generator cooling system based on this thesis.

CHAPTER TWO

2. MATHEMATICAL MODELING OF COOLING SYSTEM COMPONENTS

A mathematical model is defined as a set of equations that describes the behavior of the system. It is the art of translating problems from an application area into tractable mathematical formulations whose theoretical and numerical analysis provides insight, answers and guidance useful for the originating application. The working principle of the proposed cooling system for a hydropower generator is ensuring the flow rate of the cooling water to be proportional to the load on the generator which in turn is expressed by changes in the temperature of the surrounding air. Based on this temperature change, rate of heat transfer calculations are made and required flow rate that keeps the generator at its normal operating point is known.

The major components used in the cooling system in order to perform this operation are a butterfly valve, which allows the required flow rate of the water, a DC servomotor, that is used to adjust the butterfly valve at the required position, and an Artificial Neural Network, which acts as a controller that controls the DC servomotor depending on the calculations made using the change in temperature of the surrounding air of the generator room which is a closed system. The mathematical modeling of these components is described next.

2.1 Mathematical Modeling of Butterfly Valves

A valve is a mechanical device that controls the flow of fluid and pressure within a system or process. A valve controls system or process fluid flow and pressure by performing any of the following functions:

- Stopping and starting fluid flow
- Varying (throttling) the amount of fluid flow
- Controlling the direction of fluid flow
- Regulating downstream system or process pressure
- Relieving component or piping over pressure

There are many valve designs and types that satisfy one or more of the functions identified above. Regardless of type, all valves have the following basic parts: the **body**, **bonnet**, **trim** (internal elements), **actuator**, and **packing**.

The **body**, sometimes called the shell, is the primary pressure boundary of a valve. It serves as the principal element of a valve assembly because it is the framework that holds everything together. It resists fluid pressure loads from connecting piping. It receives inlet and outlet piping through threaded, bolted, or welded joints. Valve bodies are cast or forged into a variety of shapes.

The **bonnet** is the cover for the opening in the valve body. The bonnet is the second principal pressure boundary of a valve. In some designs, the body itself is split into two sections that bolt together. Some bonnets function simply as valve covers, while others support valve internals and accessories such as the stem, disk, and actuator. It is cast or forged of the same material as the body and is connected to the body by a threaded, bolted, or welded joint. In all cases, the attachment of the bonnet to the body is considered a pressure boundary. This means that the weld joint or bolts that connect the bonnet to the body are pressure-retaining parts. Valve bonnets, although a necessity for most valves, represent a cause for concern. Bonnets can complicate the manufacture of valves, increase valve size, and represent a significant cost portion of valve cost, and are a source for potential leakage.

The internal elements of a valve are collectively referred to as a valve's **trim** which includes a ***disk, seat, stem***, and ***sleeves*** needed to guide the stem. A valve's performance is determined by the disk and seat interface and the relation of the disk position to the seat. Because of the trim, basic motions and flow control are possible. In rotational motion trim designs, the disk slides closely past the seat to produce a change in flow opening. In linear motion trim designs, the disk lifts perpendicularly away from the seat so that an annular orifice appears.

Disk, Seat and stem

For a valve having a bonnet, the **disk** is the third primary principal pressure boundary. The disk provides the capability for permitting and prohibiting fluid flow. With the disk closed, full system pressure is applied across the disk if the outlet side is depressurized. For this reason, the disk is a pressure-retaining part. Disks are typically forged and, in some designs, hard-surfaced to provide good wear characteristics. A fine surface finish of the seating area of a disk is necessary for good sealing when the valve is closed. Most valves are named, in part, according to the design of their disks.

The **seat** or seal rings provide the seating surface for the disk. In some designs, the body is machined to serve as the seating surface and seal rings are not used. In other designs, forged seal rings are threaded or welded to the body to provide the seating surface. To improve the wear-resistance of the seal rings, the surface is often hard-faced by welding and then machining the contact surface of the seal ring. A fine surface finish of the seating area is necessary for good sealing when the valve is closed. Seal rings are not usually considered pressure boundary parts because the body has sufficient wall thickness to withstand design pressure without relying upon the thickness of the seal rings.

The **stem**, which connects the actuator and disk, is responsible for positioning the disk. Stems are typically forged and connected to the disk by threaded or welded joints. For valve designs requiring stem packing or sealing to prevent leakage, a fine surface finish of the stem in the area of the seal is necessary. Typically, a stem is not considered a pressure boundary part. Connection of the disk to the stem can allow some rocking or rotation to ease the positioning of the disk on the seat. Alternately, the stem may be flexible enough to let the disk position itself against the seat. However, constant fluttering or rotation of a flexible or loosely connected disk can destroy the disk or its connection to the stem. The two types of valve stems are rising stems and non rising stems. For a rising stem valve, the stem will rise above the actuator as the valve is opened.

This occurs because the stem is threaded and mated with the bushing threads of a yoke that is an integral part. There is no upward stem movement from outside the valve for a non rising stem design. For the non rising stem design, the valve disk is threaded internally and mates with the stem threads.

The **actuator** operates the stem and disk assembly. An actuator may be a manually operated hand wheel, manual lever, motor operator, solenoid operator, pneumatic operator, or hydraulic ram. In some designs, the actuator is supported by the bonnet. In other designs, a yoke mounted to the bonnet supports the actuator. Except for certain hydraulically controlled valves, actuators are outside of the pressure boundary. Yokes, when used, are always outside of the pressure boundary.

Packing is commonly a fibrous material (such as flax) or another compound (such as Teflon) that forms a seal between the internal parts of a valve and the outside where the stem extends through the body. Most valves use some form of packing to prevent leakage from the space between the stem and the bonnet. Valve packing must be properly compressed to prevent fluid loss and damage to the valve's stem. If a valve's packing is too loose, the valve will leak, which is a safety hazard. If the packing is too tight, it will impair the movement and possibly damage the stem.

Because of the diversity of the types of systems, fluids, and environments in which valves must operate, a vast array of valve types have been developed. Some of the common types are the globe valve, gate valve, ball valve, plug valve, butterfly valve, diaphragm valve, check valve, pinch valve, and safety valve. Each type of valve has been designed to meet specific needs. Some valves are capable of throttling flow, other valve types can only stop flow, and others work well in corrosive systems, and others handle high pressure fluids. Each valve type has certain inherent advantages and disadvantages.

Although all valves have the same basic components and function to control flow in some fashion, the method of controlling the flow can vary dramatically. In general, there are four methods of controlling flow through a valve.

1. Move a disc, or plug into or against an orifice (for example, globe or needle type valve).
2. Slide a flat, cylindrical, or spherical surface across an orifice (for example, gate and plug valves).
3. Rotate a disc or ellipse about a shaft extending across the diameter of an orifice (for example, a butterfly or ball valve).
4. Move a flexible material into the flow passage (for example, diaphragm and pinch valves).

A butterfly valve is a rotary motion valve that is used to stop, regulate, and start fluid flow. Butterfly valves are easily and quickly operated because a 90^0 rotation of the handle moves the disk from a fully closed to fully opened position. Larger butterfly valves are actuated by hand wheels connected to the stem through gears that provide mechanical advantage at the expense of speed. Butterfly valves possess many advantages over gate, globe, plug, and ball valves, especially for large valve applications. Savings in weight, space, and cost are the most obvious advantages. The maintenance costs are usually low because there are a minimal number of moving parts and there are no pockets to trap fluids. Butterfly valves are especially well-suited for the handling of large flows of liquids or gases at relatively low pressures and for the handling of slurries or liquids with large amounts of suspended solids. Butterfly valves are built on the principle of a pipe damper. The flow control element is a disk of approximately the same diameter as the inside diameter of the adjoining pipe, which rotates on either a vertical or horizontal axis. When the disk lies parallel to the piping run, the valve is fully opened. When the disk approaches the perpendicular position, the valve is shut. Intermediate positions, for throttling purposes, can be secured in place by handle-locking devices.

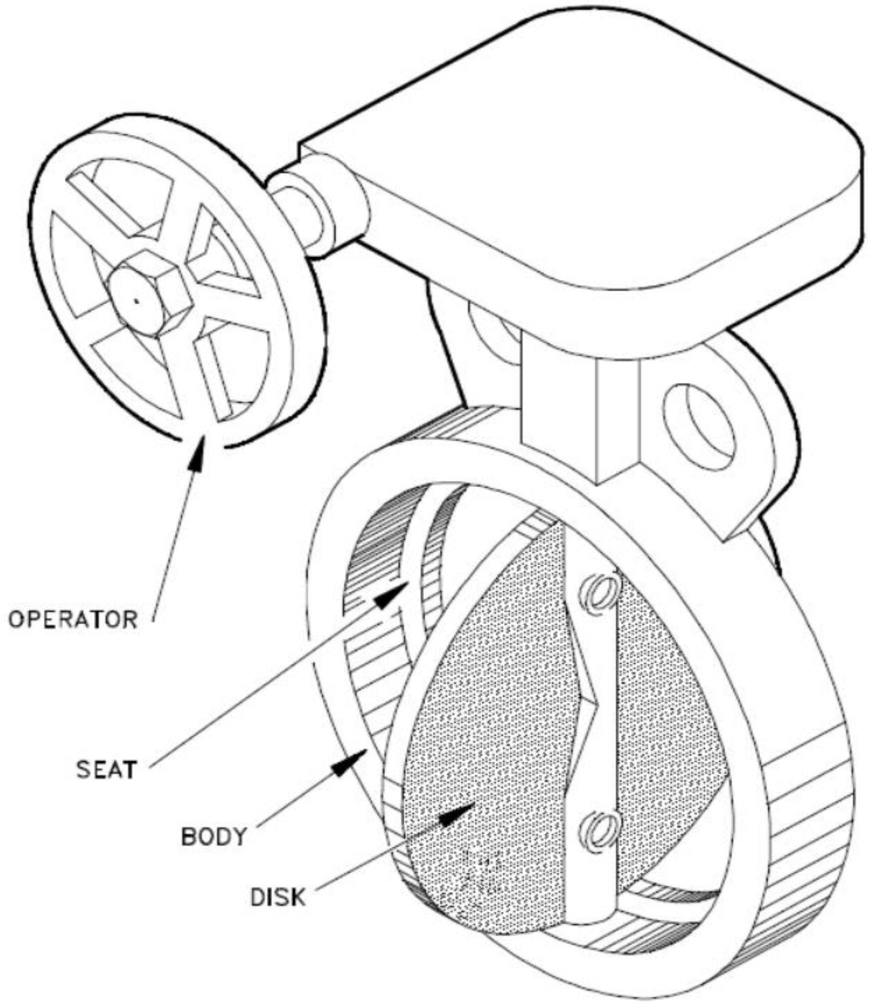


Fig.2.1 Butterfly Valve

General advantages of butterfly valves are:

- Low-pressure drop, consequently energy saving.
- Lower weight and dimensions than other types of valves.
- Maximum purchase benefits and low maintenance.
- Easy and safe operation.

- Only disc and liner are in contact with the fluid.
- Longer life which means higher profitability.
- Optimal corrosion resistance.
- Universal application options.
- Tight shut off.
- Less fluid resistance achieved through smooth flow.
- On-off or throttling applications.
- Suitable for automation with low operating torque and 90 degree operating angle.
- Quick fitting and dismantling.
- Easy replacement and renewable seat

The factor that is most useful in selecting a valve type for a given application is the flow characteristic. This characteristic is the relationship that exists between the flow rate through the valve and the valve stem travel as the latter is varied from zero to 100 percent. Different valves have different flow characteristics, depending primarily on internal construction. The characteristic that is usually graphed is the **Inherent Flow Characteristic** that is found under laboratory conditions with constant pressure drop across the valve. For a butterfly valve, the inherent equal percentage characteristic can be described by the following equation:

$$\dot{V} = \dot{V}_m R^{[\frac{X}{T}-1]} \quad (2.1)$$

Where: $\dot{V}$ = Flow Rate (gpm)
X = Valve Position
T = Maximum Valve Travel
$\dot{V}_m$ = Maximum Flow Rate (gpm)
R = Valve Rangeability

Rangeability is defined as the ratio between maximum and minimum controllable flow through the valve. Large values for rangeability are desirable because it will allow for control across a larger portion of the valve stroke.

$$\text{Rangeability} = \frac{\text{Maximum flow}}{\text{Minimum controllable flow}} \quad (2.2)$$

Valves must be sized correctly to perform the job for which they were intended. The first step in finding the size of a valve is to determine the flow coefficient (Cv) that is required for the system. Cv factor is defined as "the flow in GPM that a valve will carry with a pressure drop of 1.0 psi, when the media is 60 ºF water". This factor is determined by the construction of the valve and will not change. The required Cv factor is determined through the use of the following formula:

$$\dot{V} = Cv\sqrt{\frac{\Delta P}{Sg}} \quad (2.3)$$

Where: $\dot{V}$ = Flow Rate in gallons per minute (gpm)
Cv = Valve Flow Coefficient
ΔP = Difference in Pressure between inlet and outlet (psi)
Sg = Specific Gravity of the liquid.

The sizing pressure drop (ΔP) across the valve is measured in the valve's full open position. For the sake of controllability, this pressure drop should be as large as possible, but other system parameters prevent an excessive drop from being selected, such as available pump head, and maximum allowable pressure drop. After determining the valve Cv requirement from the preceding equation, the valve size required can be obtained.

2.2 Mathematical Modeling of DC Servomotor

The direct current (DC) motor is one of the first machines devised to convert electrical energy to mechanical energy using the principles of electromagnetism. Most of the world's adjustable speed business is addressed by DC motors. DC motor speeds can easily be varied; therefore they are utilized in applications where speed control, servo control, and/or positioning needs exist. The stator field is produced by either a field winding, or by permanent magnets. DC motors that are used in feedback controlled devices are called DC Servomotors. The word 'Servo' comes from the Latin word "Servus" meaning "servant" or "slave". Thus the servomotor is intended to react to a given command, for example a desired position or velocity. In order for the motor to be called a servomotor it has to be equipped with a velocity and position measurement unit and motor driver. It may also include a simple servomotor controller. Basically DC servomotors are classified as armature controlled DC Servomotors and field controlled DC Servomotors.

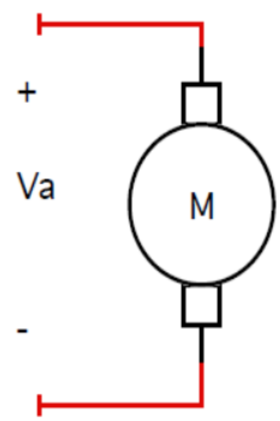


Fig.2.2 Circuit Symbol for a DC Servomotor

The main features distinguishing servomotors from other motors are:

- minimized rotor inertia
- minimized armature inductance
- improved withstand voltage and magnet saturation

- designed to withstand sudden acceleration and high frequency operation

Systems that assume motion has taken place (or is in the process of taking place) are termed "open loop". An open loop drive is one in which the signal goes "in one direction only", from the control to the motor. There is no signal returning from the motor/load to inform the control that action/motion has occurred. A stepper motor is a perfect example of an open loop system. One pulse from the control to the motor will move the motor one increment. If for some reason the stepper does not move, for example due to jamming, the control is unaware of the problem and cannot make any corrections. If a signal is returned to provide information that motion has occurred, then the system is described as having a signal which goes in "two directions": The command signal goes out (to move the motor), and a signal is returned (the feedback) to the control to inform the control of what has occurred. The information flows back, or returns. This is an example of a "closed loop" drive in which a servomotor is operating. The difference between a stepper motor and a servomotor configuration is described below.

Table 2.1. Difference between Stepper Motor and Servo Motor

	Stepper motor	**Servomotor**
Drive circuit	Simple. The user can fabricate it.	The design is very complicated
Noise and Vibration	Significant	Very little
Speed	Slow (1000 to 2000 rpm maximum)	Faster (3000 to 5000 rpm maximum)
Out-of-step Condition	Possible (will not run if too heavy load is applied)	Not possible (will rotate even if a heavier load is applied)
Control method	Open loop (no encoder)	Closed loop (uses an encoder)
Price of motor and driver	Cheap (100 to 200 US$ per set)	Expensive (more than 400 US$)

In addition, the weakness of the stepper motor is that its energy efficiency level is low and it has resonance areas which must be avoided. And in reality AC motors are more difficult to control, especially for position control, and their characteristics are quite nonlinear, which makes the analytical part more difficult. The basic reasons for using servo systems in contrast to open loop systems also include the need to improve transient response times, reduce the steady state errors and reduce the sensitivity to load parameters. Improving the transient response time generally means increasing the system bandwidth. Faster response times mean quicker settling allowing for higher machine throughput. Reducing the steady state errors relates to servo system accuracy. Finally, reducing the sensitivity to load parameters means the servo system can tolerate fluctuations in both input and output parameters. Due to this reasons armature controlled permanent magnet DC servomotor is used in this research.

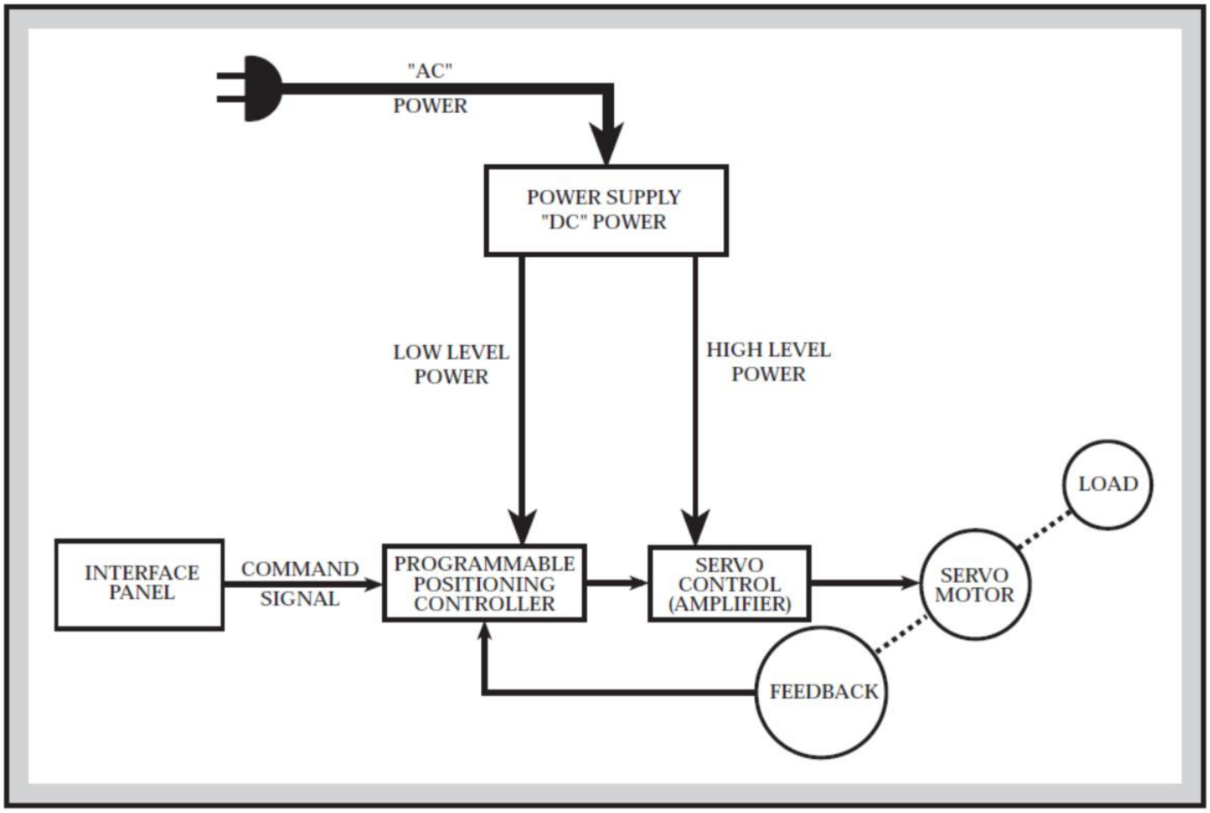


Fig. 2.3 the Concept of a Servo System

As shown in Fig. 2.3, the function, or task, of a servo can be described as follows. A command signal which is issued from the user's interface panel comes into the servo's "positioning controller". The positioning controller is the device which stores information about various jobs or tasks. It has been programmed to activate the motor/load, i.e. change speed/position. The signal then passes into the servo control or "amplifier" section. The servo control takes this low power level signal and increases, or amplifies the power up to appropriate levels to actually result in movement of the servo motor/load. Higher voltage levels are needed to rotate the servo motor at appropriate higher speeds and higher current levels are required to provide torque to move heavier loads. As power is applied onto the servo motor, the load begins to move: speed and position changes. As the load moves, so does some other "device". This other "device" is a tachometer, resolver or encoder (providing a signal which is "sent back" to the controller). This "feedback" signal is informing the positioning controller whether the motor is doing the proper job. The positioning controller looks at this feedback signal and determines if the load is being moved properly by the servo motor; and, if not, then the controller makes appropriate corrections. Therefore, a servo involves several devices. It is a system of devices for controlling some item (load). The item (load) which is controlled (regulated) can be controlled in any manner, i.e. position, direction, or speed. The speed or position is controlled in relation to a reference (command signal), as long as the proper feedback device (error detection device) is used. The feedback and command signals are compared, and the corrections made.

The equivalent circuit of armature controlled dc servomotor is shown in Fig. 2.4.

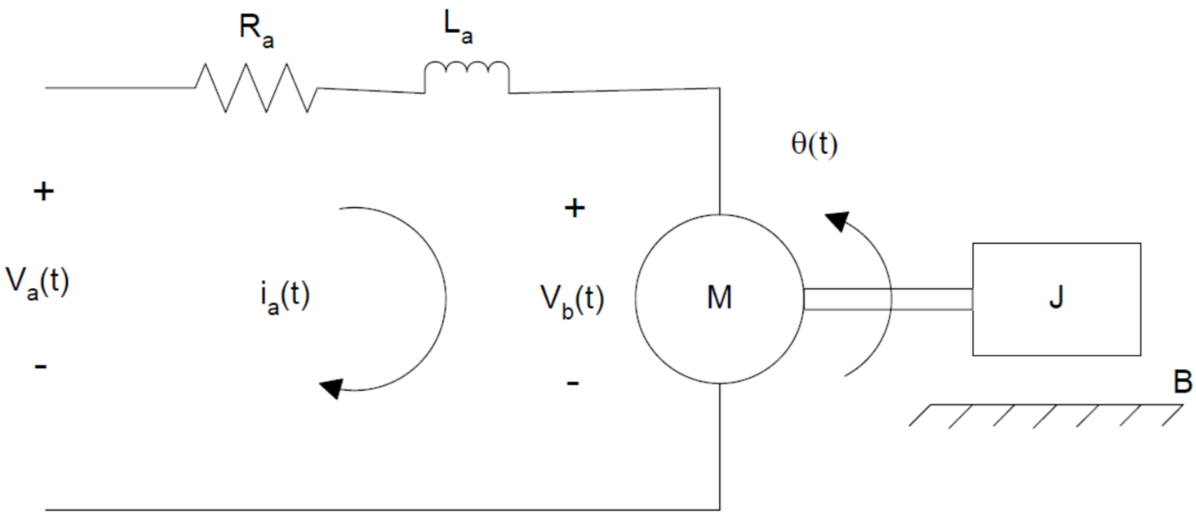


Fig.2.4 Schematic Diagram of a DC Servomotor

In order to model the DC servomotor shown in Fig.2.4, the parameters and variables are defined as follows.

R_a = armature-winding resistance, ohms
L_a = armature-winding inductance, henrys
i_a(t) = armature-winding current, amperes
V_a(t) = applied armature voltage, volts
V_b(t) = back emf, volts
$\theta(t)$ = angular displacement of the motor shaft, radians
J = moment of inertia of the motor and load referred to the motor shaft, Kg-cm^2
B = viscous-friction coefficient of the motor and load referred to the motor shaft, Nm/rpm
T = torque delivered by the motor, N-m

An armature control current is created when the armature control voltage, Va, energizes the motor. The current flows through a series-connected armature resistance, an armature inductance, and the rotational component (the rotor) of the motor. Torque is proportional to the armature current. The constant of proportionality is called the torque constant and is given the symbol K_T. For an armature controlled DC Servomotor, the field current i_f is constant and the torque T generated at the DC Servomotor shaft is given by

$$T = K_T i_a(t) \tag{2.4}$$

Where K_T is the motor torque constant (N-m/Amp).

The back electromotive force (back-emf), V_b, is a result of the rotor spinning at right angles in a magnetic field and is directly proportional to the armature angular velocity $\omega(t) = \frac{d\theta(t)}{dt}$.

$$V_b(t) = K_b \frac{d\theta(t)}{dt} \tag{2.5}$$

Where K_b is the motor back-emf constant (volt-sec/rad).

The speed of an armature controlled DC servo motor is controlled by the armature voltage Va, which is supplied by a power supply (or amplifier). A direct application of the Kirchhoff's Voltage Law to the armature circuit yields

$$L_a \frac{di_a}{dt} + R_a i_a + V_b = V_a \tag{2.6}$$

An application of Newton's moment balance equation at the motor output shaft yields

$$J \frac{d^2\theta}{dt^2} + B \frac{d\theta}{dt} = T = K_T i_a(t) \tag{2.7}$$

Assuming that all initial conditions are zero, and taking the Laplace transforms of the above differential equations, we obtain the following equations in the Laplace transform.

$$V_b(s) = K_b s\theta(s) \tag{2.8}$$
$$(L_a s + R_a) i_a(s) + V_b(s) = V_a(s) \tag{2.9}$$
$$(Js^2 + Bs)\theta(s) = T(s) = K_T i_a(s) \tag{2.10}$$

Considering $V_a(s)$ as the input, and $\theta(s)$ as the output, after rearranging equations (2.8),(2.9), and (2.10), the transfer function of the system is obtained as

$$\frac{\theta(s)}{V_a(s)} = \frac{K_T}{s(L_a J s^2 + (L_a B + R_a J)s + R_a B + K_T K_b)} \tag{2.11}$$

Block diagram representation of the DC Servomotor which is represented mathematically as in equation 2.11 is shown below.

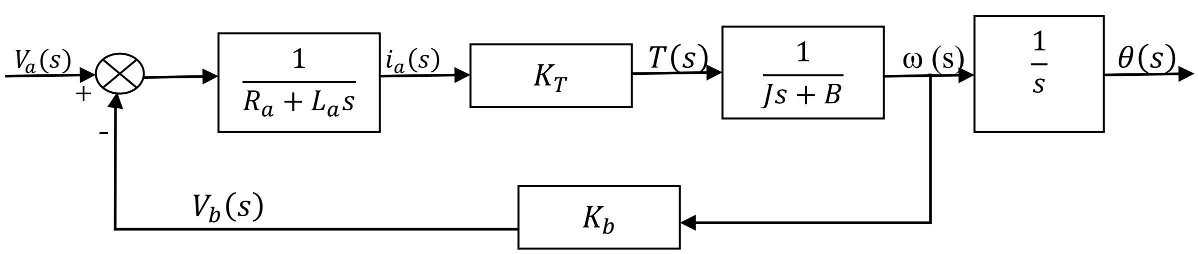


Fig.2.5 Block Diagram of a DC Servomotor System

2.3 Encoder

Encoders are mechanical to electrical transducers whose output is derived by "reading" a coded pattern on a rotating disk or a moving scale. Encoders are classified by the

- method used to read the coded element: contact or non-contact
- type of output: absolute digital word or series of incremental pulses
- physical phenomenon employed to produce the output: electrical conduction, magnetic, optical, capacitive

An encoder is a kind of pulse generator. Encoders are frequently found in modern control systems for converting linear or rotary displacement into digital coded or pulse signals. The encoders that output a digital signal are known as absolute encoders. Absolute encoders provide as an output of a distinct digital code indicative of each particular least significant increment of resolution. Incremental encoders provide a pulse for each increment of resolution but do not make distinctions between the increments. The incremental encoder provides either pulses or a sinusoidal output signal as it is rotated throughout 360 degrees. Thus distance data is obtained by counting this information. Incremental encoder's simplicity in construction, low cost, ease of application, and versatility made it by far one of the most popular encoders in control systems.

The principle of incremental encoder operation is generation of a symmetric, repeating waveform that can be used to monitor the input motion. The basic components of all optical incremental encoders are the light source, light shutter system (disk), light sensor, and signal conditioning electronics. These components will be housed and assembled to various mechanical assemblies, either rotary or linear in design depending on how motion will be monitored. The encoder's mechanical input operates the light shutter (disk) which modulates the intensity of the light at the sensor. The sensors electrical output is a function of the incident light. The encoders electrical output is produced from the sensor output by the signal conditioning electronics and can be either a sine wave, a shaped square wave, or a series of equally spaced pulses produced at regular points on the waveform.

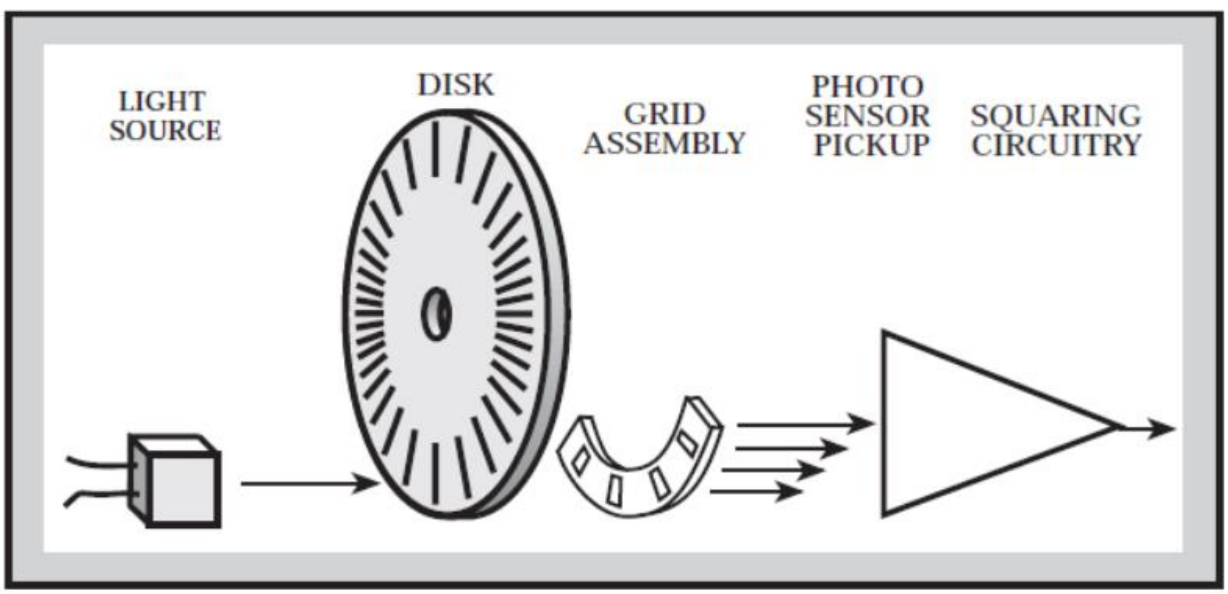


Fig.2.6 Incremental Encoder

Waveform of the sensor output is theoretically triangular but is, in practice, more nearly sinusoidal.

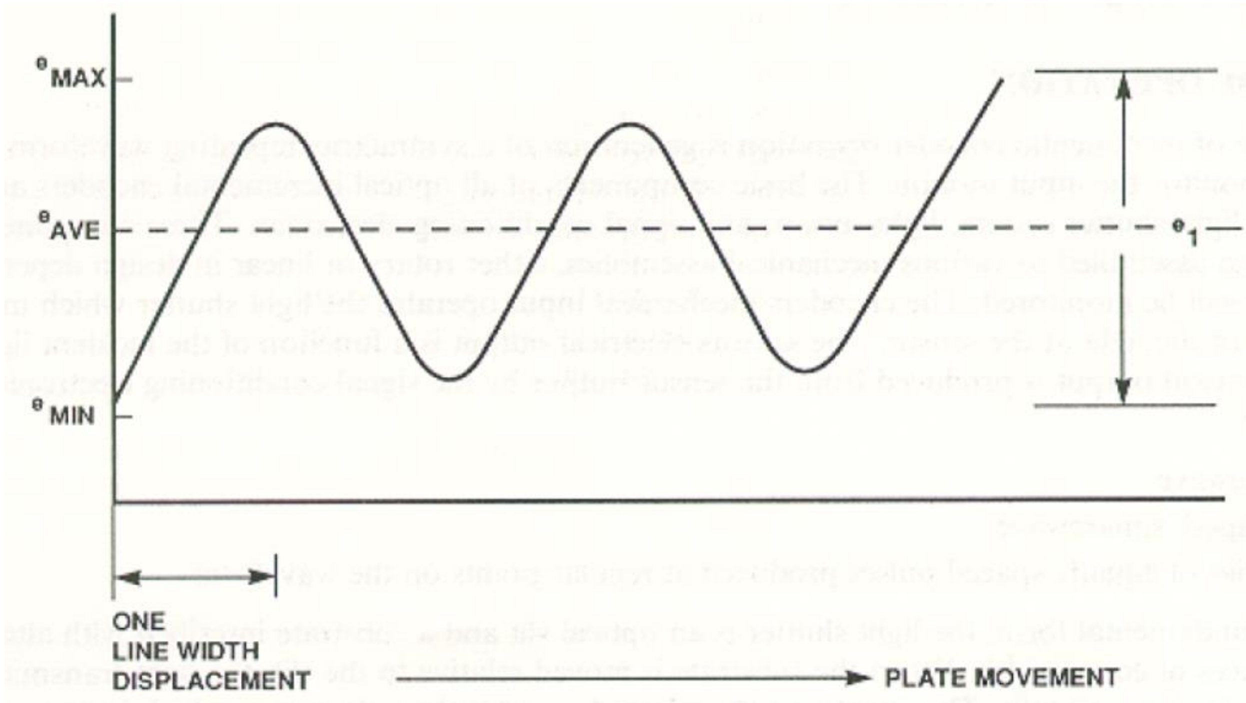


Fig.2.7 Detector Output

A common method of digitizing encoder output is to produce a pulse each time the waveform of Fig. 2.7 passes through its average value, e_{AVG}. Ideally, the waveform crossings are equally spaced and correspond to one line width displacement of the moving plate. The resulting series of equally spaced pulses can be used to precisely monitor the encoder's mechanical input.

Fig.2.8 Incremental Encoder Output

This incremental pulse is represented as a gain and is used in transforming the servo position information giving the information in terms of pulses per revolution (ppr).

2.4 Operational Amplifier

An operational amplifier or op-amp is a DC-coupled high gain electronic voltage amplifier with a differential input and, usually, a single-ended output. It is a circuit with two inputs and one output. It is a five terminal four port active element. The symbol of the op-amp with the associated terminals and ports is shown on the figure below.

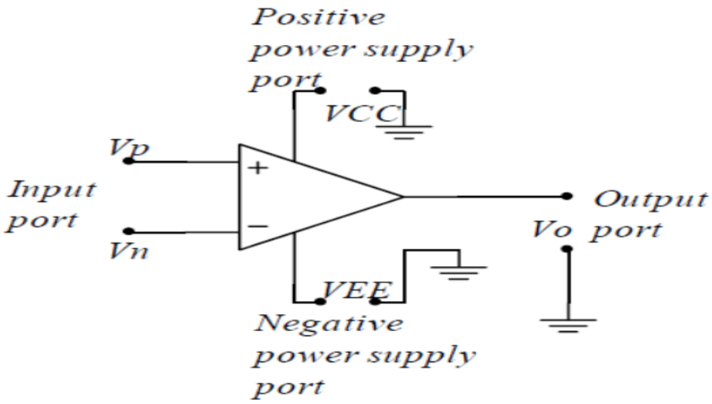
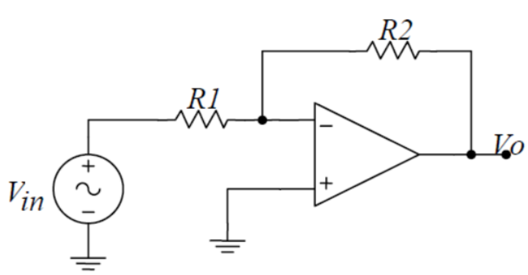


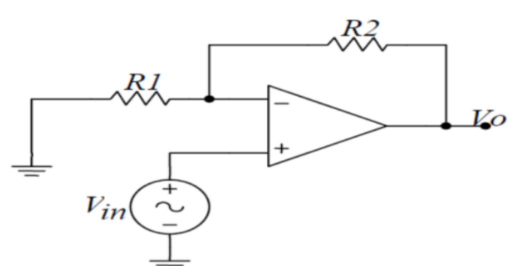
Fig.2.9 Symbol and Associated Notation of Op-amp

The power supply voltages VCC and VEE power the operational amplifier and in general define the output voltage range of the amplifier. The terminals labeled with the "+" and the "-" signs are called non-inverting and inverting respectively. The input voltage Vp and Vn and the output voltage Vo are referenced to ground.

By connecting the output terminal of the op-amp with the inverting terminal of the device a configuration called the negative feedback configuration is constructed. In negative feedback, a certain fraction of the output signal voltage, Vo, is fed back into the inverting terminal via the feedback path. With two resistors, the fundamental feedback network of a negative feedback amplifier can be constructed. Depending on the terminal at which the signal is applied, the fundamental negative feedback configuration can be in the inverting amplifier arrangement, where the input signal, Vin, is applied to the inverting terminal or in the non-inverting amplifier arrangement, where the input signal, Vin, is applied to the non-inverting terminal.



(a) Inverting amplifier



(b) Non-inverting amplifier

Fig.2.10 Negative Feedback Amplifier Configuration

In this research a non inverting amplifier is used. In a non inverting amplifier, the gain is positive and the output voltage, Vo, is in phase with the input voltage, Vin, and the gain, G, is always greater than 1. The closed loop gain of a non inverting amplifier is given by

$$G = \frac{V_o}{V_{in}} = 1 + \frac{R_2}{R_1} \tag{2.12}$$

CHAPTER THREE

3. OVERALL DESIGN OF THE COOLING SYSTEM

3.1 General Description

The working principle of the proposed cooling system is ensuring the flow rate of the cooling water to be proportional to the load on the generator. As the load on a generator changes, the heat loss on the generator windings also changes. This heat loss causes a variation on the temperature of the air inside the generator room. In this thesis this variation of temperature is studied and a proper cooling mechanism is designed.

A tubular heat exchanger is used to remove the heat dissipated from the generator. When the generator is working, there is a heat loss on the core and the windings. This heat loss is forced to the side of the generator using a fan attached to the rotor. The fan is mechanically attached to the rotor shaft at the bottom side and is driven by the rotor. The fan will keep blowing the heat towards the edges of the stator core unless the generator stops. This forced heat will be cooled by the flowing water across the tubular heat exchangers. The tubular heat exchangers are placed in such a way that circulating the stator core of the generator having an area according to the calculated heat loss of the generator.

The proposed cooling system design starts with determining the maximum heat loss of the generator. For determining this heat loss, temperature based heat transfer calculations are used. KOKA HEPP is randomly selected and heat loss of its generators is determined in this thesis. From the records of the temperature readings in the station, heat transfer calculations are done. After determining the heat loss, the size of the butterfly valve is selected. The DC Servomotor that is used to drive the butterfly valve is chosen according to the selected butterfly valve size torque requirement. Finally the integrated system is controlled by using ANN.

The general block diagram representation of the proposed cooling system is presented in Fig.3.1.

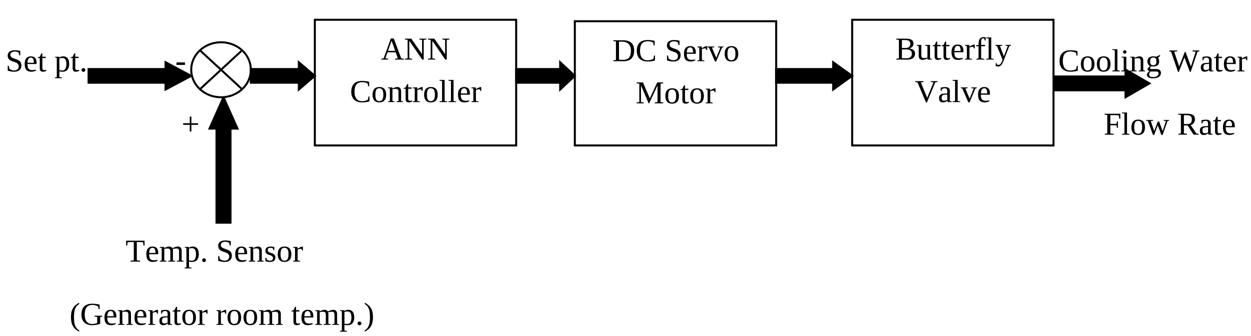


Fig.3.1 Block Diagram of the Proposed Cooling System

From the collected data the temperature of the generator room that is required in order to keep the generator at its lowest temperature (below 50 ^{0}C) is selected and is chosen as a set point for the system. The error signal generated from the reading of the temperature of the generator room and the required set point of the room is fed to the ANN controller. Proportional to this error signal, the ANN controller gives a command signal to drive the DC Servomotor. The movement of the DC Servomotor drives the butterfly valve and the required flow rate of the cooling water will flow across the tubular heat exchanger which is attached next to the butterfly valve. Thus the cooling of the generator is ensured in this manner.

3.2 Data collection

In conducting this research, the basic information that is needed in designing the hydropower generator cooling system is gathered from Koka Hydro Electric Power Plant. Koka HEPP is located on Awash river about 90km South-East of Addis Ababa in Oromia region . It was constructed by Ethiopia Electric Light Power Authority (EELPA) in 1952 E.C. At the time it was the largest hydroelectric power plant in Ethiopia which has contributed immensely to the development of the country, and still in service.The power house contains three units operating at a gross head of 212m for a total installed capacity of full load of 43.2MW. These koka hydroelectric power plant is grouped under impoundment storage stations based on type of dam structure and medium scale based on generating capacity.

The koka hydropower generator is provided with surface air coolers spaced symetrically around the periphery of the stator frame for a closed recirculating cooling system. It consists of 16 radiators arranged in a pair manner, top and bottom configuration. The cooling water is taped from the penstock.

Fig.3.2 Cooling Water Taped from Penstock

This cooling water passes through a filter and is then fed to the radiators. After the filter there is a pressure gauge that is used to monitor the filter. If the gauge reading is less than 2bar, then this indicates the filter is closed so that cleaning of the filter is done. For each generator 2 filters are used, one in use and one for a standby.

Fig.3.3 Cooling Water Filters and Their Pressure Gauge

Temperature in the generator room is taken as “cold air” and “hot air”. “Cold air” is a measurement taken near the radiator and “hot air” is a measurement of the surrounding air. These measurements are taken both in analogue using capillary sensors and RTD sensors. A record of this measurement is done every hour by the operators. Sample of the recording is shown in the appendix session.

Fig.3.4 Temperature Measurement of Generator Cooling System

The existing cooling system of Koka Hydro Electric Power Plant has no control mechanism. Simply cooling water enters to the radiators, circulates their and leaves through the outlet. The collected data's from the recorded readings and from mechanical drawings which are presented in Appendix A session are summarized and described as below.

Generator maximum rated output	12 MVA
Maximum temperature of surrounding air	45 ^{0}C
Cooling water pressure	2 bar
Velocity of circulating air	0.225 m/s
Generator room diameter	7.40 m
Generator room height	2 m
Generator diameter	5.10 m
Generator height	1.60 m
Average cooling water temperature	21 ^{0}C

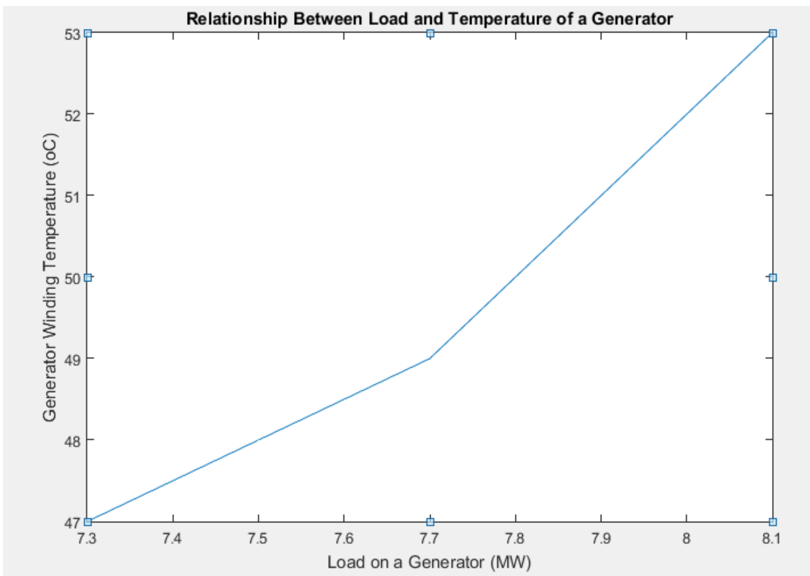


Fig.3.5 Load-Winding Temperature Characteristics of a Generator

Fig.3.5 shows the relationship between the load on a generator and the temperature of the generator windings for KOKA HEPP generators per a day. As it is seen, as the load increases so does the winding temperatures. This indicates a direct relation between the load and the winding temperatures of the generators. For this reason a cooling system that can regulate the required flow rate of the cooling water according to this load changes is going to be designed in this thesis.

3.3 Selection of Butterfly Valve Size

In order to select a specific butterfly valve, the size of the butterfly valve must be chosen carefully. The size of a butterfly valve is directly dependent on the flow rate of the flowing water through it. In this thesis in order to calculate the flow rate requirement of the generator cooling system a heat transfer rate analysis method is used. Based on the temperature recordings from the collected data the amount of rate of heat transfer is calculated. Then the required flow rate in order to remove this rate of heat transfer is determined. Using this flow rate the appropriate size of the butterfly valve is then selected.

In most hydropower stations the generator is cooled in a closed system, i.e the air is circulated there in the generator room without exiting the room. The rate of the heat transfer in the surrounding air can be calculated using the formula shown in eqn. (3.1).

$$Q = mC_p\Delta T \tag{3.1}$$

Where: Q = rate of heat transfer in the surrounding air
m = mass of surrounding air
C_p = specific heat (heat capacity) of air ($1000\,\frac{J}{K_g℃}$)
ΔT = change in temperature of the surrounding air

In order to find the mass of the surrounding air I use the following calculation using the gathered data.

$$V_1 = \pi {r_1}^2 h_1 \tag{3.2}$$

Where: V_1 = volume of generator room
r_1 = radius of the room = $\frac{7.40m}{2}$ = 3.70m
h_1 = height of the room = 2m

Then,

$$V_1 = \pi * 3.70^2 * 2$$
$$V_1 = 86.0168\, m^3$$

Similarly,

$$V_2 = \pi {r_2}^2 h_2$$

Where: V_2 = volume of the generator
r_2 = radius of the generator = $\frac{5.10m}{2}$ = 2.55m
h_2 = height of the generator = 1.60m

Then,

$$V_2 = 32.6851\, m^3$$

Approximately the volume of the surrounding air (V_3) is

$$V_3 = V_1 - V_2 = 86.0168\, m^3 - 32.6851\, m^3 = 53.3317 m^3$$

Mass of the surrounding air (m_a) is calculated using density of air ($\rho_a = 1.1455\frac{Kg}{m^3}\, at\ 35\ {}^0C$)

$$m_a = \rho_a * V_3 \tag{3.3}$$
$$m_a = 1.1455 * 53.3317 = 61.0915\, K_g$$

In this thesis we are trying to keep the generator temperature at a temperature of below 50 ^{0}C. From the collected data it is observed that the generator will be at a temperature of below 50 ^{0}C when the surrounding air is below 27 ^{0}C. So we chose 27 ^{0}C as my set point. The maximum temperature of the surrounding air is 45 ^{0}C. The rate of heat loss from the generator to the surrounding air that changes the temperature of the air from 27 ^{0}C to 45 ^{0}C is

$$Q = mC_p\Delta T$$
$$Q = 61.0915\ K_g * 1000\frac{J}{K_g℃} * (45 - 27)\frac{℃}{s}$$
$$Q = 1{,}099{,}647\ \frac{J}{s}$$

This much amount of heat transfer takes place in raising the temperature of the air from 27 ^{0}C to 45 ^{0}C. In order to keep the air temperature at 27 ^{0}C the cooling water must be able to absorb this heat loss. The mass rate of the water ($\dot{m}_w$) required to absorb the heat loss with a maximum change in temperature of 5 ^{0}C and specific heat (C_p) of $4184\,\frac{J}{K_g℃}$ is

$$\dot{m}_w = \frac{Q}{C_p\Delta T} = \frac{1{,}099{,}647\frac{J}{s}}{4184\frac{J}{K_g℃}*5℃} = 52.5644\,{}^{Kg}/_{s}$$

Using density of water ($\rho_w = 1000\frac{Kg}{m^3}$) the required flow rate ($\dot{V}$) is

$$\dot{V} = \frac{\dot{m}_w}{\rho_w} = \frac{52.5644\frac{Kg}{s}}{1000\frac{Kg}{m^3}} = 0.0525644\frac{m^3}{s} = 52.5644\,\frac{l}{s} = 693.753\,\frac{gal}{min}$$

In selecting the size of a butterfly valve the flow coefficient (Cv) that is required for the system must be first determined. Using eqn. (2.3) the flow coefficient is calculated.

$$Cv = \frac{\dot{V}}{\sqrt{\frac{\Delta P}{Sg}}}$$

Where:
Cv = Valve Flow Coefficient
$\dot{V}$ = Flow rate in gallons per minute ($693.753\,\frac{gal}{min}$)
ΔP = Difference in pressure (2 bar = 29.00755 psi)
Sg = Specific gravity of water (1).

Substituting the values, the flow coefficient gives a value of $Cv = 121.8091$. Using this value I determine the size of the butterfly valve using the standard table of flow coefficient of butterfly valve shown below.

Table 3.1 Cv Values for Butterfly Valves

Flow Rate Cv Values										
Size		Angle of Opening								
Inch	Mm	10^0	20^0	30^0	40^0	50^0	60^0	70^0	80^0	90^0
2	50	0	5	10	18	29	47	75	107	124
$2\,{}^1\!/_2$	65	0	12	22	39	64	102	163	232	270
3	80	0	17	33	57	94	149	240	341	397
4	100	0	29	55	96	158	252	404	577	671
5	125	0	44	83	145	369	381	610	871	1013
6	150	0	66	126	219	362	576	922	1318	1532
8	200	0	125	230	400	660	1050	1680	2400	2792
10	250	0	160	325	575	950	1514	2423	3462	4024
12	300	0	258	493	859	1418	2260	3618	5168	6010
14	350	0	324	617	1076	1776	2829	4530	6472	7525
16	400	0	433	826	1441	2378	3760	6068	8669	10080
18	450	0	564	1076	1876	3096	4933	7898	11283	13120
20	500	0	588	1311	2286	3774	6012	9626	13751	15990
24	600	0	1018	1942	3388	5590	8907	14688	22742	23690

The basic rule in selecting the size of a butterfly valve is selecting the size that corresponds to the calculated Cv value that exists between 30^0-60^0 valve opening positions. So looking for my Cv value of 121.8091 in the look up table between 30^0-60^0 openings, I select a **3inch** or **80mm** diameter butterfly valve for this research. Basic dimension information of the butterfly valve is shown in the appendix session.

3.4 Design of Size of Tubular Heat Exchanger (pipe)

The tubular heat exchanger will have the same diameter as that of the butterfly valve in order to fit with it. So the tube has **80mm inner diameter** and **88.90mm outer diameter** (from the butterfly valve mechanical design). Determining the length of the tube is the next task.

The proposed hydropower generator cooling system uses convection type heat transfer. Heat transfer by convection occurs in a fluid by the mixing of one portion of the fluid with another portion due to gross movements of the mass of fluid. Using **Newton's Law of Cooling** rate of heat transfer is calculated as

$$Q = UA\Delta T \tag{3.4}$$

Where: Q = rate of heat transfer
U = overall heat transfer coefficient
A = area available for the flow of heat
ΔT = difference in temperature between the water and surrounding air

The overall heat transfer coefficient for a heat transfer to a fluid flowing inside a tube is given by eqn. (3.5).

$$\frac{1}{U} = \frac{r_2}{r_1 h_1} + \frac{r_2}{k}\ln\left(\frac{r_2}{r_1}\right) + \frac{1}{h_2} \tag{3.5}$$

Where: U = overall heat transfer coefficient
r_1 = inner radius of the pipe
r_2 = outer radius of the pipe
k = thermal conductivity of the pipe material
h_1 = heat transfer coefficient of cooling water and pipe contact (inside flow)
h_2 = heat transfer coefficient of pipe and surrounding air contact (outside flow)

In order to find $h_1$and h_2 dimensionless analysis method is used. In this method three dimensionless numbers are used; **Reynolds number, Prandtl number,** and **Nusselt number**. The **Reynolds number** (**Re**) is an important dimensionless quantity in fluid mechanics used to help predict flow patterns in different fluid flow situations. At low Reynolds numbers, flows tend to be dominated by laminar (sheet-like) flow, while at high Reynolds numbers turbulence results from differences in the fluid's speed and direction, which may sometimes intersect or even move counter to the overall direction of the flow (eddy currents). These eddy currents begin to churn the flow, using up energy in the process, which for liquids increases the chances of cavitations. The Reynolds number has wide applications, ranging from liquid flow in a pipe to the passage of air over an aircraft wing. It is used to predict the transition from laminar to turbulent flow, and is used in the scaling of similar but different-sized flow situations, such as between an aircraft model in a wind tunnel and the full size version. The predictions of the onset of turbulence and the ability to calculate scaling effects can be used to help predict fluid behavior on a larger scale, such as in local or global air or water movement and thereby the associated meteorological and climatological effects.

The **Prandtl number** (**Pr**) or **Prandtl group** is a dimensionless number, named after the German physicist Ludwig Prandtl, defined as the ratio of momentum diffusivity to thermal diffusivity. It is dependent only on the fluid and the fluid state. Small values of the Prandtl number, $Pr \ll 1$, means the thermal diffusivity dominates whereas with large values, $Pr \gg 1$, the momentum diffusivity dominates the behavior. In heat transfer problems, the Prandtl number controls the relative thickness of the momentum and thermal boundary layers. When Pr is small, it means that the heat diffuses quickly compared to the velocity (momentum).

In heat transfer at a boundary (surface) within a fluid, the **Nusselt number** (**Nu**) is the ratio of convective to conductive heat transfer across (normal to) the boundary. In this context, convection includes both advection and diffusion. The conductive component is measured under the same conditions as the heat convection but with a stagnant (or motionless) fluid. A Nusselt number close to one, namely convection and conduction of similar magnitude, is characteristic of "slug flow" or laminar flow. A larger Nusselt number corresponds to more active convection, with turbulent flow.

$$N_u = \frac{Convective\ heat\ transfer}{Conductive\ heat\ transfer} = \frac{hD}{k} \tag{3.6}$$

Where: h = convective heat transfer coefficient
D = diameter of the pipe
k = thermal conductivity of the pipe material

3.4.1 Inside Flow

For the flow inside the pipe the three dimensionless numbers are calculated as below.

Reynolds Number:

$$Re = \frac{\rho v D_1}{\mu} \tag{3.7}$$

Where: ρ = density of the cooling water (1000 kg/m^3)
v = velocity of the cooling water (2 m/s)
D_1 = inner diameter of pipe (0.08 m)
μ = viscosity of the cooling water (1.002*10^{-3} kg/m.s at 20 ^{0}C)

Substituting the values gives Re = 159,680.6387. For a Reynolds Number greater than 2100 the flow is termed as turbulent flow. So for this reason the flow of the cooling water is a turbulent type of flow.

Prandtl Number:

$$Pr = \frac{C_p \mu}{k} \tag{3.8}$$

Where: C_p = specific heat of cooling water ($4184\,\frac{J}{K_g{}^\circ C}$)
μ = viscosity of the cooling water (1.002*10^{-3} kg/m.s at 20 ^{0}C)
k = thermal conductivity of the pipe material which we used copper (401 w/m ^{0}C)

And we got a value of Pr = 0.0105.

Nusselt Number:

For turbulent flow inside tubes the Nusselt number is given by:

$$Nu = 0.023 * Re^{0.8} * Pr^{n} \tag{3.9}$$

Where n = 0.4 for heating and n = 0.3 for cooling.

With Re = 159,680.6387 and Pr = 0.0105, Nu = 85.251.

Using eqn. (3.6) the heat transfer coefficient for the inside flow is determined as follows.

$$h_1 = \frac{Nuk}{D_1}$$

Substituting the values of Nu, k, and D_1 gives h_1 = 427,320.6375 w/m^2 ^{0}C .

3.4.2 Outside Flow

This deals with the flow of the surrounding air. Following the same procedure as before results

Reynolds number:

$$Re = \frac{\rho v D_2}{\mu}$$

Where:
- ρ = density of the surrounding air (1.1644 kg/m^3 at 30 ^{0}C)
- v = velocity of the surrounding air (0.225 m/s)
- D_2 = outer diameter of pipe (0.0889 m)
- μ = viscosity of the surrounding air (18.6*10^{-6} kg/m.s at 27 ^{0}C)

Substituting the values gives Re = 1,252.1995.

Prandtl number:

$$Pr = \frac{C_p \mu}{k}$$

Where:
- C_p = specific heat of surrounding air ($1000 \frac{J}{K_g °C}$)
- μ = viscosity of the surrounding air (18.6*10^{-6} kg/m.s at 27 ^{0}C)
- k = thermal conductivity of copper (401 w/m ^{0}C)

This gives a value of Pr = 0.00004638.

Nusselt number:

For heat transfer to and from gases flowing across tube banks, the Nusselt number is expressed by the equation:

$$Nu = 0.33 * C_h * Re^{0.6} * Pr^{0.3} \tag{3.10}$$

Where C_h depends on the geometrical arrangement of the tubes, in my case I have an in-line arrangement which gives a value of C_h = 1. Then with Re = 1,252.1995 and Pr = 0.00004638, Nu = 1.1939.

Using eqn. (3.6) the heat transfer coefficient for the outside flow is

$$h_2 = \frac{Nuk}{D_2}$$

Substituting the values of Nu, k, and D_2 gives h_2 = 5,385.3082 w/m$^{2\,0}$c .

Using eqn. (3.5) the overall heat transfer coefficient (U) is then determined. This gives an overall heat transfer coefficient of U = 5000.4075 w/m^{20}c. As shown in eqn. (3.4) this value of the overall heat transfer coefficient is used to determine the area of the pipe. Hence

$$A = \frac{Q}{U\Delta T}$$

The rate of heat transfer (Q) is determined in the previous section and it is found to be Q = 1,099,647 J/s. The air has a maximum temperature of 45^{0}C and the water average temperature is 25^{0}C which gives a change in temperature of ΔT = 20 ^{0}C. Solving for the area of the pipe, a 10.9956 m^2 pipe is required for the hydropower generator cooling system. The latent surface area of a pipe is calculated as

$$A = 2\pi r_2 h \tag{3.11}$$

With r_2 = the outer radius of the pipe (0.04445 m), the length required is (h)

$$h = \frac{A}{2\pi r_2} = \frac{10.9956\ m^2}{2*\pi*0.04445\ m} = 39.37\ m$$

Using a 2m long copper pipe having an inner diameter of 40mm and thickness of 4.45mm, approximately 20 pipes are used in this hydropower generator cooling system.

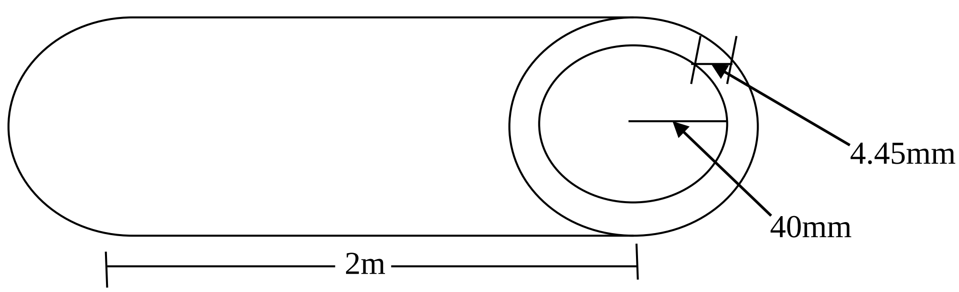


Fig.3.6 Tubular Heat Exchanger (pipe)

3.5 Butterfly Valve Torque Requirements

The amount of total torque (τ_T) required to modulate a butterfly valve disc is actually the sum of several torques which occur within the butterfly valve. Specifically:

$$\tau_T = \tau_{su} + \tau_d + \tau_{bf} + \tau_{ss} + \tau_e + \tau_h \qquad (3.12)$$

Where:
τ_{su} = Seating and Unseating Torque
τ_d = Dynamic Torque.
τ_{bf} = Bearing Friction Torque
τ_{ss} = Stem Seal Friction Torque
τ_e = Eccentricity Torque resulting from disc offset from centerline of stem
τ_h = Hydrostatic Torque

The seating/unseating torque value (τ_{su}) is a function of the pressure differential, the seat material's coefficient of friction, the finished surface of the disc edge, the amount of interference between the seat inner diameter and disc outer diameter when flanged in piping, the seat thickness, and the type of service (media) for which the valve is being used. Its value is largest in the first 20^0 of rotation while the disc and seat are in contact. After the disc clears the seat material the value of τ_{su} will drop to zero. In a symmetrical disc design, dynamic torque (τ_d) occurs between the closed position, 0^0 and the full open position, 90^0. With the disc in the partially open position, velocity of the fluid passing the leading disc edge is less than the velocity passing the trailing edge. These variance in velocity between the leading disc edge and trailing disc edge results in an unbalanced distribution of pressure forces on the upstream side of the face of the disc. The total pressure forces acting perpendicular to the disc face on the leading edge half of the disc are greater than the total pressure acting perpendicular on the trailing half of the disc. This uneven distribution of pressure on the disc face (exists on both sides of the disc) results in a torsional force which tries to turn the disc to the closed position. The magnitude of τ_d is greatest when the disc is between 75° and 85° of rotation. Bearing friction torque (τ_{bf}) occurs because the differential pressure across the valve disc generates a force which is applied against the disc and subsequently is transmitted to the stem. As the stem is forced against the bearing supports, bushing and interior body, friction forces are created between the stem and the stem supports. The overall torque requirement for a butterfly valve is primarily a function of τ_{su}, τ_d and τ_{bf}. The magnitudes of the other torque components are generally insignificant in comparison.

The two major conditions for determining total valve operating torque (τ_T) exists as follows:

CASE I (Angle = 0°, Disc in Closed Position)

$$\tau_T = \tau_{su} + \tau_{bf} + \tau_{ss} + \tau_h \qquad (3.13)$$

Stem seal friction torque values and hydrostatic torque values are insignificant when compared to seating/unseating and bearing friction torques. Bearing friction torque is determined by using the following equation:

$$\tau_{bf} = 0.21 * {D_v}^2 * \frac{d}{2} * \Delta P \tag{3.14}$$

Where: D_v = Valve Diameter, inches (3 in)
d = Diameter of Shaft, inches (from table B-1 ⌀U = 14mm = 0.5512 in)
ΔP = Differential Pressure across Valve, psi (29.00755 psi)

Substituting the values the bearing friction torque is determined to be 15.1 lb-in or 1.7 N-m. The magnitude of the required seating/unseating torque (τ_{su}) for the butterfly valve have been determined by actual testing. The result seating/unseating torque incorporating all bearing friction and stem seal friction torques for three classes of services is presented in the table below.

Table 3.2 Seating/Unseating Torque

		Valve Differential Pressure (PSIG)						
		Full Disc					Reduced Disc	
	Valve Size Inches	0 psi	50 psi	100 psi	150 psi	175 psi	0 psi	50 psi
Class A Non-Corrosive, Lubricating Service	1	54	59	65	70	73	54	59
	1.5	81	86	91	97	100	81	86
	2	109	114	119	123	128	109	114
	2.5	169	178	187	196	200	169	178
	3	220	236	250	264	273	220	236
	4	341	364	387	410	423	225	248
	5	510	560	610	660	687	324	374
	6	632	712	792	872	912	344	488
	8	1,182	1,341	1,500	1,660	1,741	735	894
	10	1,764	2,018	2,272	2,526	2,653	1,204	1,358
	12	2,701	3,110	3,519	3,928	4,132	1,665	2,074
	14	3,818	4,500	5,182	5,864	—	2,318	3,000
	16	4,638	5,819	7,000	8,182	—	2,699	3,880
	18	5,265	7,065	8,865	10,665	—	2,970	4,788
	20	7,000	9,364	11,728	14,091	—	3,356	6,243
Class B General Service	1	59	65	71	77	80	59	65
	1.5	89	95	100	106	110	89	95
	2	120	125	130	135	140	120	125
	2.5	185	195	205	215	220	185	195
	3	245	260	275	290	297	245	260
	4	375	400	425	450	462	252	267
	5	560	615	670	725	755	355	410
	6	695	783	871	953	1,003	427	537
	8	1,300	1,475	1,650	1,825	1,915	808	983
	10	1,960	2,240	2,520	2,800	2,940	1,213	1,493
	12	2,970	3,420	3,870	4,320	4,545	1,830	2,280
	14	4,200	4,950	5,700	6,450	—	2,550	3,300
	16	5,100	6,400	7,700	9,000	—	2,967	4,267
	18	5,850	7,850	9,850	11,850	—	3,267	5,267
	20	7,700	10,300	12,900	15,500	—	4,267	6,867

Class C Severe Service	1	74	82	89	97	100	74	82
	1.5	111	119	125	133	137	111	119
	2	151	157	163	169	175	151	157
	2.5	231	244	257	269	275	231	244
	3	306	325	344	363	375	306	325
	4	468	500	532	563	582	316	348
	5	700	769	838	907	944	444	513
	6	870	980	1,090	1,200	1,255	525	672
	8	1,625	1,844	2,063	2,282	2,394	1,011	1,230
	10	2,450	2,800	3,150	3,500	3,675	1,517	1,867
	12	3,712	4,275	4,838	5,400	5,682	2,287	2,850
	14	5,251	6,188	7,125	8,063	—	3,189	4,126
	16	6,375	8,000	9,625	11,250	—	3,709	5,334
	18	7,315	9,815	12,315	14,815	—	4,084	6,584
	20	9,625	12,875	16,125	19,375	—	5,334	8,584

Looking under Class B (General Service) for my butterfly valve having a diameter of 3in and with a differential pressure of 29.00755psi, the seating/unseating torque is found to be approximately 254 lb-in or 28.7 N-m. Thus τ_T = 254 lb-in = 28.7 N-m for case I.

<u>Case II</u> (Disc in Partial to Full Opening Position)

$$\tau_T = \tau_{bf} + \tau_{ss} + \tau_d \tag{3.15}$$

To determine dynamic torque (τ_d), the following equation is applied:

$$\tau_d = C_{dt} * D^3 * \Delta P \tag{3.16}$$

Where: C_{dt} = Coefficient of Dynamic Torque (based on disc shape and angle of opening) (dimensionless)
D = Diameter of Disc (inches) (3in)
ΔP = Pressure Differential across Valve (psi)(29.00755 psi)

Table 3.3 Coefficient of Dynamic Torque for Butterfly Valves

Angle of Opening	0°	10°	20°	30°	40°	50°	60°	70°	75°	80°	90°
C_{dt}	0	0.0126	0.0140	0.0251	0.0505	0.0809	0.1394	0.2384	0.3419	0.401	0

For throttling applications it is best to choose C_{dt} at an angle of 60^0. So taking C_{dt} = 0.1394, τ_d = 109.1786 lb-in = 12.34 N-m. Dynamic Torque values including all bearing friction and stem seal friction torques are shown in the torque chart described in the table below.

Table 3.4 Dynamic Torque per Pressure Drop (lb-in/psi)

Angle of Opening		10°	20°	30°	40°	50°	60°	70°	75°	80°	90°
Valve Size inches	2	0.113	0.126	0.225	0.453	0.726	1.251	2.139	3.068	3.052	0.000
	2.5	0.216	0.240	0.430	0.865	1.386	2.388	4.085	5.858	5.827	0.000
	3	0.367	0.408	0.732	1.473	2.359	4.065	6.952	9.970	9.918	0.000
	4	0.855	0.949	1.702	3.425	5.486	9.454	16.168	23.187	23.065	0.000
	5	1.650	1.833	3.287	6.613	10.593	18.253	31.217	44.769	44.533	0.000
	6	2.494	2.771	4.969	9.997	16.014	27.594	47.192	67.680	67.323	0.000
	8	6.603	6.736	12.077	24.298	38.925	67.073	114.707	164.506	163.64	0.000
	10	11.990	13.322	23.885	48.056	76.985	132.653	226.862	325.353	323.640	0.000
	12	20.892	23.214	41.619	83.735	134.142	231.141	395.295	566.910	563.925	0.000
	14	30.039	33.377	59.841	120.396	192.873	332.342	568.366	815.119	810.828	0.000
	16	45.652	50.724	90.942	182.970'	293.115	505.070	863.764	1,238.762	1,232.240	0.000
	18	65.909	73.232	131.295	264.159	423.178	729.184	1,247.040	1,788.435	1,779.020	0.000
	20	91.415	101.573	182.105	366.387	586.945	1,011.373	1,729.637	2,480.549	2,467.498	0.000

Taking the reading at 60^0 the total torque required to operate the butterfly valve for case II is

$$\tau_T = \frac{4.065\ lbin}{psi} * 29.00755 psi = 117.915 \text{ lb-in} = 13.32 \text{ N-m}$$

Comparing the two cases the final torque requirement for the butterfly valve that is going to be used in the hydropower generator cooling system is a torque of **254 lb-in** or **28.7 N-m**.

3.6 Gear Ratio

The butterfly valve is operated by the DC servomotor. For the linkage of the two (the butterfly valve and the DC servomotor) a gear system is used. Here a gear system having a ratio (N) of 40:1 is chosen. As the servo is the prime mover, speed reduction takes place. The butterfly will have $1/40^{th}$ of the speed of the DC servomotor. In contrast the torque of the DC servo is multiplied by the ratio. Hence the torque requirement of the butterfly valve was calculated as 28.7 N-m. In order to satisfy this requirement the DC servomotor must produce a minimum torque of

$$\tau_m = \frac{\tau_{bv}}{N} = \frac{28.7}{40} = 0.7175 \text{ N-m} \tag{3.17}$$

3.7 DC Servomotor Selection

Satisfying the butterfly valve torque requirement, the following DC servomotor is selected for the hydropower generator cooling system.

Table 3.5 DC Servomotor Specification

General		
Continuous Stall Torque	lb-in	8.75
	N-m	0.99
Continuous Current	amps	2.68
Peak Torque	lb-in	59
	N-m	6.7
Peak Current	amps	24.7
Viscous Damping	lb-in/krpm	0.05
	Nm/krpm	5.6 E-03
Thermal Resistance	°C/watt	2.4
Thermal Time Constant	MIn	35
Mechanical Time Constant	msec	8.77
Electrical Time Constant	msec	2.76
Rated Speed	rpm	2800
Rated Voltage	volts	100
Electrical		
Torque Constant	lb-in/amp	2.67
	N-m/amp	0.302
Voltage Constant	Vpk/krpm	31.6
	v/r/s	0.302
Resistance	ohms	2.9
Inductance	mH	8.0
Mechanical		
Inertia	lb-in-s^2	0.00244
	Kg-cm^2	2.75
Maximum Speed	rpm	4000
Weight	lbs/Kg	10/4.5

Substituting the parameter of the motor in eqn. (2.11), the reduced transfer function of the DC servomotor becomes

$$\frac{\theta(s)}{V_a(s)} = \frac{1}{0.000007285s^3+0.002642s^2+0.3025s} \tag{3.18}$$

3.8 Incremental Encoder

For the incremental encoder, a TRD-MX series type encoder is used. The series encoder having an output of 100 ppr (pulses per revolution) is used and it is supplied with 5V DC and its maximum allowable number of revolution is 6000 rpm.

3.9 Pulse to Analogue Voltage Converter

The encoder outputs pulses according to the servomotor movement. This pulse is then converted to analogue voltage so that this voltage is used as a reference for the amplifier. For this thesis, ZU252 type pulse to analogue voltage converter is chosen. This converter gives an output of 0-10V. It is adjusted so that it gives an output linearly proportional to the pulse output of the incremental encoder.

3.10 Amplifier

The DC servomotor is driven by an operational amplifier. In this design of hydropower generator cooling system, the selected DC servomotor has a rated voltage of 100 volts and the pulse to analogue voltage converter gives an output of a maximum of 10 volts. The op amp input voltage is an error voltage emerged from the difference between the command signal and the pulse to analogue voltage converter. As a result this error signal has a maximum value of 10 volts. Considering the rated voltage of the DC servomotor which is 100 volts I choose an amplifier that has a gain of 10 so that the DC servo requirement is satisfied.

In this design a non inverting operational amplifier is used. As shown in Fig.2.10b a non inverting operational amplifier having a positive voltage supply (VCC) of 15 volts, a negative voltage supply (VEE) of 15 volts, and having resistor values of R_1 = 1kΩ and R_2 = 9kΩ is selected. Using eqn.(2.12) the gain is calculated as

$$G = 1 + \frac{R_2}{R_1} = 1 + \frac{9k\Omega}{1k\Omega} = 10$$

3.11 Butterfly Valve Opening Angle

The designed hydropower generator cooling system works on the principle of allowing sufficient flow rate of cooling water in proportion to the temperature rise of the generator room which in turn is expressed in terms of heat transfer rates. In order to satisfy this principle, for each change in temperature of the generator room, the rate of heat transfer taking place in the room is calculated using eqn. (3.1). Using the same equation, the required flow rate to overcome the heating is calculated. According to this flow rate the butterfly valve is adjusted. The equation that relates the flow rate with the valve opening angle is described in eqn. (2.1).

$$\dot{V} = \dot{V}_m R^{[\frac{X}{T}-1]}$$

The maximum flow rate ($\dot{V}_m$) for a 3in diameter butterfly valve is found to be 2138 gal/min which is calculated using the Cv value (Table 3.1). The valve rangeability (R) value is presented in the table below.

Table 3.6 Butterfly Valve Rangeability Values

Valve Size, in.	Rangeability Values	
	70° Rotation	90° Rotation
2	18:1	31:1
2-1/2	18:1	31:1
3	19:1	33:1
4	25:1	42:1
5	32:1	57:1
6	39:1	70:1
8	67:1	119:1
10	80:1	147:1
12	92:1	163:1
14	92:1	163:1
16	92:1	163:1
18	92:1	163:1
20	92:1	163:1

From the table it is seen that a butterfly valve having a diameter of 3inch has a rangeability value of 33:1 for 90^0 rotations. With a maximum valve travel (T) of 90^0, the valve opening angle is calculated from the known flow rate as

$$X = 59.269 * (\log \frac{\dot{V}}{2138} + 1.51851) \tag{3.19}$$

This butterfly valve angle is multiplied by the gear ratio and the servo angle is determined. Using eqn. (3.18) the voltage that must be supplied to the DC servo is calculated. Finally this required voltage is supplied by the Artificial Neural Network to the DC Servomotor so that the hydropower generator cooling system functions as designed. Below is a table that summarizes the overall design of the hydropower generator cooling system.

Table 3.7 Overall Design of a Hydropower Generator Cooling System

Temperature Set Point = 27 ^{0}c, Gear Ratio = 40:1					
Surrounding air temperature (^{0}c)	Rate of heat transfer (Q) (J/s)	Required flow rate of cooling water ($\dot{V}$) (gal/min)	Butterfly valve opening angle (X)	DC servomotor opening angle (θ)	ANN command signal (volts)
20 – 27	0	0	0^0	0^0	0
28	61,091.5	38.54	0^0	0^0	0
29	122,183	77.08	4.473^0	178.9^0	0.732
30	183,274.5	115.6	14.91^0	596.4^0	2.44
31	244,366	154.2	22.31^0	892.6^0	3.652
32	305.457.5	192.7	28.06^0	1122^0	4.59
33	366,549	231.3	32.75^0	1310^0	5.359
34	427,640.5	269.8	36.72^0	1469^0	6.01
35	488,732	308.3	40.16^0	1606^0	6.57
36	549,823.5	346.9	43.19^0	1728^0	7.07
37	610,915	385.4	45.9^0	1836^0	7.511
38	672,006.5	424	48.35^0	1934^0	7.912
39	733,098	462.5	50.59^0	2024^0	8.28
40	794,189.5	501	52.65^0	2106^0	8.616
41	855,281	539.6	54.56^0	2182^0	8.927
42	916,372.5	578.1	56.34^0	2253^0	9.217
43	977,464	616.7	58^0	2320^0	9.491
44	1,038,555.5	655.2	59.56^0	2382^0	9.745
45	1,099,647	693.8	61.03^0	2441^0	10

The relationship between the rate of heat loss of the generators and the temperature of the air in the room is illustrated in Fig.3.7.

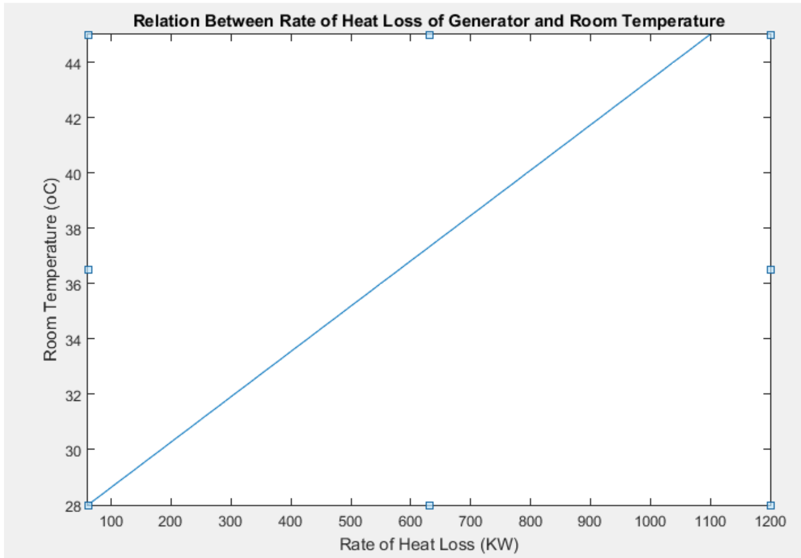


Fig.3.7 Rate of Heat Loss-Room Temperature Characteristics of a Generator

CHAPTER FOUR

4. SIMULATION STUDIES AND ANALYSIS OF RESULTS

4.1 Simulink Model

The simulink model for the proposed hydropower generator cooling system is shown in Fig. 4.1.

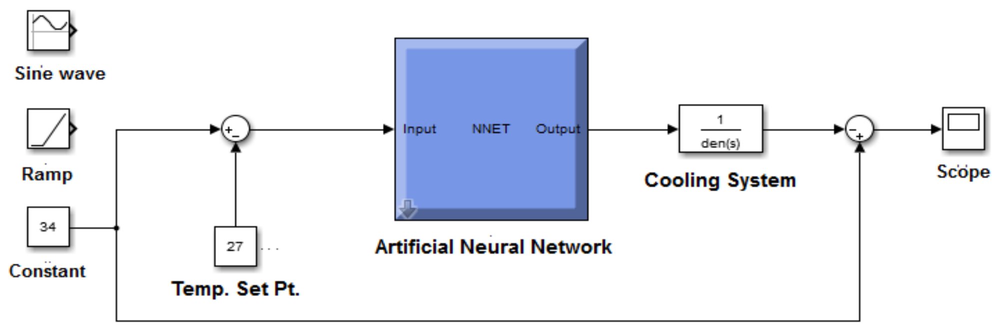


Fig. 4.1 Generalized Simulink Diagram of the Hydropower Generator Cooling System

The designed cooling system consists of the DC Servomotor with its amplifier and encoder and a butterfly valve that allows the required flow rate through the tubular heat exchanger. The detailed design of the generator cooling system using simulink model is shown in Fig. 4.2.

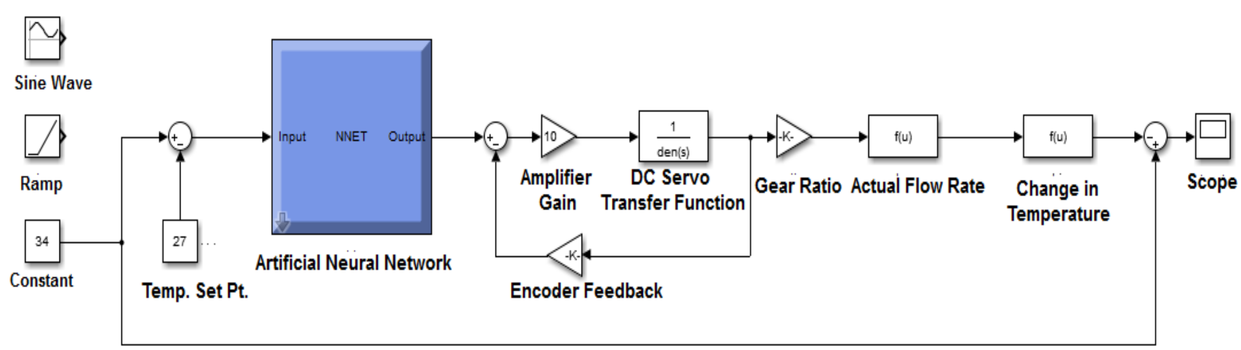


Fig. 4.2 Detailed Simulink Diagram of the Hydropower Generator Cooling System

4.2 ANN Training

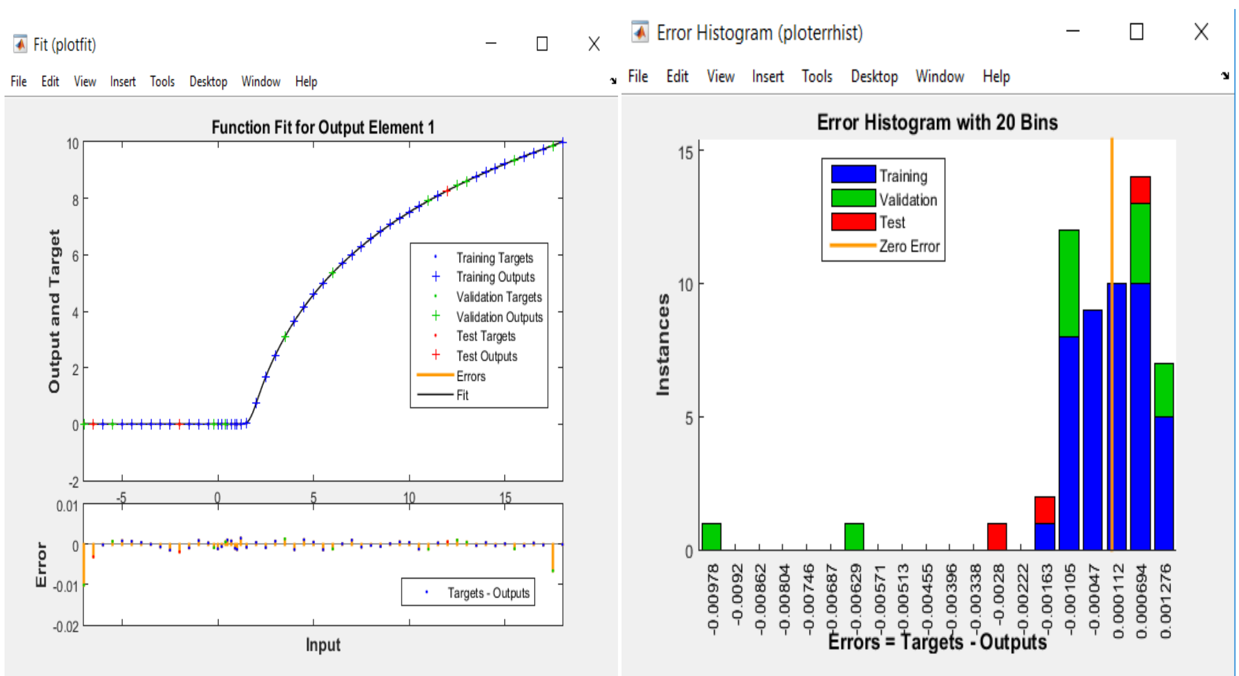
In the design, Artificial Neural Network is used as a controller to adjust the flow of the cooling water according to the surrounding air temperature. Here, a three layer (one input layer, one hidden layer, and one output layer) feed forward ANN is used. In the hidden layer twelve neurons are used. Increasing the number of neurons causes over fitting which is indicated by a good performance on the training set but a worse performance on the test set. Back propagation learning algorithm (Levenberg - Marquardt) is used to train the network. The ANN is trained using the two data's, the temperature error and the corresponding required voltage data for driving the DC servomotor. The hidden layer neurons are activated by using tan sigmoid activation function ($f(s) = \frac{1}{1+e^{-s}}$) and for the output a pure linear activation function ($f(s) = s = Wx + b$) is used.

Sample of the training data that is used to train the ANN is described in Table 4.1 below.

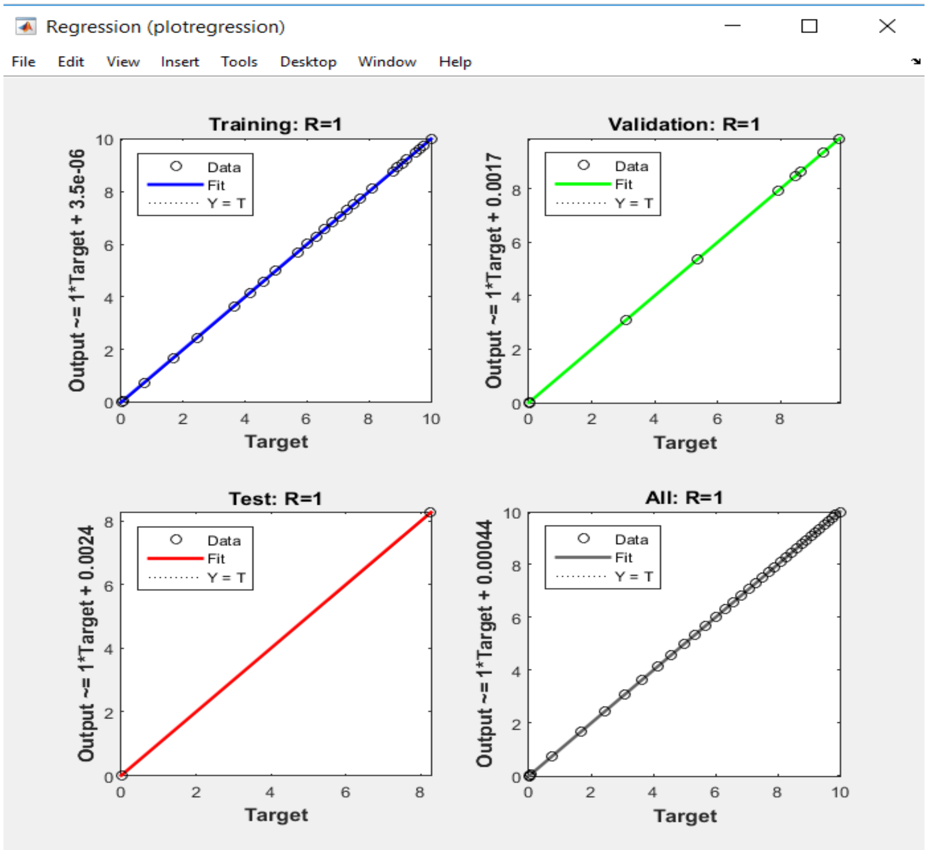
Table 4.1 Sample of Training Data

Temperature error (^{0}c)	-7	-4	0	2	4	6	8	10	12	14	16	18
Voltage Command Signal	0	0	0	0.732	3.652	5.359	6.57	7.511	8.28	8.927	9.491	10

The overall range of the training data used to train the ANN is indicated on the fit plot below. After training the ANN, the regression, error histogram and fit plots are shown in Fig. 4.3.



(a) (b)



(c)

Fig. 4.3 ANN Training Performance Output

(a) Fit Plot, (b) Error Histogram Plot, (c) Regression Plot

As it is seen in the Fig. 4.3 the error shown after training is almost zero which indicates the targets and actual output data's are matched correctly. Also in the regression plot it is found R=1 which indicates the target and the output line up perfectly. So from the plots it can be seen that the ANN is trained perfectly and is ready to perform the cooling system.

Here in these design three different types of input signals; a sinusoidal signal, a ramp signal and a constant one are used. The sinusoidal signal is used to represent a change in the temperature of the generator surrounding air at the start and stop of the generator. An increase or decrease of the temperature in a linear way is represented by using the ramp signal. For a constant load the temperature of the surrounding air will be kept constant so that a constant signal represents this phenomenon. For each type of input signals the performance of the designed cooling system is illustrated in the next session.

4.3 Response of the System for a Sine Wave Input Signal

At the starting of the generator the room temperature is at 20 ^{0}C. Then the temperature gradually increases and reaches 45 ^{0}C when the generator generates its rated output. For a decrease on the load on a generator, the temperature decreases and when the generator stops it return to the room temperature (Fig. 4.4).

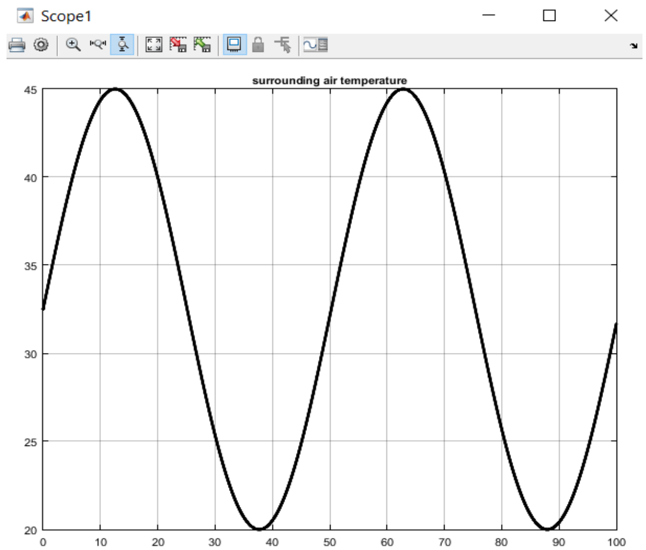


Fig.4.4 Surrounding Air Temperature (Sine Wave Signal)

For this input signal the ANN, the DC servomotor, the reference signal and the final surrounding air temperature results are illustrated in Fig 4.5.

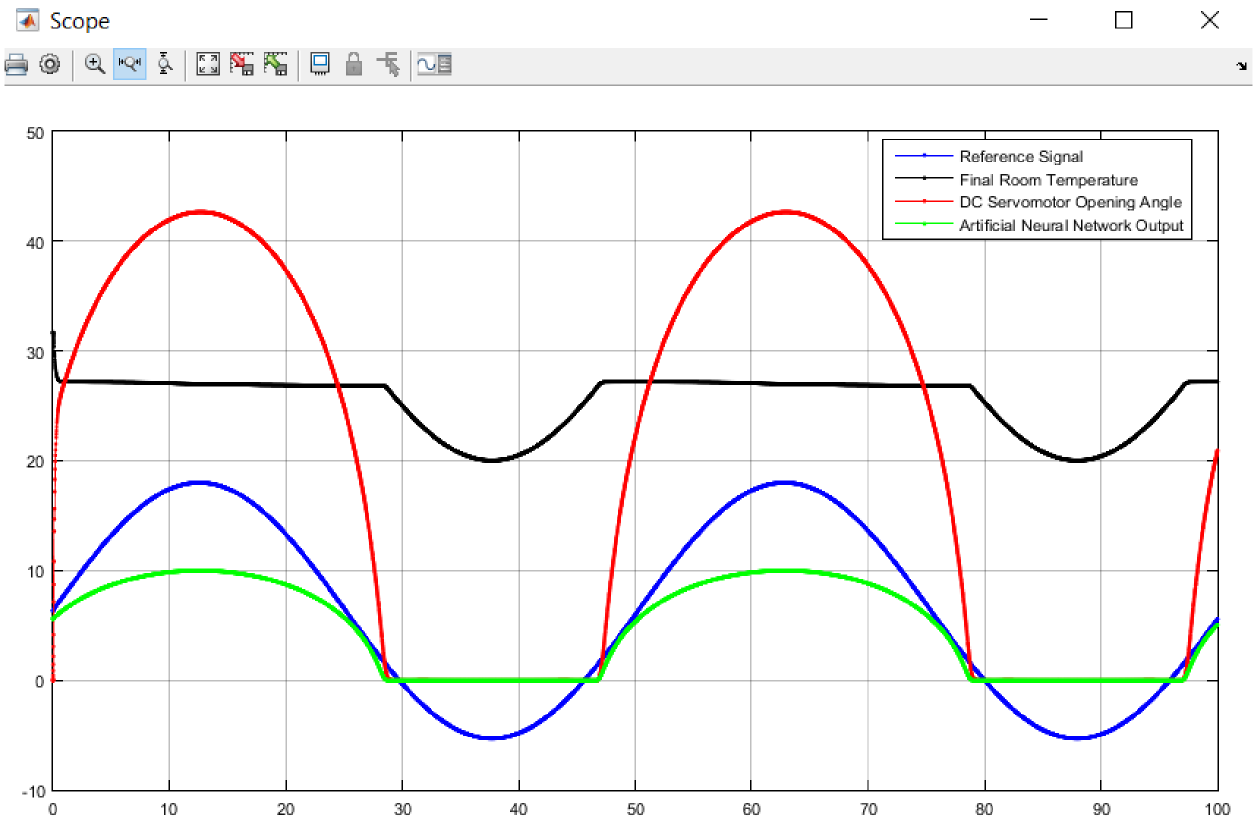


Fig.4.5 Simulink Output of the Cooling System for a Sine Wave Input Signal

From the results shown in Fig. 4.5 it is seen that the ANN gives an output of 0 – 10 volts as designed. When the reference signal is below the zero position, the ANN gives no driving force to the DC servomotor (0 volt) keeping the servo in a stop position. Thus this results the butterfly valve to be closed so that no flow of cooling water takes place keeping the final surrounding air temperature same as the input. Whereas when the air temperature exceeds the set point which is indicated by a positive reference signal, the ANN gives an output according to the trained data. This ANN output drives the DC servomotor which is coupled with the butterfly valve. Then the required flow rate of the cooling water is supplied which keeps the final temperature of the room at 27 ^{0}C as shown in the figure above. Thus for a sine wave input signal the designed cooling system works perfectly.

4.4 Response of the System for a Ramp Input Signal

When the load on the hydropower generator increases or decreases gradually with respect to time, the temperature of the surrounding air in the generator room will increase or decrease with respect to the load. This phenomenon is represented using a ramp signal (Fig. 4.6). Taking this signal as an input to the cooling system the response of the system is described below.

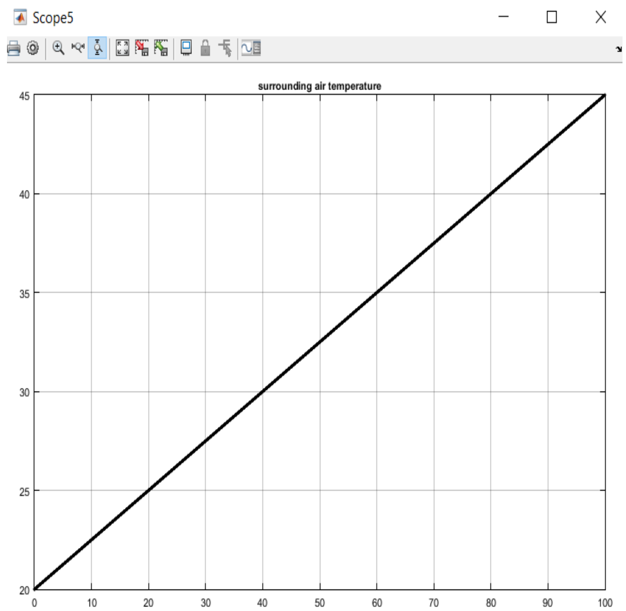


Fig.4.6 Surrounding Air Temperature (Ramp Signal)

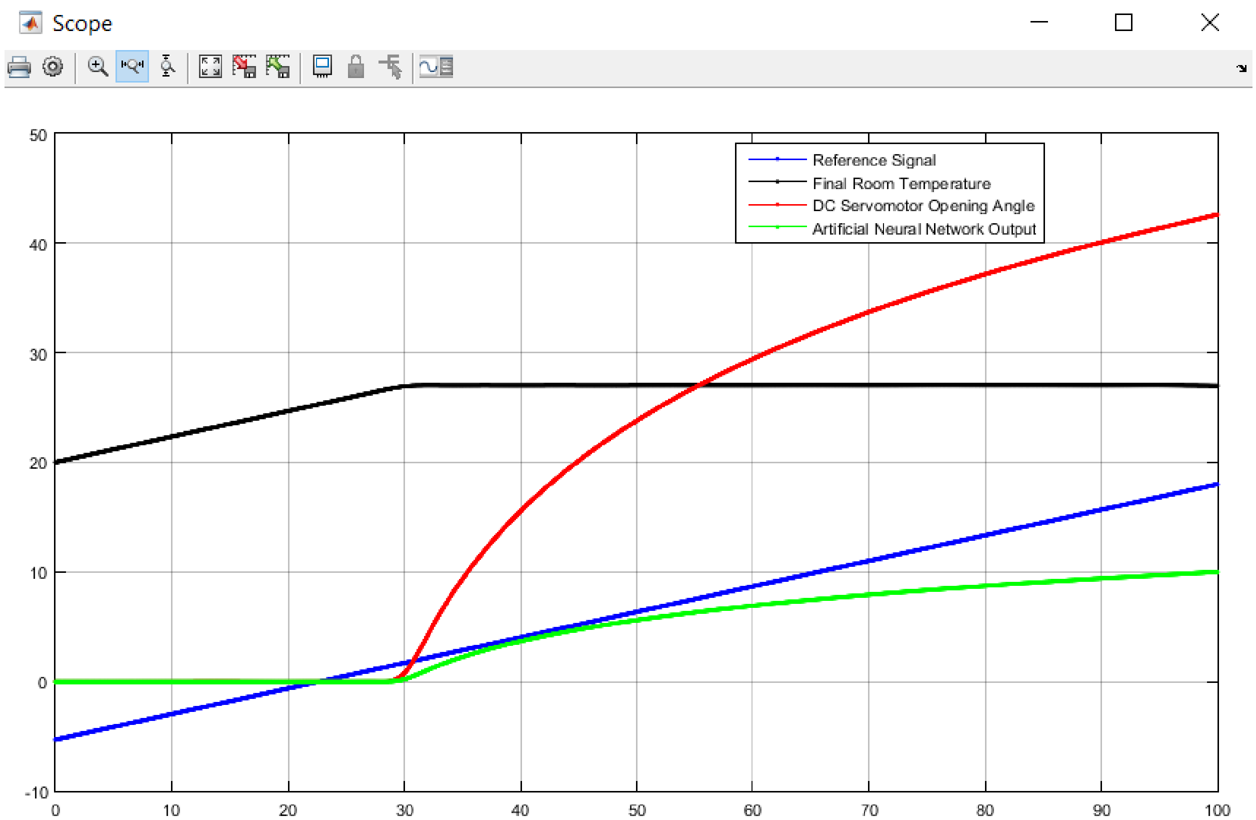


Fig.4.7 Simulink Output of the Cooling System for a Ramp Input Signal

From the result displayed in Fig. 4.7, for an increase in temperature of the surrounding air from 20 ^{0}C up to 45 ^{0}C the system responses accordingly. The ANN and the DC servomotor give an output of greater than zero (positive value) for an input signal of the air temperature reading of greater than 27 ^{0}C which is indicated in Fig. 4.7 as a positive reference signal.

In the ramp signal display it reaches 27 ^{0}C nearly at 30sec where as the system keeps the air temperature at 27 ^{0}C for the remaining time starting from 30sec. Thus keeping the temperature of the surrounding air constant keeps the generator to be below 50 ^{0}C hence enhancing the performance of the generator. The designed cooling system tries to stabilize the air temperature at a temperature of 27 ^{0}C when it exceeds the set point. For the surrounding air temperature below the set point (27 ^{0}C) the cooling system will remain dormant letting the temperature as it is. Hence the designed system is guaranteed for a ramp type change of the surrounding air temperature existing in the closed generator room.

4.5 Response of the System for a Constant Type Input Signal

The third input representing the temperature of the surrounding air is a constant type input signal (Fig. 4.8). For some period of time the load on the generator may be kept constant. In this case the heat dissipated to the air will also be constant thus representing this special case with a constant input signal will be best.

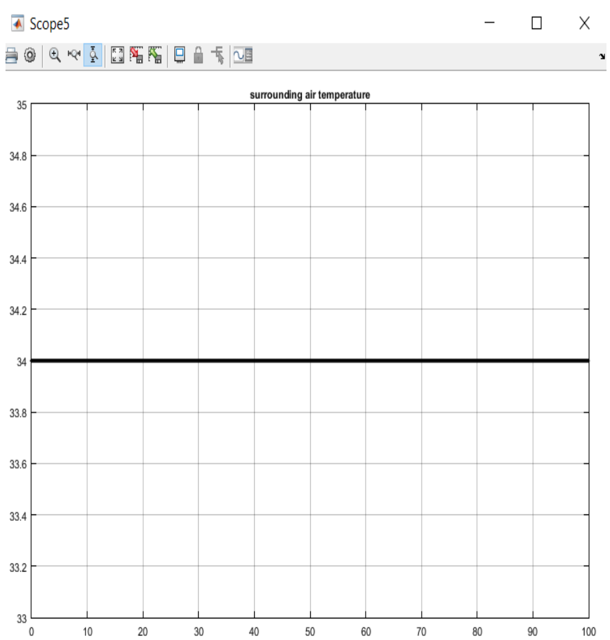


Fig.4.8 Surrounding Air Temperature (Constant Signal)

As a sample, 34 ^{0}C room temperature is randomly selected as an input to the designed cooling system. Below is a display of the system results for this constant input signal source (Fig. 4.9).

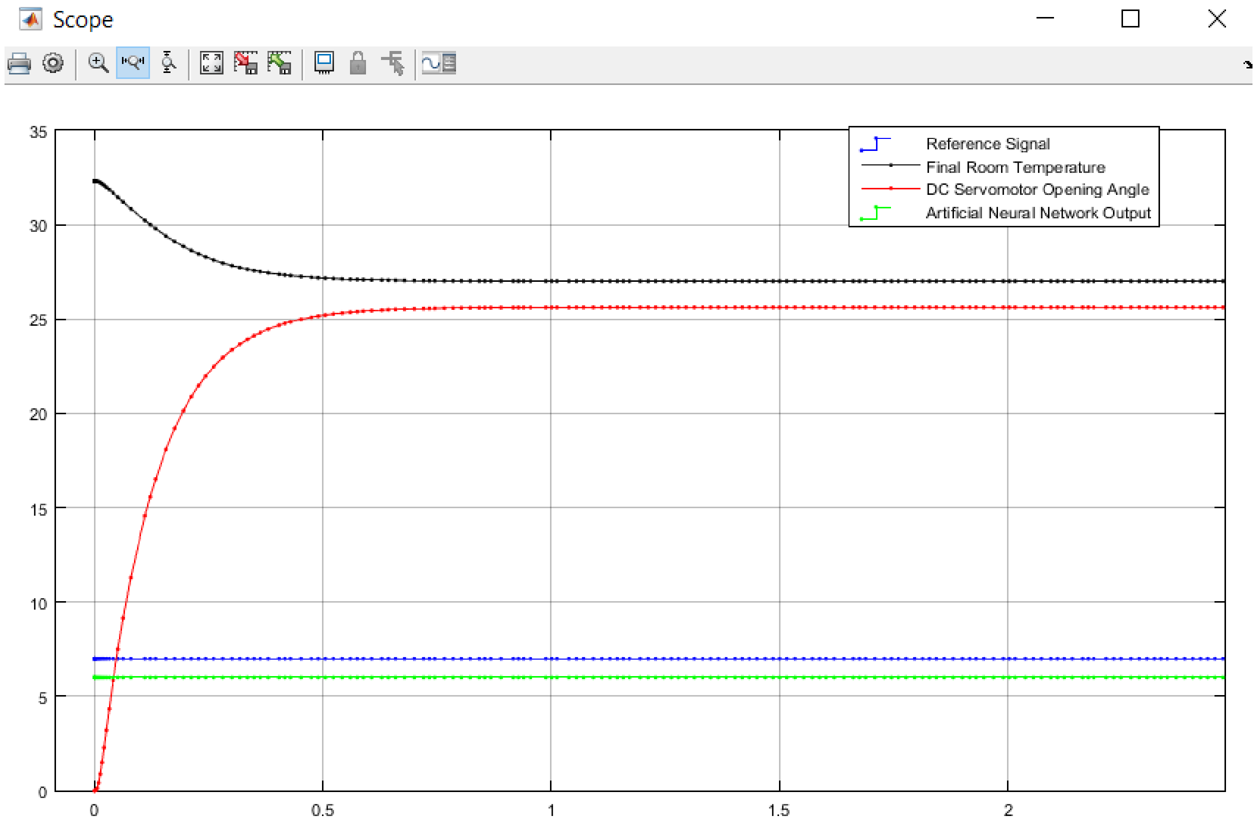


Fig.4.9 Simulink Output of the Cooling System for a Constant Input Signal

When the temperature of the surrounding air is at 34 ^{0}C, the ANN gives an output of 6volt to drive the DC servomotor. This output of the ANN is then amplified by the op-amp and is fed to the DC servomotor. The DC servomotor reaches steady state and rotates the butterfly valve with a settling time of around 0.6sec and with zero overshot at all. The final temperature of the surrounding air declines from 34 ^{0}C which is the initial temperature of the surrounding air and keep at a constant of 27 ^{0}C within 0.6sec of time period. The result show that the designed cooling system keeps the room temperature at the desired point ensuring the generator temperature to be below 50 ^{0}C.

CHAPTER FIVE

5. CONCLUSION, RECOMMENDATION and FUTURE WORK

5.1 Conclusion

One of the factors that determine the lifetime of a generator is temperature. As all losses generate heat it is important to have a cooling system that can maintain a sufficiently low operating temperature within all parts of the generator. In this thesis a non linear control of hydropower generator cooling system is designed. Considering the data obtained from one of the earliest hydropower stations in Ethiopia which is the Koka Hydropower station, a suitable and efficient cooling system is developed for the generators. In designing the cooling system a temperature based heat transfer analysis method is used. According to the heat transfer calculation, desired butterfly valve size and tubular heat exchanger length that guarantees the exchange of the heat from the air surrounding the generator in the closed generator room to the cooling water is appropriately selected. For these nonlinear characteristics of the cooling system an Artificial Neural Network is used as a controller that guides the flow of the required amount of the cooling water according to the load on the generator which in turn is expressed in terms of the heat dissipation of the generator and the temperature of the air surrounding it. The aim of this research was to design an efficient cooling system for hydropower generators particularly in terms of temperature to keep the generator below 50 ^{0}C. From the collected data from Koka Hydropower Plant it is seen that the generator will have a temperature of below 50 ^{0}C when the surrounding air has a temperature of around 27 ^{0}C - 28 ^{0}C. So the set point is chosen to be 27 ^{0}C. With this set point the overall system is designed.

The designed cooling system is simulated using MATLAB software. For the system three different types of input signals that represent the actual changes in temperature of the surrounding air are used. A sine wave, a ramp and a constant type input signals are used and this signals are generated from the temperature sensor which is placed in the generator room. From the simulink results is it seen that for the three types of inputs the designed cooling system keeps the surrounding air temperature at 27 ^{0}C when the temperature in the room is greater than 27 ^{0}C. The system responds to the changes in the temperature having a settling time of 0.6 seconds and without any overshot.

From the results it can be concluded that the designed cooling system for the hydropower generator particularly for the Koka hydropower generator works effectively and efficiently. Comparing the designed cooling system with the existing one which is cooling with the use of radiators using a simple on/off control mechanism to allow the flow of the cooling water, the designed ANN based cooling technique enables the control of the flow rate of the cooling water according to the load on the generator which makes it more reliable. Finally the results show that this heat transfer based design of hydropower generator cooling system is a reliable, efficient and effective type of cooling mechanism enhancing the life time and performance of the generator thus increasing the efficiency of the power plant at all.

5.2 Recommendation

The designed generator cooling system is based on the data gathered from Koka Hydropower Plant. So this design is mainly functional in efficient way for this hydropower plant. But the mechanism used to design the system can be used for any hydropower station. The use of this design of hydropower generator cooling system is recommended according to the data collection of the power plant in which the cooling system is going to be installed on. Based on the data this design can be used only by changing some of the parameters according to the data collected for any hydropower plant generator ensuring an efficient and effective cooling mechanism.

5.3 Suggestions for Future Work

The work done in this thesis could be extended by taking its hardware implementation. It is worth noting that the control approach proposed here can be effortlessly extended to any hydropower rating of a generator. Furthermore, the developed control strategy can be integrated with Fuzzy Logic Control and the resulting Adaptive Neuro-Fuzzy Control may further improve the performance of the proposed controller.

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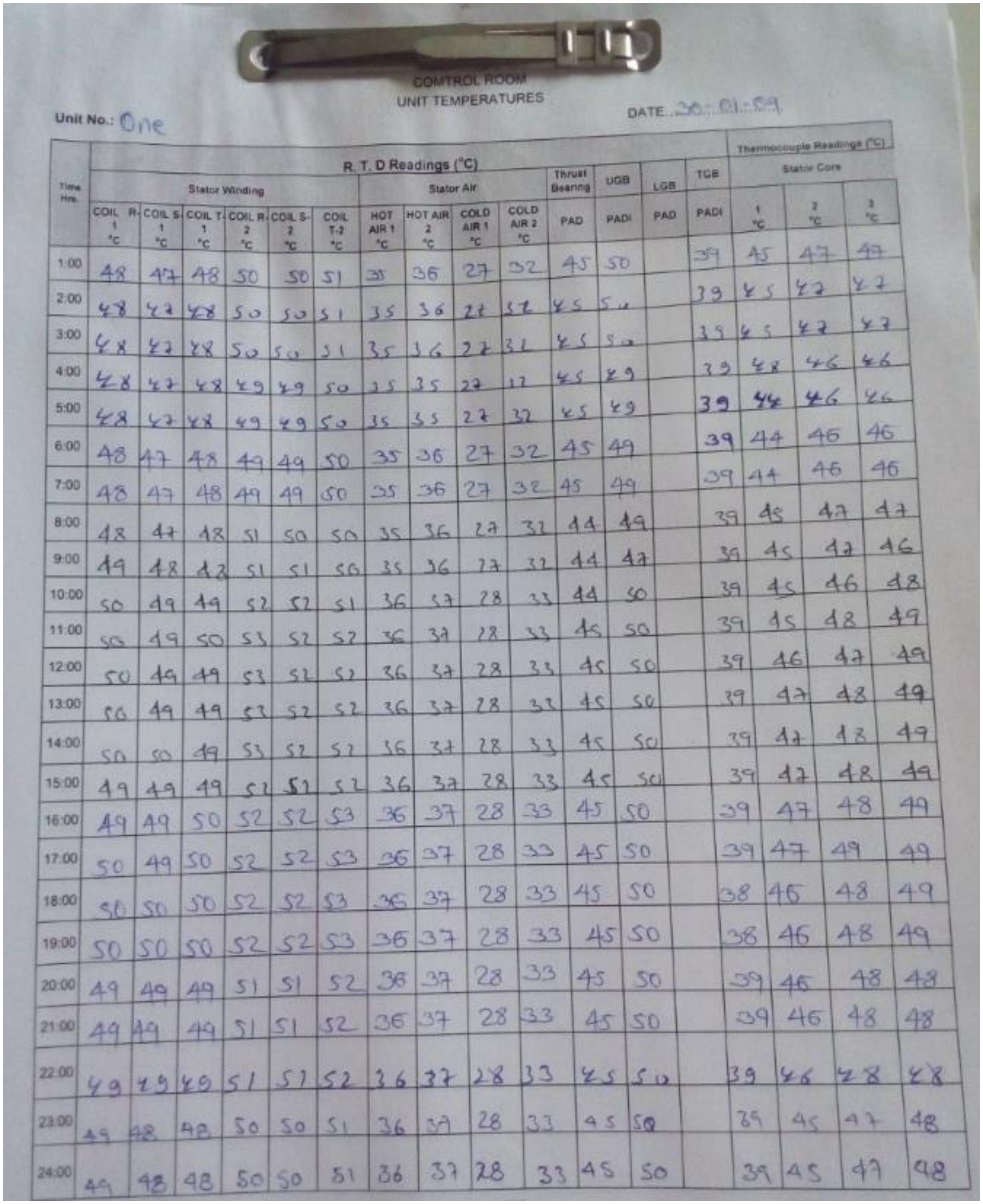
APPENDICES

Appendix A: KOKA HEPP Collected Data

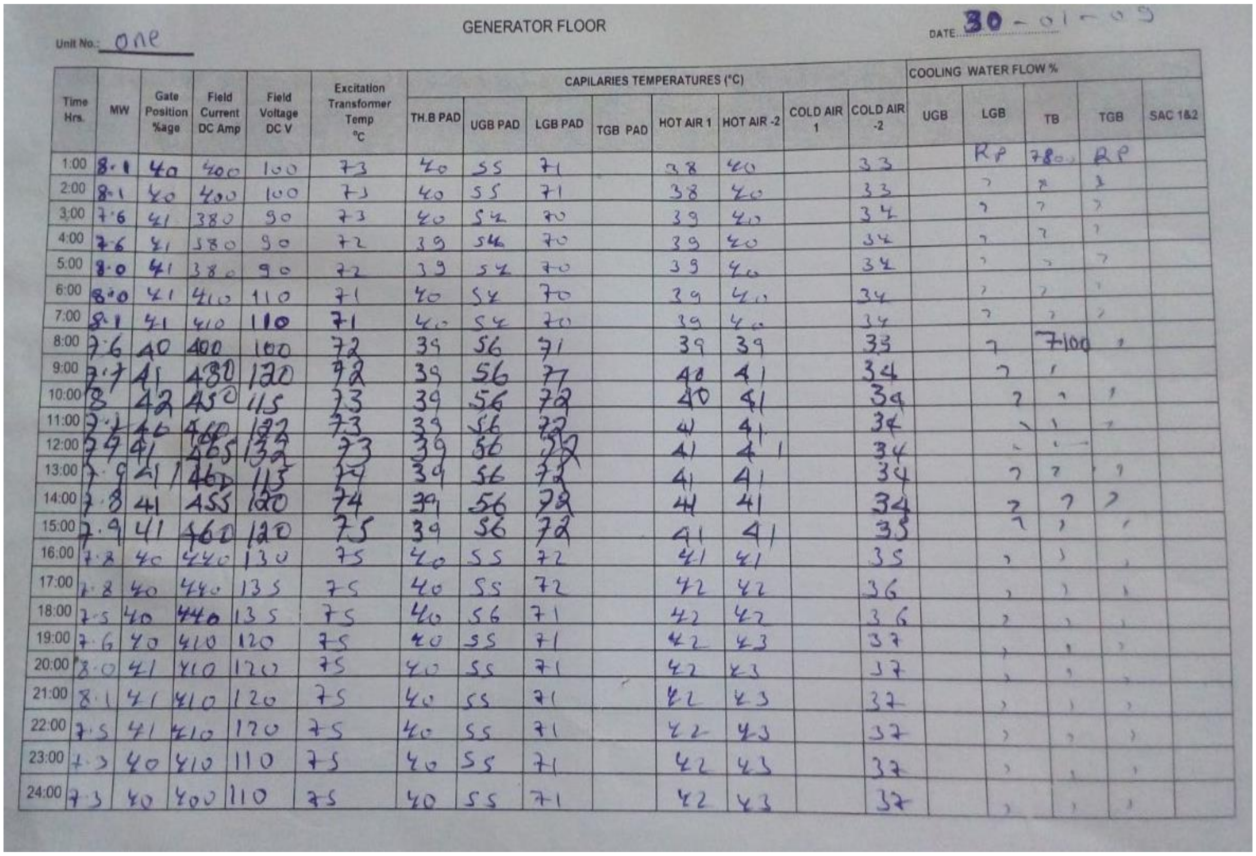


COMTROL ROOM
UNIT TEMPERATURES

Unit No.: One

DATE 30.01.09

Time Hrs.	R. T. D Readings (°C)														Thermocouple Readings (°C)		
	Stator Winding						Stator Air				Thrust Bearing	UGB	LGB	TGB	Stator Core		
	COIL R-1 °C	COIL S-1 °C	COIL T-1 °C	COIL R-2 °C	COIL S-2 °C	COIL T-2 °C	HOT AIR 1 °C	HOT AIR 2 °C	COLD AIR 1 °C	COLD AIR 2 °C	PAD	PADI	PAD	PADI	1 °C	2 °C	3 °C
1:00	48	47	48	50	50	51	35	36	27	32	45	50		39	45	47	47
2:00	48	47	48	50	50	51	35	36	27	32	45	50		39	45	47	47
3:00	48	47	48	50	50	51	35	36	27	32	45	50		39	45	47	47
4:00	48	47	48	49	49	50	35	35	27	32	45	49		39	48	46	46
5:00	48	47	48	49	49	50	35	35	27	32	45	49		39	44	46	46
6:00	48	47	48	49	49	50	35	36	27	32	45	49		39	44	46	46
7:00	48	47	48	49	49	50	35	36	27	32	45	49		39	44	46	46
8:00	48	47	48	51	50	50	35	36	27	32	44	49		39	45	47	47
9:00	49	48	47	51	51	50	35	36	27	32	44	47		39	45	47	46
10:00	50	49	49	52	52	51	36	37	28	33	44	50		39	45	46	48
11:00	50	49	50	53	52	52	36	37	28	33	45	50		39	45	48	49
12:00	50	49	49	53	52	52	36	37	28	33	45	50		39	46	47	49
13:00	50	49	49	53	52	52	36	37	28	33	45	50		39	47	48	49
14:00	50	50	49	53	52	52	36	37	28	33	45	50		39	47	48	49
15:00	49	49	49	52	52	52	36	37	28	33	45	50		39	47	48	49
16:00	49	49	50	52	52	53	36	37	28	33	45	50		39	47	48	49
17:00	50	49	50	52	52	53	36	37	28	33	45	50		39	47	49	49
18:00	50	50	50	52	52	53	36	37	28	33	45	50		38	46	48	49
19:00	50	50	50	52	52	53	36	37	28	33	45	50		38	46	48	49
20:00	49	49	49	51	51	52	36	37	28	33	45	50		39	46	48	48
21:00	49	49	49	51	51	52	36	37	28	33	45	50		39	46	48	48
22:00	49	49	49	51	51	52	36	37	28	33	45	50		39	46	48	48
23:00	49	48	48	50	50	51	36	37	28	33	45	50		39	45	47	48
24:00	49	48	48	50	50	51	36	37	28	33	45	50		39	45	47	48



Unit No.: one

GENERATOR FLOOR

DATE 30 - 01 - 09

Time Hrs.	MW	Gate Position %age	Field Current DC Amp	Field Voltage DC V	Excitation Transformer Temp °C	CAPILARIES TEMPERATURES (°C) TH.B PAD	UGB PAD	LGB PAD	TGB PAD	HOT AIR 1	HOT AIR -2	COLD AIR 1	COLD AIR -2	COOLING WATER FLOW % UGB	LGB	TB	TGB	SAC 1&2
1:00	8.1	40	400	100	73	40	55	71		38	40		33		RP	780	RP	
2:00	8.1	40	400	100	73	40	55	71		38	40		33		"	"	"	
3:00	7.6	41	380	90	73	40	54	70		39	40		34		"	"	"	
4:00	7.6	41	380	90	72	39	54	70		39	40		34		"	"	"	
5:00	8.0	41	380	90	72	39	54	70		39	40		34		"	"	"	
6:00	8.0	41	410	110	71	40	54	70		39	40		34		"	"	"	
7:00	8.1	41	410	110	71	40	54	70		39	40		34		"	"	"	
8:00	7.6	40	400	100	72	39	56	71		39	39		33		"	7100	"	
9:00	7.7	41	480	120	72	39	56	77		40	41		34		"	"		
10:00	8	42	450	115	73	39	56	72		40	41		34		"	"	"	
11:00	7.7	40	460	122	73	39	56	72		41	41		34		"	"	"	
12:00	7.9	41	465	132	73	39	56	72		41	41		34		"	"		
13:00	7.9	41	460	113	74	39	56	72		41	41		34		"	"	"	
14:00	7.8	41	455	120	74	39	56	72		41	41		34		"	"	"	
15:00	7.9	41	460	120	75	39	56	72		41	41		35		"	"	"	
16:00	7.8	40	440	130	75	40	55	72		41	41		35		"	"	"	
17:00	7.8	40	440	135	75	40	55	72		42	42		36		"	"	"	
18:00	7.5	40	440	135	75	40	56	71		42	42		36		"	"	"	
19:00	7.6	40	410	120	75	40	55	71		42	43		37		"	"	"	
20:00	8.0	41	410	120	75	40	55	71		42	43		37		"	"	"	
21:00	8.1	41	410	120	75	40	55	71		42	43		37		"	"	"	
22:00	7.5	41	410	120	75	40	55	71		42	43		37		"	"	"	
23:00	7.3	40	410	110	75	40	55	71		42	43		37		"	"	"	
24:00	7.3	40	400	110	75	40	55	71		42	43		37		"	"	"	

Fig. A-1 Records of Different Sensor Readings

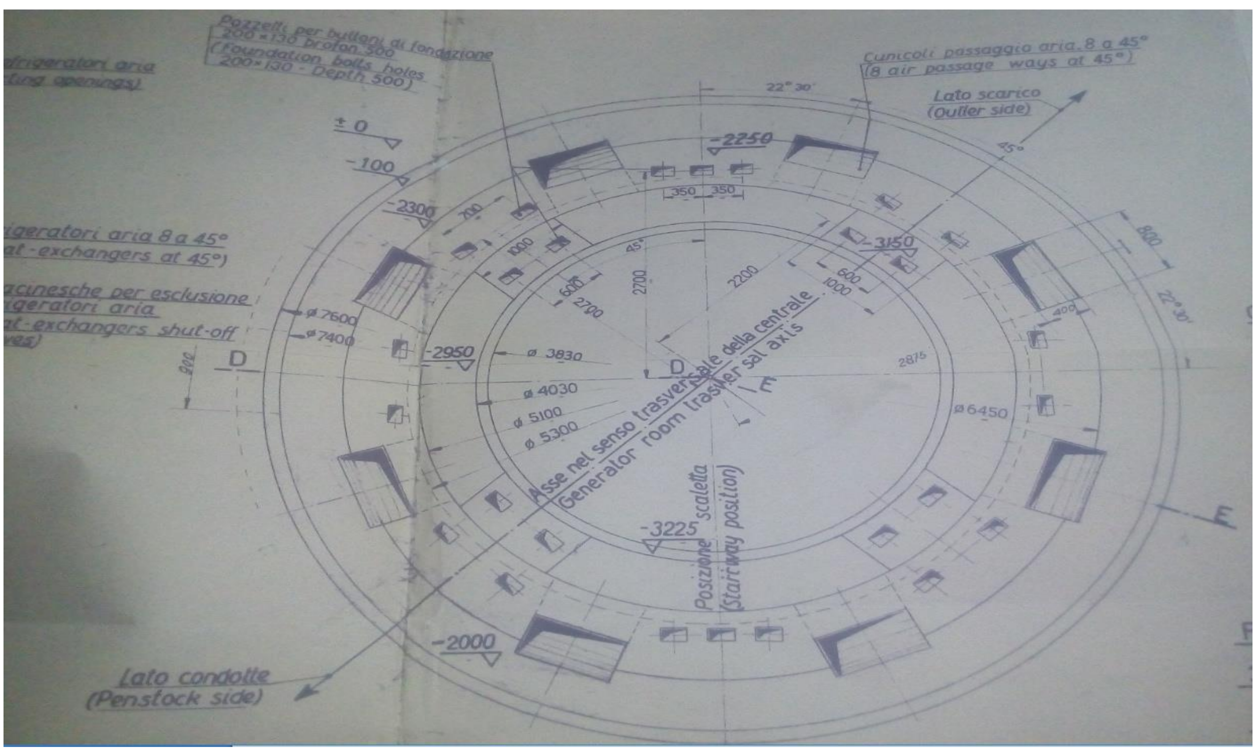


Fig. A-2 Mechanical Design of the Generator Room

Appendix B: Butterfly Valve Technical Data

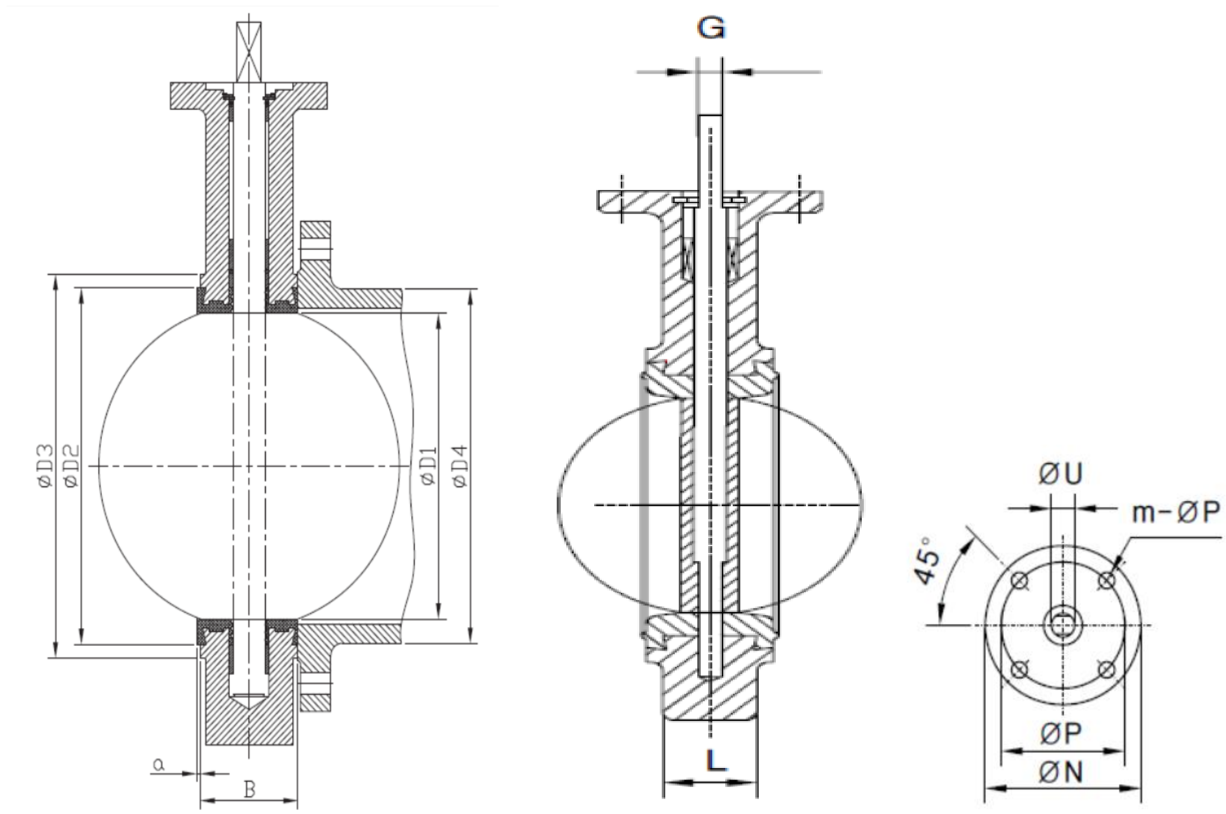


Fig. B-1 Mechanical Design of Butterfly Valves

Where D1: Chord Dimension of Protruding Part of Disc
D2: Outer Diameter of Seat
D3: Outer Diameter of Body
D4: Outer Diameter of Pipe
B: Face to Face Dimension of the Valve when Mounted
a: Projecting Dimension of Seat

Table B-1 Dimensional Information of Butterfly Valves

Size		D1	D2	D3	D4	B	A	Stem		øN	øP	m	L
Inch	mm							øU	G				
2	50	45	76.3	89	60.32	42	1.5	14	10	90	70	4	43
$2\frac{1}{2}$	65	57	89	108	73.02	44.7	1.5	14	10	90	70	4	46
3	80	73	103.9	120	88.90	45.2	2.0	14	10	90	70	4	46
4	100	96	135	150	114.30	52.1	1.3	16	12	90	70	4	52
5	125	116	159	181	141.30	54.4	1.75	19	15	90	70	4	56
6	150	147	188.4	208	168.28	55.8	1.4	19	15	90	70	4	56
8	200	194	238.2	260	219.08	60.0	1.4	22	18	90	70	4	60
10	250	242	292.4	320	273.05	65.6	2.2	28	20	125	102	4	68
12	300	292	344	375	323.85	76.9	1.5	28	20	125	102	4	78

*all dimesions are in mm.