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Temperature Control of Air Conditioning Compressor System on Electric Vehicles

Rizky Pangestu
Electrical Engineering, Universitas Indonesia
Kampus UI Depok 16424, West Java, Indonesia
rizky.pangestu@ui.ac.id

Bernadeta Wuri Harini
Electrical Engineering, Universitas Indonesia
Kampus UI Depok 16424, West Java, Indonesia
wuribernard@usd.ac.id

Dr. Ir. Feri Yusivar, M.Eng
Electrical Engineering, Universitas Indonesia
Kampus UI Depok 16424, West Java, Indonesia
yusivar@yahoo.com

The compressor on the electric bus is used on the part of the bus cooling system where the compressor is coupled with an induction motor. The room temperature on an electric bus depends on how we control the speed of the electric motor to rotate the impeller blade contained in the compressor so that the refrigerant can be channeled through the condenser and lower the room temperature. Therefore, an inverter is needed as a converter of electrical power from a 400 V DC battery source to 3 phase AC electricity, which in turn will be varied in frequency based on the Proportional Integral Differential (PID) controller to control the induction motor speed using vector control. IGBT switches contained in the inverter receive input signals in the form of ON and OFF pulses generated through the Sinusoidal Pulse Width Modulation (SPWM) method.

Keywords—induction motor, inverter, SPWM, PID, compressor, temperature

I. INTRODUCTION

Along with the development of technology in the automotive world, the need for fuel oil needed for conventional vehicles is increasing. But along with the rising need for fuel, oil prices also continue to rise. In addition, air pollution generated from oil-fueled motor vehicles is also increasing. Therefore, the automotive industry has developed many types of hybrid cars and electric cars. Electric vehicle technology is a new innovation that has the same function as conventional fueled vehicle technology, but electric vehicles have some differences with conventional buses.

Some differences between electric buses and conventional buses are found in the voltage source driving the electric bus and the cooling system. On electric buses, the driving source of the bus comes from the DC battery. To be able to drive the 3 phase induction motor contained in this cooling system, an inverter is needed to function to convert the DC voltage into a 3 phase voltage. After that, the voltage can be used to drive a 3 phase induction motor which is coupled to the compressor. Then, in a conventional bus cooling system, the compressor in the cooling system is coupled directly to the car engine. Whereas in the electric bus cooling system, the compressor system is coupled with an induction motor to regulate the rotation speed of the impeller blade inside to let out the cooling gas. The output temperature of the electric bus cooling system will be affected by how fast the impeller blade rotates. So we need an induction motor speed control system that is coupled with a compressor to control the temperature of the cooling gas on the electric bus. Induction motor speed control method used in this study is a rotor field oriented vector which is also called the Rotor Field Oriented Control (RFOC). The author uses this method because this method is the most frequently used method and has been widely researched by available scientific journal sources. The induction motor speed controller and the coolant gas temperature controller both use IP (Integral-Proportional) controllers to control their respective setpoints.

In this simulation, the author will simulate an inverter drive that uses 6 IGBT switches used to drive an induction motor coupled with a air conditioning compressor system on an electric bus.

II. COMPRESSOR SYSTEM DEFINITION AND MODELING

There are 2 main components of the compressor system on the bus. The first one is the compressor itself and the second one is the heat exchanger which works as a condenser and evaporator on a refrigeration system. The following will be discussed about the modeling of each of these components.

A. Compressor System

The compressor is a mechanical device used to compress gas or air to get to higher pressure and use the compressed air to the next system. Compressors can be used in transportation equipment such as cars where the function is to suck refrigerants (cooling gas or freon) through low pressure pipes and drain them into the air condenser through pipes. The temperature and pressure of the refrigerant will be compressed and will be passed through the air condenser.

In this study, the author uses a centrifugal compressor that is modeled based on the physical design parameters of the compressor to get the correct behavior from the compressor system (Gravdahl and Egeland, 1999).

Centrifugal compressor works by accelerating the fluid entering the compressor into high speed then decreasing its speed to convert kinetic energy in the gas to potential energy. The fluid that enters the compressor has a large flow called mass flow in units of kg/s. The author will model the compressor by looking at its characteristic curves and using the appropriate mathematical model.

1) Compressor Surge

Each compressor has its own characteristic curves as shown below.

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by Harini Bernadeta Wuri

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Rizky Pangestu
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Universitas Indonesia
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rizky.pangestu@ui.ac.id

Bernadeta Wuri Harini
Electrical Engineering,
Universitas Indonesia
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wuribernard@usd.ac.id

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1) Compressor Surge

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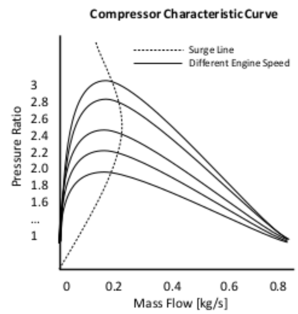

Fig. 1. Compressor Characteristic Curve

The image shows the surge line on a compressor for variations in the rotational speed of the compressor. If the mass flow of a compressor drops below (on the left) of the surge line, then the compressor will enter the area where the compressor's work will be unstable and various variations of pressure and mass flow can occur (e.g. reverse mass flow and oscillation from the output pressure). This will cause the compressor to become damaged and can affect other equipment.

In fact, a compressor cannot be modeled to get a measurement when the compressor works to the left of the compressor characteristic curve (left of the surge point) due to the things described earlier. So that the compressor modeling at this writing, the author can only model the right side of the surge point because the compressor can only work in that section.

2) Mathematical Model of Centrifugal Compressor

The modeling of the compressor which will be explained in this paper is modeling based on the physical shape of the centrifugal compressor.

The average of the diameter D_1 of the inducer is defined by:

$$D_1^2 = \frac{1}{2}(D_{l1}^2 + D_{h1}^2) \quad (1)$$

Where D_{l1} is the diameter at the end of the inducer and D_{h1} is the diameter at the hub of the chassis. When the compressor works, there will be a slip between the gas and the impeller in the compressor. Ideally, the gas speed is the same as the impeller speed. The slip factor is defined as:

$$\sigma = \frac{C_{\theta 2}}{U_2} \quad (2)$$

Where σ is the slip factor, $C_{\theta 2}$ is the gas velocity at the impeller end and U_2 is the velocity at the end of the impeller. However, because usually the $C_{\theta 2}$ value is unknown, another approximation is used that the slip value always remains at a certain value (Gravdahl and Egeland, 1999). If the slip factor is constant, the enthalpy without loss given to the gas can be defined as:

$$\Delta h_{0c,ideal} = \sigma U_2^2 \quad (3)$$

According to (Watson and Janota, 1982), (Ferguson, 1963) and (Nisenfeld, 1982) there are 2 considerable efficiency losses on centrifugal compressors, namely incident loss and fluid friction loss. Both losses occur in the impeller and diffuser and can be defined as:

$$\Delta h_{ii} = \frac{1}{2}\left(U_1 - \frac{\cot(\beta_{1b})\, m}{\rho_{01} A_i}\right)^2 \quad (4)$$

$$\Delta h_{id} = \frac{1}{2}\left(\frac{\sigma D_2 U_1}{D_1} - \frac{m(\alpha_{2b})}{\rho_{01} A_d}\right)^2 \quad (5)$$

$$\Delta h_{fi} = k_{fi} m^2 \quad (6)$$

$$\Delta h_{fd} = k_{fd} m^2 \quad (7)$$

$$\alpha_{2b} = \arctan\left(\frac{D_1 \tan(\beta_{1b})}{\sigma D_2}\right) \quad (8)$$

With :

Δh_{ii} = incident loss at the impeller
Δh_{id} = incident loss on diffuser
Δh_{fi} = loss of fluid friction at the impeller
Δh_{fd} = loss of fluid friction on the diffuser

By combining the above equations, we get 1 isentropic efficiency equation from a compressor, namely:

$$\eta_i(m, U_1) = \left(\frac{\Delta h_{0c,ideal}}{\Delta h_{0c,ideal} + \Delta h_{id} + \Delta h_{id} + \Delta h_{id} + \Delta h_{id}}\right) - \Delta n_{bf} - \Delta n_v \quad (9)$$

Where Δn_{bf} and Δn_v are other losses which have a small enough value which can reduce efficiency.

After obtaining the efficiency equation, using the isentropic compression equation, the equation can be changed to the ratio between the gas pressure released by the compressor, namely:

$$p_2 = p_{01}\left(1 + \frac{\Delta h_{0c,ideal} \times \eta_i(m, U_1)}{T_{01} C_p}\right)^{\frac{k}{k-1}} \quad (10)$$

And the compressor torque τ_c can be formulated as :

$$\tau_c = \frac{|m| D_2 \sigma U_2}{2} \quad (11)$$

B. Heat Exchanger (Condenser and Evaporator)

Heat exchanger model that used in this paper will be working as evaporator and condenser in the compressor system. After the pressure is increased by the compressor, the refrigerant gas will enter the HE system and has its temperature reduced so it can be used to cool the bus room. The evaporator will change its form from gas to fog that will be transported into the room through the air conditioner.

The flow inside the heat exchanger pipes or channels is considered to be one-dimensional, and discretized in as many elements as required. The basis of the numerical procedure is to decouple the calculation of the fluid flows from each other.

Then, both fluid flows along the heat exchanger are calculated through the integration of the one-dimensional conservation equations. For the refrigerant, the two fluid (liquid & vapour) separated model under equilibrium is considered.

Specific correlations for evaporation and condensation heat transfer in pipes and plate heat exchangers have been implemented into the model.

$$\frac{dp}{dz} = -\frac{d(\rho u^2)}{dz} - f\frac{1}{2}\rho\frac{u^2}{D_h} - \frac{d(zg\rho)}{dz} \quad (12)$$

$$A.G.\frac{d\left(i + \frac{u^2}{2}\right)}{dz} = Ph(T_w - T) \quad (13)$$

(8) Where the fluid exchanges heat with the surrounding walls at T_w (pipe wall). These equations are the governing equations for the water flow in the HE and the heated vapour or subcooled liquid.

The continuity equation states the conservation of the mass flow rate and the mass velocity all along a fluid path. Its value is known from the inlet conditions, so that G (mass velocity) will be considered as a known constant in the following analysis. Therefore only the rest of the equations must be integrated.

III. VECTOR CONTROL AND TEMPERATURE CONTROL

This temperature control system that will be modeled consists of two sub-systems which are the vector control system and temperature control system. The author use several system as the step of making the whole model. The first one is the system used to confirm the compressor model is right. This first system is an open loop system consists of a motor that given constant voltage and the compressor as can be seen in Fig 2. After that, the vector control is applied with compressor as the load as can be seen in Fig 3. Here on this system, PI and IP controller is already used. Which PI controller is used for the FOC and IP controller for the speed control. The IP controller is used here because it is easier to tune the I and P constant. The classic closed loop with feedback control system is applied here. Then as the final model, the temperature control is applied to the previous model that can be seen in Fig 4. Author adds one more outer loop and one more IP controller to the previous system that can be seen on the figure. Hence, the author uses double control loop on this simulation.

So in shot, to simulate this system, we are using double control loop. One is for the speed control and second is for the temperature control. After the speed control loop of the induction motor has done, the gas temperature of the condenser is used as the feedback for calculating error of the temperature that will be controlled by IP controller as the speed reference.

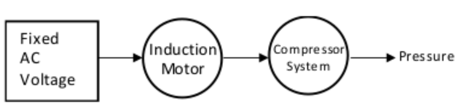

Fig. 2. Open Loop System for Compressor Testing

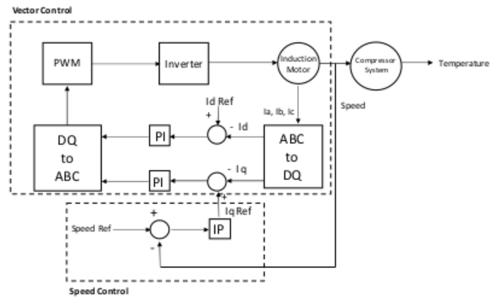

Fig. 3. Closed Loop System (without temperature control)

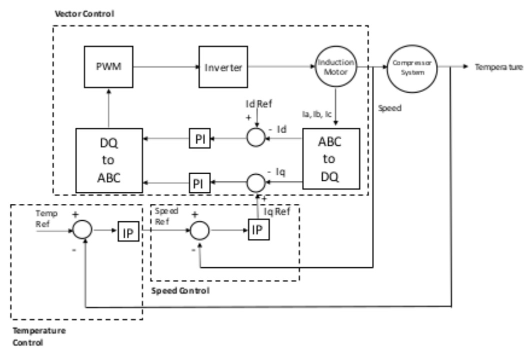

Fig. 4. Closed Loop System (with temperature control)

To be able to control the speed and torque of an induction motor, the frequency and voltage supplied to the motor must be controlled. The method is usually called Variable Frequency Drive (VFD) or vector control.

Vector control method controls motor torque directly by observing motor current while moving and controlling the magnitude of instantaneous magetizing current (id and iq) to remain constant and flux on the rotor is also constant. Such controls are called vector controls because their control is based on vector representations of currents and magnetic fluxes of the motor. There are 2 types of vector controls, namely DTC (Direct Torque Control) and FOC (Field Oriented Control). The method that the author used in this simulation is the FOC method.

FOC is a method of contolling the AC motor field by changing the system coupled to a decoupled system. This is useful for controlling the system of current entering the motor separately, so that the flux and torque can also be controlled separately. The motor torque produces a current iq calculated from the ratio between the reference speed and the actual speed of the motor.

In the induction motor, the stator current is divided into two rotating axes namely id and iq. The stator iq current is controlled to produce the desired torque, while the stator id current has an effect on the rotor flux density value. The equation of the stator current id and iq can be written with the formula as below.

$$\begin{bmatrix} id \\ iq \end{bmatrix} = \frac{2}{3}\begin{bmatrix} sin\theta & \sin(\theta-\frac{2\pi}{3}) & \sin(\theta+\frac{2\pi}{3}) \\ cos\theta & \cos(\theta-\frac{2\pi}{3}) & \cos(\theta+\frac{2\pi}{3}) \end{bmatrix}\begin{bmatrix} ia \\ ib \\ ic \end{bmatrix} \quad (14)$$

Where the value of θ is obtained through the integral speed that can be written in the formula

$$\theta = \int_0^t \omega\, dt = \int_0^t \omega r\, dt + \int_0^t \omega o\, dt \quad (15)$$

The ωo value is obtained from the speed of the induction motor, while the value of ωr can be calculated using a formula:

$$\omega r = \frac{2Lr}{P\tau}\frac{Te}{\lambda} \quad (16)$$

And because the formula of *id, iq,* and τ :

$$id = \frac{\lambda}{Lm} \quad (17)$$

$$iq = \frac{2Lr}{PLm}\frac{Te}{\lambda} \quad (18)$$

$$\tau = \frac{Lm + Lr}{Rr} \quad (19)$$

Then :

$$\omega r = \frac{iq}{id}\frac{1}{\tau} \quad (20)$$

Table 1. Motor Specifications

MOTOR SPECIFICATION	
Power	4kW
Rated Frequency	50 Hz
Voltage	400 Volt
R_S	1.405 Ω
$L_{\sigma S}$	0.005839 H
L_M	0.1722 H
R_R	1.395 Ω
σ	0.0645
P	2 pair

IV. IMPLEMENTATION

This simulation is divided into three systems based on its components that consists of : Inverter, Induction Motor (Vector Control) and its load (Compressor System). We can see the whole simulation on Fig. 5.

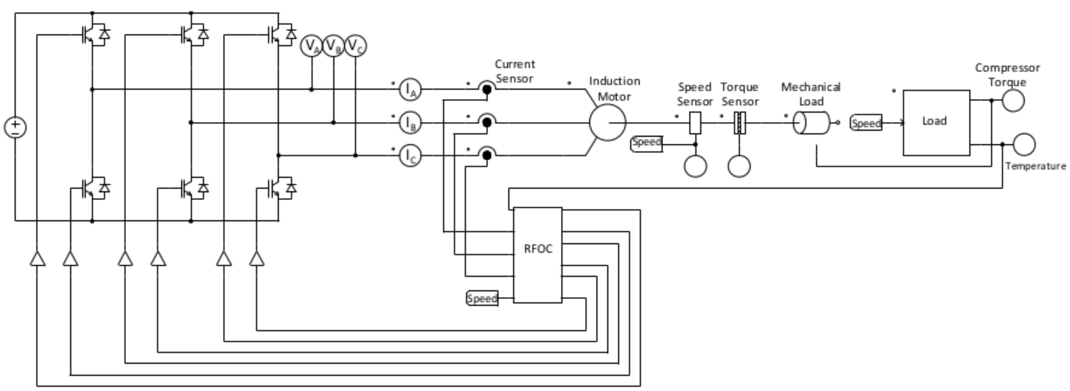

Fig. 5. System Simulation

It has been shown in the preceding sections that the rotor flux and torque produced by the machine can be independently controlled through the d and q axis components of the stator current respectively. However, most of electronic power converters used in electric drives operate as a voltage source (i.e. they apply a voltage to the machine), the resulting current being a function of the machine parameters and of its operating point. It is possible to express the stator voltage as a function of the stator currents and the rotor flux in a rotor flux synchronous reference frame as :

$$V_{ds} = R_s' i_{ds} + \sigma L_s \frac{di_{ds}}{dt} - \omega_s \sigma L_s i_{qs} - R_r \frac{L_m}{L_r^2}\psi_{dr} \quad (21)$$

$$V_{qs} = R_s' i_{qs} + \sigma L_s \frac{di_{qs}}{dt} - \omega_s \sigma L_s i_{ds} - \omega_m \frac{L_m}{L_r}\psi_{dr} \quad (22)$$

This simulation use the Squirrel Cage Induction Motor type provided by simulation application. The specification used for the induction motor is provided in this table :

Starting from the left, there is an inverter with a 400 V DC voltage source. This inverter consists of 6 IGBTs that received an ON/OFF signal from the PWM control on the RFOC block diagram (located below the Induction Motor in Fig. 5. By default, this simulation has a sampling time of 0.001 ms but because this simulation will be implemented on a real microcontroller, the sampling time used has to be readjusted to 1 ms for the speed control and 0.1 ms for the vector control.

As for the PWM, the author use a SPWM method with a isosceles triangle as a carrier. There were two isosceles trangle used with a different amplitude of 0.01 that is allocated as a dead time between ON and OFF condition ofeach IGBT switch. This dead time is needed because in real-life condition, IGBT will have time to switch from its ON to OFF condition which is not neglectable by any means. If there is no dead time used, there will be a time where the upper and lower switch are both ON and will cause the circuit to shorted.

Then the induction motor is coupled with a compressor system with torque as its feedback to the motor. This ensures that the compressor is giving torque as long as the motor is running. Speed sensor and torque sensor on the motor is used as the output from the motor. From the compressor load, there is a temperature output that will be used as a feedback to the vector control.

Since the simulation support C-Blocks feature, the previous explained model of compressor and vector control can be written as an equations by using C language on the simulation application. There is no need to declare which C-Blocks model is discrete or continue because the equations itself will be automatically calculated as discrete if we already put a sampling time (delta t) in it, or will be calculated as continue if there is no sampling time in it.

V. RESULT AND ANALYSIS

By using the mathematical model in the previous section, it can be seen that the compressor output pressure in the simulation is already having the same characteristic with the compressor characteristic curve. Which is the output pressure will decrease as the mass flow increases.

The author only use the model stated in Fig 2. to get the result at the Fig 6. Input pressure is set as a constant of 9800 Pascal. One speed variation is used so there is only one curve there. The x axis is the mass flow that applied to the compressor. The y axis is set as the pressure output and not pressure ratio. But, as stated before, this model can only give the right side of the surge line of the compressor which is the real working range of the compressor output pressure based on its mass flow. With the amount of mass flow less than 2 kg/s, this compressor can increase the pressure by 400 Pa at its most.

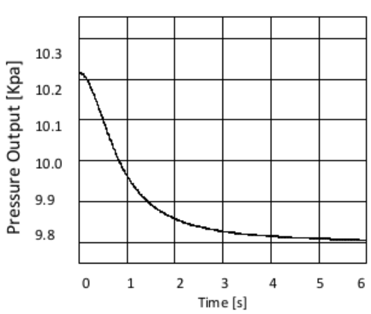

Fig. 6. Compressor Characteristic Curve

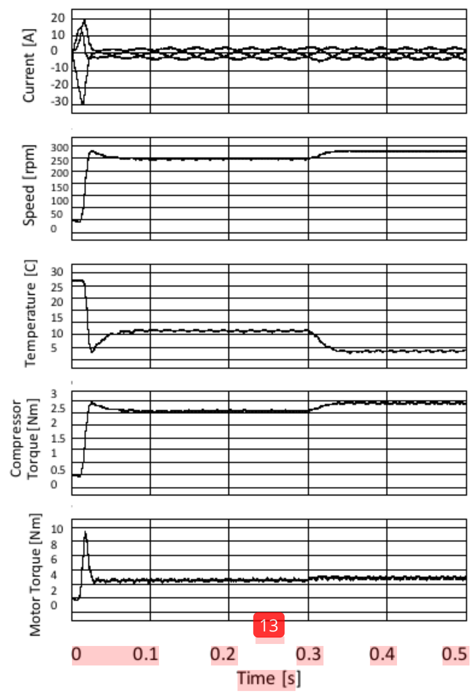

Fig. 7. Current, Speed, Temperature, Compressor Torque and Motor Torque Graph

From Fig. 7. we can see 5 output results from the simulation. These are the result from the model on Fig. 3 The first one is the rotor current of the motor. From the figure there can be seen that a three phase current with an amplitude of 10 A at the beginning of the simulation (this happened because the motor needs a high starting current) and starting to enter the steady state by the amount of 5 A.It means that the inverter already working as it should be. Then the speed response of the motor is also shown. The author use the first speed setpoint at 250 rpm at 0 s and second speed setpoint at 280 rpm at 0.3 s. Two setpoints were used to test the speed control is already working even with second setpoint after the first setpoint have already reached the steady state. There is a little oscillation there caused by the motor working at a low rating speed. This oscillation is a result of PWM switching, which is always ON and OFF everytime so it causes some up and down change in a short of time on the speed graph (this can be more clearly seen at Fig. 8.). Then we can compare it with the temperature graph. As we can see, the temperature is going down as the speed increases and vice versa. Then it means that the compressor model already works as the physical model of compressor. But, the temperature range is too narrow that with 30 rpm speed decrease there is around -10^{o} Celcius differences. This is caused by the parameters of the compressor not being calculated or provided correctly by the literature so several parameter might not be the correct one. Both torque of motor and compressor can be seen too with an amount around 3-5 Nm. This torque produced by the compressor acts as a quite small loads for the motor because the motor torques oscillates but it still maintains the reference speed with the load attached.

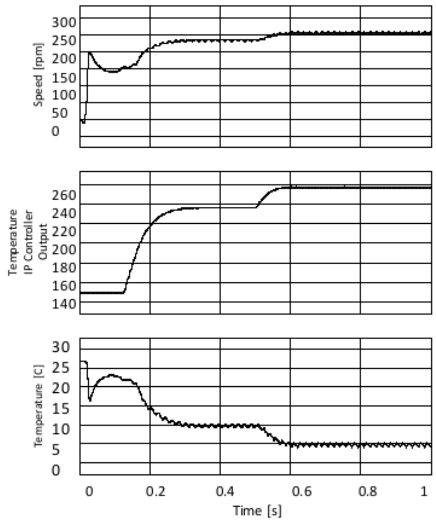

Fig. 8. Speed, Temperature IP Controller Output, Temperature

After the author confirms the speed control already works by using model on Fig. 3, the model on Fig. 4 is implemented. The compressor can work between the range of 150 rpm until 600 rpm. The model is given the setpoint of 10° Celcius at 0 s and 5° Celcius at 0.5 s. As we can see, the system is starting to move to the appropriate speed for the given temperature at 0.2 s and 0.6 s. There are a few oscillations in the steady-state with the amount of ± 0.5° Celcius.

VI. CONCLUSION

In this paper, the model of temperature control of air conditioner compressor system on electric vehicle is proposed, and the result implemented already shown with the result such as the motor can be controlled with a speed range of 200 rpm - 600 rpm and the temperature can already be controlled by using double loop IP controller. The temperature control needs to have slower response than the speed control because it acts as the outer loop in this system.

ACKNOWLEDGMENT

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