

Modeling and Simulation of a Bacteriological Incubator

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Bacteriological incubator consists of equipment that provides an optimized environment for the development of bacteria, frequently used for the development and testing of antibiotics and antiseptics. This work aims to perform white box modeling and simulation of a bacteriological incubator with an operating range of up to 50 °C, presenting a comparison of the P (Proportional), PI (Proportional-Integral) and PD (Proportional-Derivative) compensators in the process loop, enunciating the advantages and disadvantages. Furthermore, the simulation was developed in the MATLAB/Simulink environment, using as reference the commercial bacteriological incubator BF115, manufactured by BINDER. As a result, for the heating process, the PI compensator meets the design requirements and it was verified in computer simulation, while the others, even for cooling, do not meet these requirements and they are not implementable.

Keywords: Bacteriological Incubator. Modelling. Compensation. Simulink. Temperature.

Bacteriological incubators consist of equipment that provides an optimized environment for the development of bacteria, through the monitoring and compensation of thermal variables, such as temperature and humidity [1]. This equipment is often at microbiology laboratories for cultivation of pathogenic bacteria, antibiotics research, clinical diagnosis, and antiseptic efficiency testing.

Typically, the air in the bacteriological incubator has a reference temperature of 37 °C (degrees Celsius), which is the average temperature of human body, so it reaches the carbon dioxide levels necessary to promote maintenance of cell growth. For the system design of a bacteriological incubator, the microorganisms require a particular set of parameters, such as temperature and humidity, for their growth and development [2].

As work references, Abo Al-kibash and Dana [2] made a similar proposal but focusing on prototyping and experimental results. On the other hand, Pugachuk, Burkhanova and Fominykh [3] contributed to expressing the temperature variation inside one bacteriological incubator. Zermani and Mami [4] proposed a theoretical modeling for

incubator systems, but for another application scenario, to reduce heat loss by evaporation.

Thus, based on literature and similar works' research, if the process' variables have not been monitored and compensated correctly, then this equipment as a product will not guarantee an optimized environment for bacterial cultures. In this context, the paper aims to develop a mathematical model of a bacteriological incubator system, to understand the process dynamic behavior for monitoring and compensating the average internal temperature.

This document has been segmented into the following content: section 2 – methodology – about the system plant, actuator and compensators' modeling, section 3 – results – about the simulation of the process, section 4 – conclusion – and references.

Materials and Methods

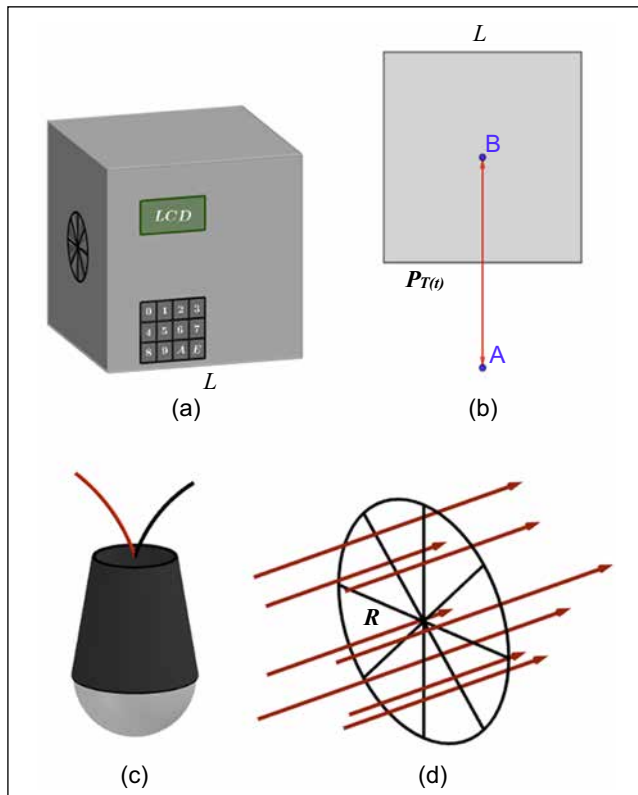
To model the process of a bacteriological incubator system, this work assumes a Proof of Concept (PoC) with one temperature sensor, two temperature actuators – one for heating and another for cooling – and cubic geometry for the plant – with side L – to promote the same geometry on front, side and superior view (Figure 1).

The graphical representation adopted to model the bacteriological incubator system is shown in Figure 2. The system input-output relationship is the

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Figure 1. PoC bacteriological incubator system.

user's desired temperature and the average internal temperature measured.

In this system, it absorbs thermal energy and dissipates it to the environment using actuators that amplify or attenuate the average internal temperature, operating one at a time through a switching system.

PoC Creation Steps

The steps used to develop the PoC is shown in Figure 3.

Thermal Process Modeling

For simplicity, the PoC is considered a linear time-invariant (LTI) system, and the system identification method based on white box model is applied to obtain the process transfer function using the Laplace transform [5].

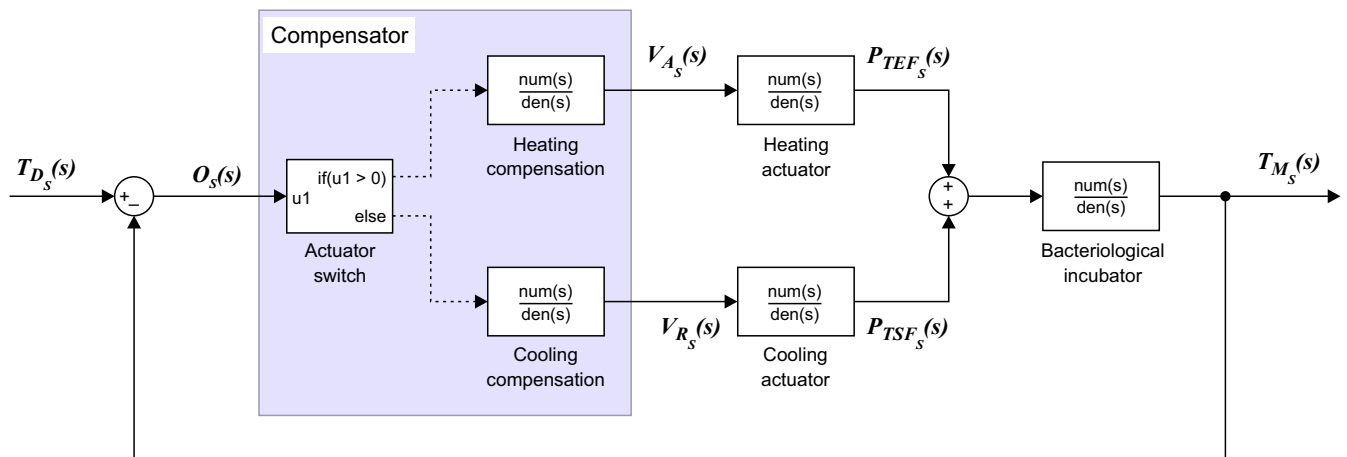
Typically, the thermal processes exhibit one thermal resistance (R) and one thermal capacitance (C), being analogous to modeling an RC series electrical circuit [6]. This type of modeling helps extract system performance specifications for control, such as the time constant.

The principle of calorimetry indicates the law of conservation energy in thermal systems, i.e., the variation in thermal energy from point A to point B, ΔE_{TAB} , is directly proportional to the variation in temperature at point B occasioned by point A, ΔT_B , given by Equation (1).

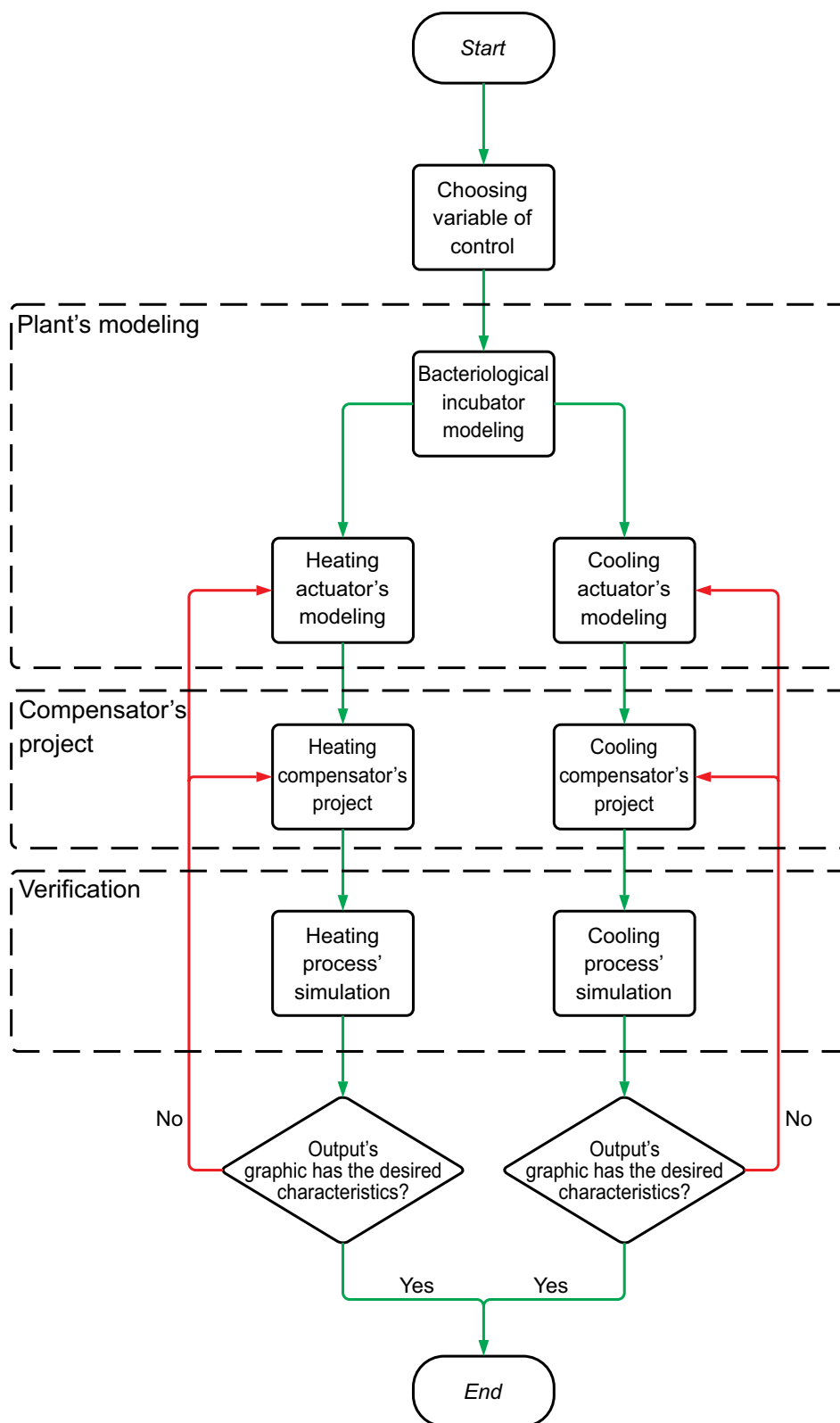
$$\Delta E_{TAB} = m_B c_B \Delta T_B \quad (1)$$

where m_B is equal to the mass at point B and c_B the specific heat also at point B.

To express the thermal interaction of the bacteriological incubator with the actuators and the

Figure 2. Graphical representation of the bacteriological incubator system.

Source. Own authorship

Figure 3. PoC steps' diagram.

Source. Own authorship

environment, point A is not contained within it and point B is, which expresses the thermal absorption and dissipation by PoC, as shown in Figure 1b. Besides, taking unit time steps, the thermal power is equal to expression 1 and considering the PoC has a constant volume and operation environment, which means m_{inc} (interior mass of the bacteriological incubator) and c_{inc} (interior specific heat of the bacteriological incubator) constant, we arrive at Equation (2).

$$P_T(t) = \frac{dE_T(t)}{dt} = m_{inc}c_{inc} \frac{dT_M(t)}{dt} \quad (2)$$

The average temperature $T_M(t)$ inside the bacteriological incubator, using the arithmetic mean, is equal to the sum of the temperatures of the n point objects present inside it, $T_n(t)$, divided by their quantity (which comprises the volume V of the PoC itself), as described in Equation (3).

$$T_M(t) = \frac{\sum_{n=1}^k T_n(t)}{V} \quad (3)$$

Since the inside of the bacteriological incubator consists more of air than something else, the mass inside it is calculated as a function of air density, d_{ar} . Thus, the thermal capacitance C_T of the PoC is given by Equation (4).

$$P_T(t) = L^3 d_{ar} c_{ar} \frac{dT_M(t)}{dt}$$

$$P_T(t) = C_T \frac{dT_M(t)}{dt} \quad (4)$$

The thermal power that interacts with the bacteriological incubator, $P_T(t)$, is the sum of the thermal powers from the environment and the actuators, named natural thermal power, $P_{TN}(t)$, and forced thermal power, $P_{TF}(t)$, respectively, given by Equation (5).

$$P_{TN}(t) + P_{TF}(t) = C_T \frac{dT_M(t)}{dt} \quad (5)$$

Natural thermal power is directly proportional to the variation between the average temperature inside the bacteriological incubator and the environment temperature. Considering the heat

transfer process by thermal convection occurs in the bacteriological incubator at each unit time interval, this calculates according to Equation (6).

$$P_{TN}(t) = h_{ar} A (T_A(t) - T_M(t)) \quad (6)$$

where: $T_A(t)$ is the environment temperature and $R_T = \frac{1}{h_{ar}A}$ the thermal resistance, given by the inverse of the product between the area A , perpendicular to the plane in which the thermal energy transfer occurs, and the thermal convection coefficient h_{ar} .

Substituting Equation (6) into equation (5), comes the ordinary differential equation (ODE) that expresses the thermal dynamics of the PoC, as described in Equation (7).

$$\frac{dT_M(t)}{dt} + \frac{T_M(t)}{R_T C_T} = \frac{T_A(t)}{R_T C_T} + \frac{P_{TF}(t)}{C_T} \quad (7)$$

In this work, the bacteriological incubator operates in an environment where the temperature is constant. Applying the Laplace transform, the transfer function of the process results in Equation (8).

$$\frac{T_{Ms}(s)}{P_{TFs}(s)} = \frac{1}{C_T} \cdot \frac{1}{s + \frac{1}{R_T C_T}} \quad (8)$$

Heating Actuator Modeling

The actuator for heating the interior of the bacteriological incubator used in this work is an incandescent lamp, for illustrative purposes. The effective thermal power, $P_{TEF}(t)$, is a percentage of the electrical power present in the incandescent lamp, $P_S(t)$, associated with the useful energy conversion process, according to Equation (9).

$$P_{TEF}(t) = \eta P_S(t) = \frac{\eta}{R_A} V_A^2(t) \quad (9)$$

where $V_A(t)$ is the voltage source and R_A the electrical resistance of the incandescent lamp. In addition, η expresses the conversion factor, with $0 \leq \eta \leq 1$.

The expression $V_A^2(t)$ is linearized in the operating region from 0 to 150 V (frequently the voltage present in non-industrial places), so the Laplace transform can be applied, presenting

operating points equal to 0 and 116,13 V, with $R^2 = 86,52\%$, as outlined in Figure 4.

Thus, the transfer function of the heating actuator in the operating region is according to Equation 10.

$$\frac{P_{TEFS}(s)}{V_{AS}(s)} = \frac{116,13\eta}{R_A} \quad (10)$$

Cooling Actuator Modeling

In this work, a fan was adopted as an actuator for cooling the interior of the bacteriological incubator, as shown in Figure 1d. The fan has air flow corresponding to the rate of variation of volumetric flow of the fluid perpendicular to the plane that contains it, as described in Equation (11).

$$Q_{AR}(t) = \frac{dV_{AR}(t)}{dt} = A_{ST}v(t) \quad (11)$$

The fluid velocity has a natural portion ($v_N(t)$) and a forced portion ($v_F(t)$) resulting from the non-action and action of the fan, respectively. Since $v_F(t)$ is directly proportional to the rotation speed at the tip of the fan blades ($v_{rotation}(t)$) and this work addresses the process as LTI, $v_F(t) = k_1 v_{rot}(t)$. Therefore:

$$Q_{AR}(t) = A_{ST}(v_N(t) + v_F(t))$$

$$Q_{AR}(t) = A_{ST}v_N(t) + A_{ST}k_1v_{rot}(t) \quad (12)$$

with $k_1 > 0$.

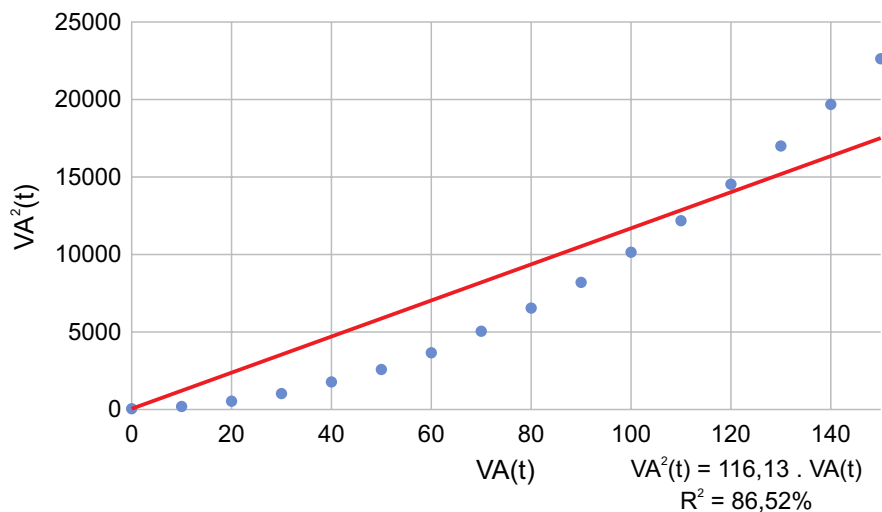
The angular rotation speed of the fan is directly proportional to the rotation speed at the tip of its blades ($v_{rot}(t) = R\omega(t)$), the cross-sectional area, perpendicular to the plane where there is the volumetric air flow, is the area of the circle that comprises the fan blades, see Figure 1d ($A_{ST} = \pi R^2$). Furthermore, $A_{ST}v_N(t)$ expresses the natural volumetric air flow rate (Q_{ARN}), which is considered constant, and $k_1\pi = k_{vent}$ as a factor related to ventilation, we arrive at Equation (13).

$$Q_{AR}(t) = Q_{ARN} + k_{vent}R^3\omega(t)$$

$$\frac{Q_{ARs}(s)}{\omega_s(s)} = k_{vent}R^3 \quad (13)$$

Since the volumetric air flow and the forced output thermal power are directly proportional and, once again, this work addresses the process as LTI, Equation (14) is defined, where $k_T > 0$ and expresses a thermal constant of the fan.

Figure 4. Linearization graphic of $V_A^2(t)$.



Source. Own authorship

$$\frac{P_{TSFs}(s)}{Q_{ARs}(s)} = k_T \quad (14)$$

By taking the product of the transfer expressions in Equations (13) and (14), we arrive at the relationship between the forced output thermal power and the angular rotation speed of the fan, shown in Equation (15).

$$\frac{P_{TSFs}(s)}{\omega_s(s)} = k_{vent} R^3 k_T \quad (15)$$

The angular rotation speed of the fan is caused by the activation of a DC electric motor present in it, modeled as follows:

$$\frac{\omega_s(s)}{V_{Rs}(s)} = \frac{k_m}{R_a J} \cdot \frac{1}{s + \frac{k_c}{J}} \quad (16)$$

Consequently, by taking the product of Equations (15) and (16), the actuator input-output relationship for cooling is obtained according in Equation (17).

$$\frac{P_{TSFs}(s)}{V_{Rs}(s)} = G_{vent} \frac{1}{s + \frac{k_c}{J}} \quad (17)$$

where $k_m > 0$ is the motor torque constant, $k_c > 0$ the load torque constant, $R_a > 0$ the armature resistance, $J > 0$ the motor moment of inertia and

$$G_{vent} = \frac{k_{vent} k_m k_T R^3}{R_a J} \text{ the static gain.}$$

Controller Design

The compensation design of this process relies on P (Proportional), PI (Proportional- Integral) and PD (Proportional-Derivative) controllers, aiming to meet the following performance specifications: time constant in closed loop lower than in open loop; offset in steady state less than or equal to 2% for the step input; and zero natural oscillation.

Tables 1 and 2 describe the controller gains calculated for heating and cooling processes, respectively.

Table 1. Controllers gains for the heating process.

	P	PI	PD
K_p	$\geq \frac{21,518R_A}{\eta R_T}$	$> \frac{0,0086R_A}{\eta R_T}$	$\geq \frac{21,518R_A}{\eta R_T}$
K_i	0	$\frac{K_p}{R_T C_T}$	0
K_d	0	0	$R_T C_T K_p$

Table 2. Controllers gains for the cooling process.

	P	PI	PD
K_p	$\geq \frac{2499k_c}{G_{vent} R_T J}$	$\leq \frac{C_T \left(\frac{k_c}{2J}\right)^2}{G_{vent}}$	$\frac{K_d}{R_T C_T}$
K_i	0	$\frac{K_p}{R_T C_T}$	0
K_d	0	0	$\geq \frac{2499k_c C_T}{G_{vent} J}$

Results

To verify the developed mathematical model, a case study for a commercial bacteriological incubator was assumed, so to evaluate the dynamic behavior of this equipment.

Bacteriological Incubator Characteristics

The BF115 bacteriological incubator, manufactured by BINDER, is adopted as a reference [7]. It has a heating time to 37 °C of 480 seconds (8 minutes), information shown in the datasheet as “Heating up time to 37 °C”, and this is the establishment time t_s for the step input.

From this information, the time constant of the open loop process (τ) is a quarter of the value of the settling time, resulting in $\tau = R_T C_T = 120$ seconds (2 minutes).

For the other constants, an incandescent lamp with a 90% conversion factor (η), a 100 W power

module for 110 Vrms AC power, resulting in $R_A = 242 \Omega$, and a household fan with 15 cm radius and average measurements found in the market for the other constants.

All parameter values used in the simulation are present in Table 3.

Computer Simulation

From the system parameters described in Table 3, the bacteriological incubator was simulated in MATLAB/Simulink 2024b software, applying P, PI and PD controllers in the closed-control loop, whose gains obtained for heating and cooling processes are presented in Table 4 and 5, respectively.

In heating process simulation shown in Figure 5, the P, PI, and PD controllers simultaneously meet the performance specifications. However, P and PD present gains that accelerate the setpoint tracking process, which consequently requires actuator efficiency that makes it inconvenient for practical implementation. In contrast, the PI presents an acceptable response, both for the design requirements and practical implementation, thus indicating the choice among the controllers mentioned.

For the cooling process shown in Figure 6, the P and PI controllers cannot simultaneously meet these three design requirements. For the P controller, one must choose to either nullify the natural oscillation or respect the offset requirement. For the PI controller, due to the controller's zero offset with the bacteriological incubator pole and the selected measurements for the simulation constants, the closed-loop time constant requirement for the process is not met.

In contrast, the PD controller simultaneously meets all the performance specifications; however, it offers gains that accelerate the setpoint tracking process, making it inconvenient for practical implementation. Furthermore, despite the PD controller's null natural oscillation, a sudden setpoint change causes oscillation in the output due to the process's setpoint tracking speed. In short,

Table 3. Parameter values used in simulation for bacteriological incubator compensation.

Bacteriological incubator	Incandescent lamp	Fan
$R_T = 0.83$	$\eta = 0.9$	$G_{vent} = 0.03$
$C_T = 144.46$	$R_A = 242$	$k_c = 0.0001$
		$J = 0.007$

Table 4. Controller gains for heating process simulation.

	P	PI	PD
K_p	7000	10	7000
K_i	0	0.083	0
K_d	0	0	840000

Table 5. Controller gains for cooling process simulation.

	P	PI	PD
K_p	1200	0.12	1250
K_i	0	0.001	0
K_d	0	0	150000

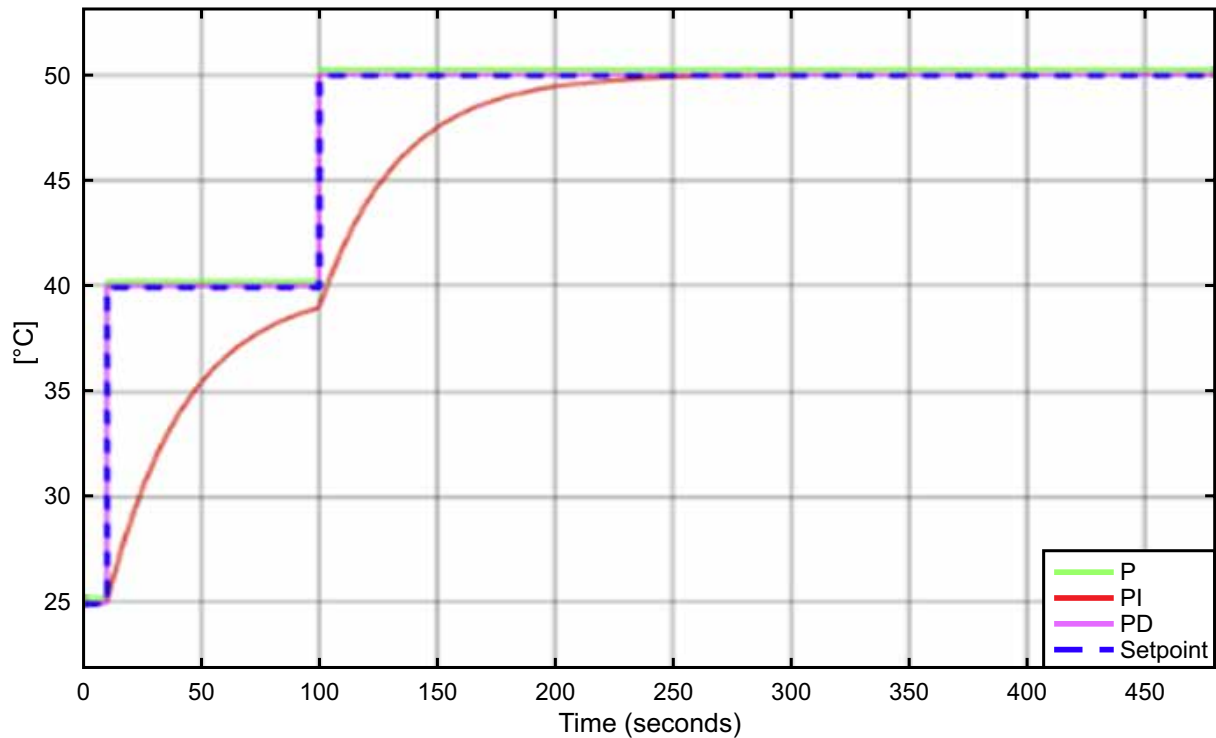
none of these three controllers are recommended for implementation in healthcare equipment.

Conclusion

This work modeled and simulated a bacteriological incubator, aiming at setpoint tracking and characteristics found in commercial versions of the equipment.

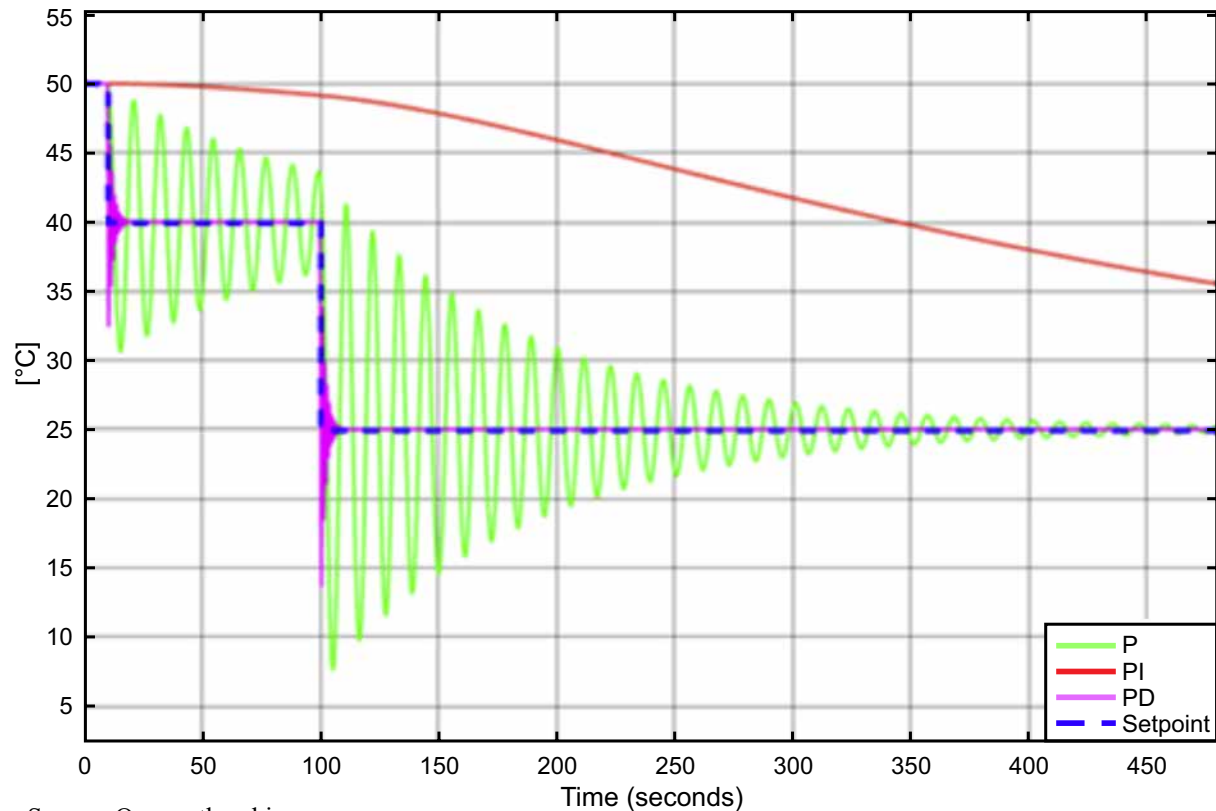
For future work, it is recommended to add noise to the process loop and PID compensation to assess output behavior due to unwanted inputs, such as ambient temperature variations, and to assess how optimized the PID is compared to the others mentioned in this work for this process.

Figure 5. Heating process curve response.



Source. Own authorship

Figure 6. Cooling process curve response.



Source. Own authorship

Furthermore, we propose producing a prototype of this process to further verify the results presented in this work.

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