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TRANSIENT ANALYSIS OF A HEATING AND HUMIDIFICATION SYSTEM FOR MECHANICAL VENTILATOR AIR FOR PATIENTS IN INTENSIVE CARE

Francisco Kleber Regis Castro
Universidade Federal do Paraná, Programa de Pós-Graduação em Engenharia Mecânica - PGMEC
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
klebercastro22@hotmail.com

Murilo Gasparin Rampi
Universidade Federal do Paraná, Programa de Pós-Graduação em Engenharia e Ciência dos Materiais - PIPE
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
murilorampi@gmail.com

Isabela Fernanda Rocha Corrêa
Universidade Federal do Paraná, Departamento de Engenharia Mecânica
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
isabelafrc@gmail.com

Lauber de Souza Martins
Universidade Federal do Paraná, Programa de Pós-Graduação em Engenharia e Ciência dos Materiais – PIPE, Núcleo de Pesquisa e Desenvolvimento de Energia Autossustentável - NPDEAS
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
AdventHealth University –Department of Health and Biomedical Sciences
Orlando FL, USA - 32803-1226
lauber@ufpr.br

Fernando Gallego Dias
Universidade Federal do Paraná, Programa de Pós-Graduação em Engenharia e Ciência dos Materiais – PIPE, Núcleo de Pesquisa e Desenvolvimento de Energia Autossustentável - NPDEAS
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
gallego@ufpr.br

José Viriato Coelho Vargas
Universidade Federal do Paraná, Departamento de Engenharia Mecânica; Núcleo de Pesquisa e Desenvolvimento de Energia Autossustentável - NPDEAS
Cx. P. 19011 – 81531-990 – Curitiba, PR, Brazil
vargasjvcv@gmail.com

Abstract. *Mechanical ventilators are equipment built to function and provide artificial respiration to patients who are clinically unstable or unable to breathe spontaneously. This is the case for patients infected with severe acute illness and severe respiratory syndrome coronavirus 2 (SARS-CoV-2). International recommendations standardize that the air supplied to the mechanically ventilated patient should have 100% relative humidity and temperature between 31 °C and 35 °C. A transient mathematical model of a mechanical ventilator with an air heating and humidification system was developed to provide air with properties recommended by international guidelines, which will bring less discomfort to patients during intubation and a speedy recovery. The results found were relevant and satisfactory. It can be established that the proposed mechanical ventilator is a possible solution of simple equipment for intensive care units and hospital emergencies that today face problems in meeting the parameters according to international recommendations and thereby efficiently control parameters such as temperature and relative humidity of the air output in real-time, avoiding inflicting medical problems in ICU beds, such as hypothermia, hyperthermia, mucus, lesions in the bronchial tree, mucosal ulceration, inflammation, and cryostasis. In contrast to these conditions, the proposed control system aims to establish and restore lives.*

Keywords: *Mathematical Model and Simulation, Air Humidification, Control Parameters, Temperature, Relative Humidity*

1. INTRODUCTION

The increased rates of disease dissemination and the number of critically ill patients who require the use of mechanical ventilation due to COVID-19 have raised scientific questions about the specific methods and equipment management for COVID-19 treatment. It is true that the correct handling of the parameters of mechanical ventilators reduces mortality and reduces health costs for patients, however, with constant development and the search for more efficient procedures that allow a speedy return to spontaneous breathing, it is concluded that early intubation is essential for successful treatment (Suzumura *et al.*, 2020).

As delayed intubation may worsen the outcome of coronavirus disease 2019 (Vita *et al.*, 2021), the demand for mechanical ventilation has exponentially increased during the COVID-19 pandemic. Before the pandemic, endotracheal intubation was mostly reserved for patients with high-energy trauma or a risk of impaired consciousness. However, for COVID-19, the recommendations are for early intubation of COVID-19 positive patients with the oxygen demands higher than 5 Liters per minute to keep oxygen saturation at a minimum of 93%, even for patients without impaired consciousness (Associação de Medicina Intensiva Brasileira, 2020).

With the complexity and high cost of production of mechanical ventilators, the uncertainty of the parameters applied in artificial respiration, and the growing concern about the availability of devices in ICU beds, the study of optimization of existing mechanical ventilation equipment is necessary, so that they meet the new demand for low cost and fast manufacturing processes. To recognize the ventilation mode, the adequacy of the inspiratory and expiratory time, the ability to detect changes in the patient's condition, and monitoring through the volume, pressure, and flow curves over time are essential (Vita *et al.,* 2021).

Mechanical ventilators are devices used to help clinically unstable patients or patients who are not capable of breathing spontaneously to recover their health. This is the case for critical patients who are experiencing acute respiratory distress syndrome due to COVID-19. It was experimentally verified that the temperature at the trachea level of a normal person varies between 32 ºC and 34ºC, the absolute humidity (AH) varies between 25 and 35mgH_2O L^{-1} and the relative humidity (RH) is approximately 95% (Shelly, 1998; Branson, 1999). Based on the experimental data, the Compêndio de Medicina Intensiva (2004) published by the Brazilian Association of Intensive Care (Associação Brasileira de Medicina Intensiva) recommends that the temperature of the air provided to intubated patients (under mechanical ventilation) must be between 32 ºC and 34ºC and the relative humidity between 95% and 100%. Internationally, similar conditions were standardized to provide air a with relative humidity (RH) of 100%, an absolute humidity (AH) of 36 to 40 mgH_2O L^{-1}, and a temperature between 31ºC and 35°C (Cairo, 2013).

As seen by those guidelines, air humidity plays an important role in mechanical ventilation. The control of humidity must be accurate enough so that any response of the patients, while intubated, can be associated with their physiological interaction with medication and not with the excess lack of humidification of the inhaled air. Studies have shown that data regarding humidity presented by manufacturers are questionable (Lellouche *et al.,* 2009).

Additional measurement instruments have been used in practice to monitor humidity during the use of mechanical ventilators, such as hygrometer-thermometers, which unfortunately are not always available (Ashry and Modrykamien, 2014). Therefore, other ways of monitoring humidification are employed, such as secretion characteristics. There is a direct relation between the volume of the secretion and the level of humidification. High levels of humidification lead to an increase in secretion volume and low humidification leads to crusting and a decrease in volume (Sottiaux, 2006).

Overheating is also an issue in mechanical ventilation. High air temperature may cause localized hyperthermia, and increase patient metabolic activity to maintain normothermia. Such an increase in temperature may also cause protein, enzyme, cardiac, and respiratory dysfunctions and even delay extubation. Conventional mechanical ventilators do not have temperature sensors measuring the local temperature of the insufflated air, only the oscillation of the temperature is measured. Under this circumstance, the temperature of the air at the interface, tube-patient is not controlled (Sottiaux, 2006; Ashry and Modrykamien, 2014; Ferreira *et al*., 2018).

As pointed out by Oliveira *et al.* (2019), the heating of the inspired air and the active humidification systems are not being carried out adequately, since most of the time (65.8% according to the study), the heating level was below expected, corresponding to 30ºC - 33ºC. At the same time, it was noted that, on 87.5% of occasions, the amount of water present in the humidifier cups was different from the recommended value, noting that they have an inversely proportional relationship with the viscosity of the secretions of patients on invasive mechanical ventilation.

This study aims to present a mathematical model that predicts the response of a fan connected to an oil-free compressor at one end and the patient at the other end. The model will be used for real-time control of air temperature and humidity at the machine/patient interface (trachea) to keep the air insufflated under the conditions recommended by the protocols. The model will be used to design a ventilator that controls parameters such as temperature and humidity in real time to avoid inflicting additional medical conditions on patients caused by poor temperature and humidity control. The proposed study differs from other research on the subject of mechanical ventilation for having the ability to present a thermodynamic system that can be optimized through a programming code and as a suggestion for future work on the control of parameters by an automation process.

2. MATHEMATICAL MODEL

The mathematical model for the transient regime is written for the proposed system represented schematically in Figure 1. The system is composed of an acrylic box with volume $V = L\,H\,t_w$, length L, height H, and width t_w. The governing equations are based on the principles of conservation of species and energy.

The humidity of the air entering the system is a known parameter, as shown in Figure 1. Based on this, the mass flow rate of moist air delivered by a mechanical ventilator that does not adequately treat the air before delivery to the patient is written as:

$$\dot{m}_{in} = \dot{m}_{a,in} + \dot{m}_{v,in} \tag{1}$$

where $\dot{m}_{a,in}$ is the dry air inlet mass flow rate, and $\dot{m}_{v,in}$ is the water vapor inlet flow rate.

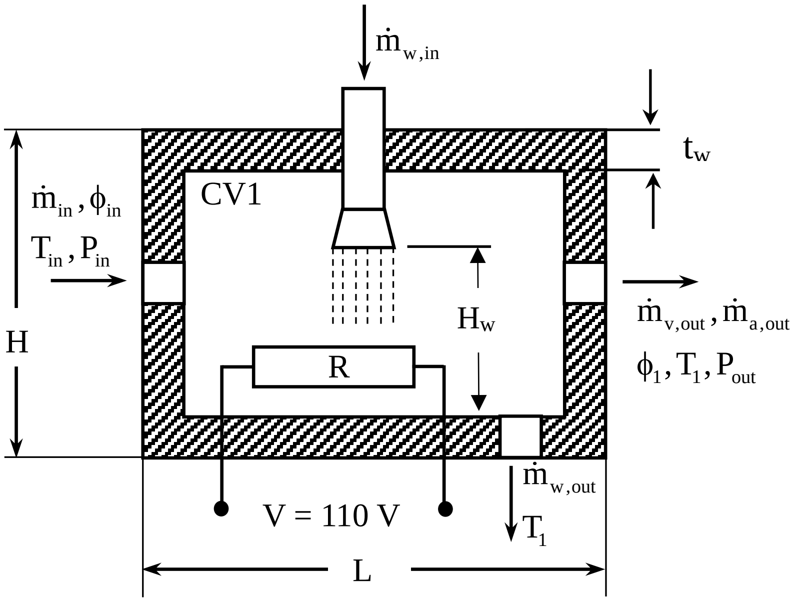


Figure 1. Illustration of the control volume (acrylic box) referring to the mathematical model of the proposed system.

The steam inlet pressure, the mole fraction of steam, and the absolute humidity are calculated according to the equations below:

$$P_{v,in} = \phi_{in} P_{v,sat}(T_{in})\,;\, x_{v,in} = \frac{P_{v,in}}{P_{in}} \tag{2}$$

$$\omega_{in} = 0.622 \frac{P_{v,in}}{P_{in} - P_{v,in}} = \frac{\dot{m}_{v,in}}{\dot{m}_{a,in}};\ \dot{m}_{v,in} = \omega_{in}\dot{m}_{a,in} \tag{3}$$

where $P_{v,in}$ is the inlet steam pressure, ϕ_{in} is the inlet relative humidity, T_{in} is the inlet temperature of moist air, $P_{v,sat}(T_{in})$ is the saturation pressure of the steam at temperature T_{in}, $x_{v,in}$ is the mole fraction of steam inlet, P_{in} is the total pressure of humid air intake, and ω_{in} is the absolute humidity of the air intake.

The mass flow rates can then be calculated as follows:

$$\dot{m}_{a,in} = \frac{\dot{m}_{in}}{1+\omega_{in}};\ \ \dot{m}_{v,in} = \dot{m}_{in} - \dot{m}_{a,in} \tag{4}$$

Applying the species conservation principle in CV1, we obtain the following relation:

$$\dot{m}_{v,out} = \dot{m}_{v,in} + \frac{dm_{v1}}{dt} \tag{5}$$

for which we assume a quasi-static operating system, so that at any time step, Δt the rate of water vapor generation, $\dot{m}_{v1}$ and the mass of water vapor m_{v1}, within CV1 are obtained by solving the problem starting value (PVI):

$$\dot{m}_{v1} = \frac{dm_{v1}}{dt} = \bar{h}_m A_{mt}(\rho_w - \rho_1) \tag{6}$$

where $m_{v1} = m_{v1,0}$ when t = 0 s, $\dot{m}_{v,out}$ is the mass flow rate of steam output, $m_{v1,0}$ is the initial mass of steam in CV1, t is the time, $\rho_w = 1/v_g(T_1)$ is the density of the steam at the air/liquid water jet interface, $v_g(T_1)$ is the specific volume of the steam saturated at temperature T_1, $\rho_1 = m_{v1} / V_1$ is the mass concentration of steam in CV1, A_{mt} is the total mass transfer area defined by the number of jets of the sprinkler nozzle that is shown in Figure 1, and $\bar{h}_m$ is the average mass transfer coefficient determined through the empirical correlation of Dittus and Boelter for ducts according to the equation below (Bejan, 2014):

$$\bar{h}_m = \frac{D}{D_h}\overline{Sh}\,;\ \overline{Sh} = 0.024\ Sc^{0.4}\ Re_{D_h}^{0.8} \tag{7}$$

Equation (7) is valid for $Sc \geq 0.5$ $2500 \leq Re_{D_h} \leq 10^6$ and, where is $\overline{Sh}$ the average Sherwood number, $Re_{D_h} = U_a D_h / \nu_a$ is the Reynolds number based on the hydraulic diameter $D_h = 4\ W\ H/p$, $p = 2\ (W + H)$ is the perimeter of the duct (box), $U_a = \dot{m}_{w,in} / (n\rho_w A_c)$ is the average relative air/water jet velocity, $\dot{m}_{w,in}$ is the mass flow rate of liquid water coming from the reservoir below CV1, *n* is the number of orifices in the sprinkler jet, ρ_w is the density of the liquid water, $A_c = \pi\ d_o^2 / 4$ is the cross-section of each water jet, d_o is the diameter of the orifice of each water jet water, ν_a is the kinematic viscosity of air, $Sc = \nu_a / D$ is the Schmidt number, and *D* is the mass diffusivity of vapor in the air.

The total mass of dry air in CV1, m_{a1}, the mole fraction of water vapor in CV1, x_{v1}, the partial pressure of water vapor in CV1, P_{v1}, and the relative humidity in CV1, ϕ_1, are calculated as follows:

$$m_{a1} = \rho_a(T_1)\ \forall_1 \tag{8}$$

$$x_{v,1} = \frac{m_{v1} / M_{H_2O}}{m_{v1} / M_{H_2O} + m_{a1} / M_a} \tag{9}$$

$$P_{v,1} = x_{v,1} P_{in} \tag{10}$$

$$\phi_1 = \frac{P_{v,1}}{P_{v,sat}(T_1)} \tag{11}$$

where $\rho_a(T_1)$ is the density of air T_1, M_{H_2O} is the molecular mass of water, M_a is the molecular mass of dry air, and $P_{v,sat}(T_1)$ is the saturation pressure of water at temperature T_1.

Considering the dry airflow to be incompressible and the principle of conservation of mass, we obtain:

$$\dot{m}_{a,out} = \dot{m}_{a,in} \tag{12}$$

Considering that the air/water mixture leaves CV1 with relative humidity ϕ_1, that is, with known vapor pressure $P_{v,1}$, it is possible to determine the CV1 output mass flow as follows:

$$\omega_1 = 0.622 \frac{P_{v,1}}{P_{in} - P_{v,1}} = \frac{\dot{m}_{v,out}}{\dot{m}_{a,out}} \tag{13}$$

where ω_1 is the absolute humidity at CV1. Soon,

$$\dot{m}_{v,out} = \omega_1 \dot{m}_{a,out} \tag{14}$$

and

$$\dot{m}_{out} = \dot{m}_{a,out} + \dot{m}_{v,out} \tag{15}$$

where $\dot{m}_{out}$ is the mass flow rate of the humid air output.

Consequently, applying the mass balance for liquid water at CV1 we obtain:

$$\dot{m}_{w,out} = \dot{m}_{w,in} - \dot{m}_{v1} \tag{16}$$

where $\dot{m}_{w,out}$ is the mass flow rate of liquid water leaving CV1 and returning to the water reservoir below CV1.

The initial condition of Equation (4), the mass of steam in CV1, $m_{v1} = m_{v1,0}$ when t = 0 s, is calculated as follows:

$$m_{v1,0} = \frac{x_{v,in} m_{a1} M_{H_2O}}{(1 - x_{v,in}) M_a} \quad (17)$$

Radiant heat transfer is not considered, as air is a nonparticipating medium. Therefore, the energy balance for CV1 is expressed as follows:

$$-U_\infty A_\infty (T_1 - T_\infty) + \dot{Q}_R + \dot{m}_{a,in} c_{p,a} (T_{in} - T_1) + \dot{m}_{v,in} h_g (T_{in}) - \dot{m}_{v,out} h_g (T_1) = C_{th,1} \frac{dT_1}{dt} \quad (18)$$

where, U_∞ is the global heat transfer coefficient between CV1 and the environment, for $H = L = t_w$, that is, a cubic box, $A_\infty = 6\ H^2$ is the total heat transfer area, T_∞ is the ambient temperature, $\dot{Q}_R = V^2 / R$ is the heat generated by the electrical resistance, V is the electrical voltage, R is the electrical resistance of the heater, $c_{p,a}$ is the specific heat at constant air pressure, $h_g(T_{in})$ is the specific enthalpy of saturated steam at temperature T_{in}, and $h_g(T_1)$ is the specific enthalpy of saturated steam at temperature T_1.

The thermal capacity of CV1, $C_{th,1}$, is determined as follows:

$$C_{th,1} = m_m c_m + m_{a1} c_{v,a} + m_{v1} c_{v,v} + m_{w1} c_w + m_{acr} c_{acr} \quad (19)$$

where m_m is the mass of the electrical resistance made of brass, c_m is the specific heat of the brass, $c_{v,a}$ is the specific heat of air at constant volume, $c_{v,v}$ is the specific volume of water vapor at constant volume, $m_{w1} = \rho_w n\ A_c H_w$ is the mass of water inside CV1, c_w is the specific heat of liquid water, $m_{acr} = 6\ \rho_{acr} H^2 e_{acr}$ is the mass of the acrylic box, ρ_{acr} is the density of the acrylic, and e_{acr} is the thickness of the wall.

3. RESULTS

In this section, we present the results of the proposed control volume mathematical model (CV1) for controlling the temperature and relative humidity of the mechanical fan outlet. In Figures 2 and 3 below, the initial results of the proposed control system are shown and in Figures 4, 5, 6 and 7 the results of the parametric analysis used to understand the physical behavior of the system are shown. The values of the variables and physical properties necessary for the solution of the ordinary differential equations (ODEs) system in this study are shown in Table 1.

Parametric analysis was performed by varying only one parameter and keeping all other parameters of the model constant. We define the transient regime of the $t_{rt}(s)$ $t_{ru}(s)$ system as the time required for the air conditions in the proposed control volume to reach the temperature and relative humidity specified by international protocols, i.e., 309.65 K (36.5°C) and 98%.

Table 1. Variables and physical properties for the reference case.

$c_{acr} = 2{,}16\ kJ\ kg^{-1}K^{-1}$	$m_m = 0{,}2\ kg$	$T_\infty = 298{,}15\ K$
$c_{p,a} = 1{,}005\ kJ\ kg^{-1}K^{-1}$	$M_a = 28{,}97\ kg\ kmol^{-1}$	$U_\infty = 1{,}5\ W\ m^{-2}\ K^{-1}$
$c_{p,m} = 0{,}38\ kJ\ kg^{-1}K^{-1}$	$M_{H_2O} = 18\ kg\ kmol^{-1}$	$V = 110\ V$
$c_{p,v} = 1{,}864\ kJ\ kg^{-1}K^{-1}$	$\dot{m}_{in} = 0{,}01\ kg\ s^{-1}$	$\rho_{acr} = 1300\ kg\ m^{-3}$
$c_{v,a} = 0{,}718\ kJ\ kg^{-1}K^{-1}$	$\dot{m}_{w,in} = 0{,}01\ kg\ s^{-1}$	$\rho_w = 1000\ kg\ m^{-3}$
$c_{v,v} = 1{,}402\ kJ\ kg^{-1}K^{-1}$	$n = 10$	$\rho_\infty = 1{,}125\ kg\ m^{-3}$
$d_o = 0{,}003\ m$	$P_{in} = 1{,}2 \times 10^5\ N\ m^{-2}$	$\upsilon_{ar} = 1{,}58 \times 10^{-5}\ m^2\ s^{-1}$
$D = 2{,}5 \times 10^{-5}\ m^2\ s^{-1}$ @300 K	$P_\infty = 1{,}0 \times 10^5\ N\ m^{-2}$	$\phi_{in} = 0{,}2$
$e_{acr} = 0{,}00635\ m$	$R = 70\ \Omega$	
$L = 0{,}2\ m$		

The reference variables and physical properties presented in Table 1 were used to generate the results presented in Figures 2 and Figure 3 for three values of the humid air inlet temperature in CV1 (293.15 K; 298.15 K and 303.15 K). It is noted that the results obtained are by the expected physical trends, that is, higher humid air inlet temperatures mean that the system has a shorter transition time until the temperature reaches the desired value according to international standards (309.65 K).

The sawtooth behavior shown in Figure 2 is due to the adopted control system. When the temperature reaches 309.65 K, the electrical resistance is turned off, that is, heat generation ceases and consequently, the system temperature begins to drop. When the temperature reaches 308.65 K, the electrical resistance is turned on again and heat generation resumes. The temperature then oscillates between 308.65 K and 309.65 K as the electrical resistance turns on and off, respectively, which gives the observed sawtooth characteristic.

This behavior was observed experimentally by Ferreira (2006), which confirms that the mathematical model captures the expected physical trends according to experimental data documented in the scientific literature.

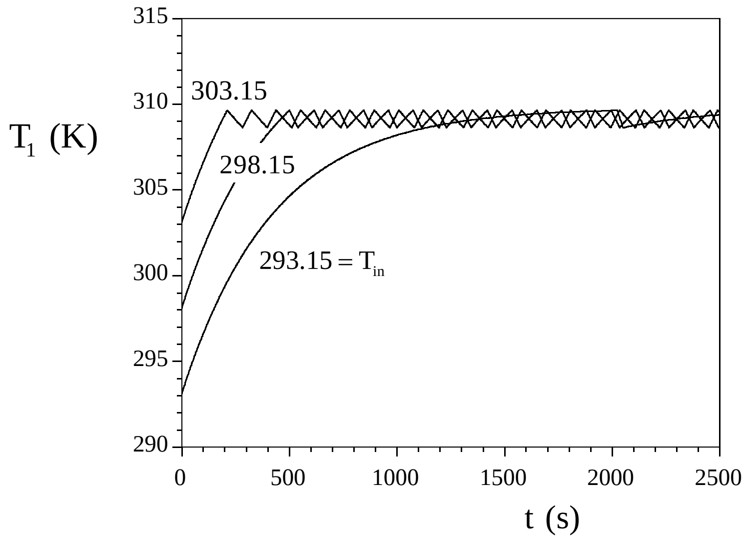


Figure 2. Temporal evolution of temperature in CV1 for different humid air inlet temperatures.

The control system adopted for humidity was a mechanism similar to that adopted for temperature control. When the relative humidity is below 85%, we allow water to enter CV1 ($\dot{m}_{w,in}$) at a flow rate of 0.01 kg/s.

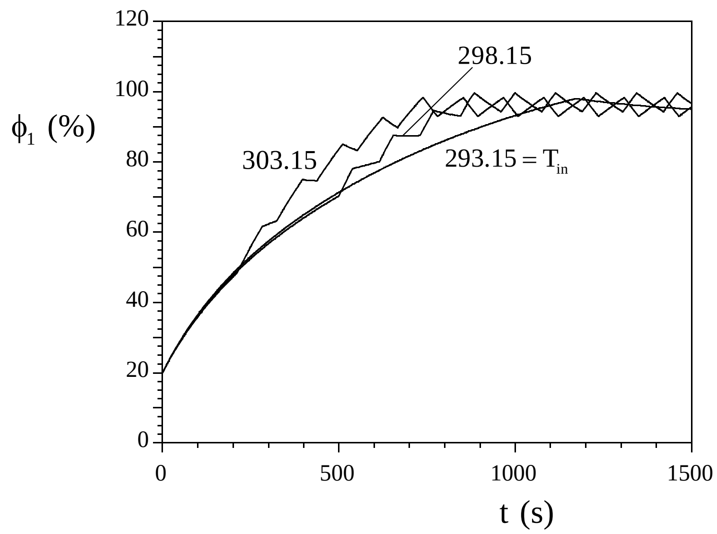
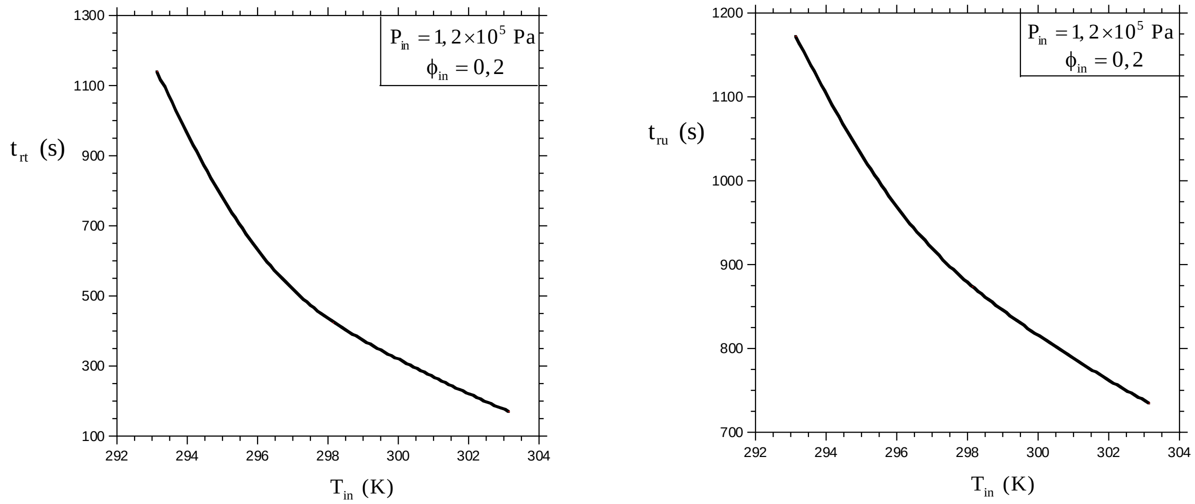


Figure 3. Temporal evolution of relative humidity in CV1 for different humid air inlet temperatures.

Figures 4 and 5 show the effect of humid air inlet temperature on the time to reach the recommended air supply temperature and relative humidity, respectively graphs Figure on the left and Figure on the right.

The graphs prove that for higher humid air inlet temperatures, less energy is needed for the air to reach the recommended temperature $t_{rt}(s)$ and relative humidity $t_{ru}(s)$, that is, the electrical resistance must remain on for less time, therefore the transition time is shorter for both parameters analyzed.



Figures 4 e 5. Effect of moist air inlet temperature on time to reach recommended air temperature and relative humidity.

Figure 6 below shows the water vapor density gradient in the control volume. The result shows that the water vapor density gradient inside CV1 depends on temperature, as it increases with time and this also increases with increasing humid air inlet temperature, Tin.

The graph curves show the same qualitative behavior, but we see that (T_{in} = 303.15 K) has a significant quantitative influence on the different curves. The variation in water vapor mass inside CV1 is proportional to the water vapor density gradient, $\Delta\rho = \rho_w - \rho_1$ therefore allowing more water vapor to diffuse into the air inside the mechanical fan (CV1).

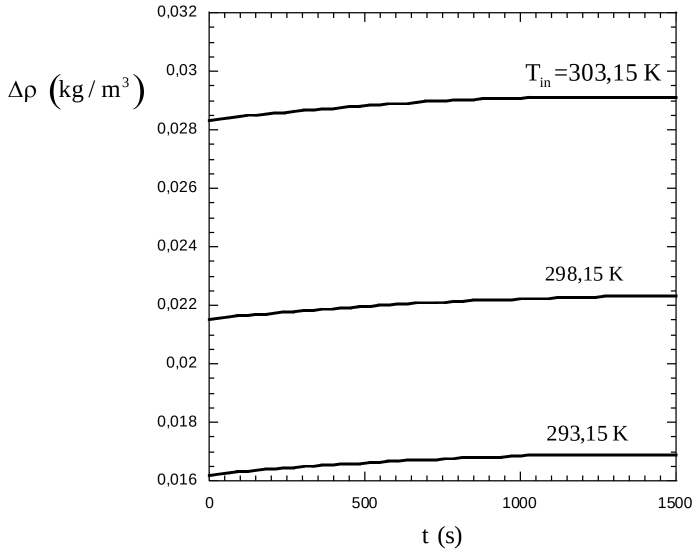


Figure 6. Effect of humid air inlet temperature on the water vapor density gradient ($\Delta\rho = \rho_w - \rho_1$) inside CV1.

Figure 7 below shows the influence of the humid air inlet pressure on the system's transient regime for the humid air inlet temperature T_{in} = 293.15 K and initial air relative humidity of 20%. We draw attention to the fact that the outlet temperature increased by only 0.6%, therefore, we can assume that the influence of the P_{in} pressure on the transient regime time $t_{rt}(s)$, is negligible. However, the P_{in} pressure has a more significant influence on the $t_{ru}(s)$ transient time. As the moist air inlet pressure increases, the $t_{ru}(s)$ time decreases. According to Equations 4.10 and 4.11, we can see that the relative humidity is directly proportional to the P_{in} pressure, therefore, the increase in the moist air inlet pressure causes the system humidity to increase by the same molar fraction of water vapor, which reduces the amount of water to be added by the sprinkler nozzle, thus reducing the time needed to reach the desired humidity.

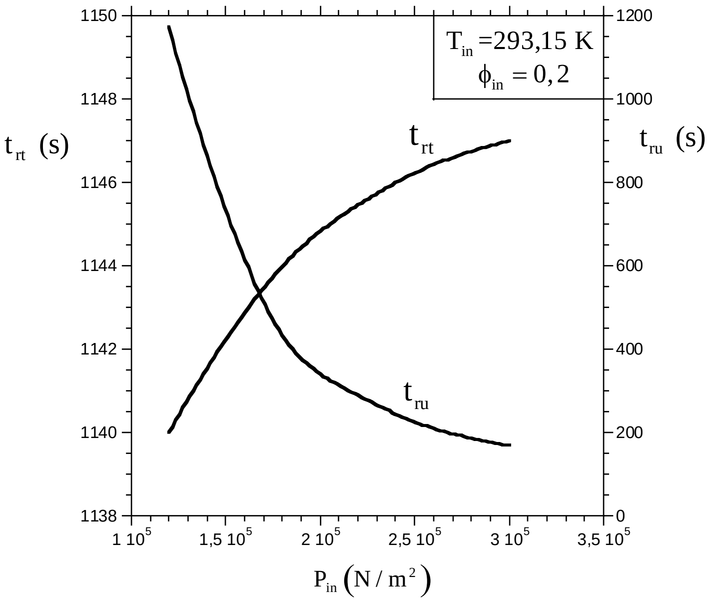


Figure 7. Effect of humid air inlet pressure on the transition time for initial temperature conditions of 293.15 K and relative humidity of 20%.

Figure 8 below shows the effects of sprinkler nozzle diameter on transition times. The result shows that as the diameter increases, the box moistens more slowly, with impact and resulting in higher values of $t_{rt}(s)$ and $t_{ru}(s)$. Two important effects are observed that oppose each other:

- Effect 1: mass transfer area

When the diameter of the spray nozzle increases, the surface of the water jet cylinder that forms inside the mechanical fan (CV1) increases, which the area of water mass exchange between the liquid column and the air. Therefore, the desired relative air humidity is achieved in a shorter time when compared to smaller sprinkler nozzle diameters;

- Effect 2: speed of the liquid water jet

When we keep the mass flow rate of liquid water that enters the fan constant and increase the diameter of the spray nozzle, we see that the droplet speed decreases. We consider that the box is small enough to maintain the liquid water jet between the top and bottom of the fan with a constant diameter. As the velocity of liquid water decreases with increasing diameter of the spray nozzle, the Reynolds number and Sherwood number also decrease, which causes a reduction in the mass transfer coefficient (Equation 4. 7), which also reduces the humidification rate of CV1.

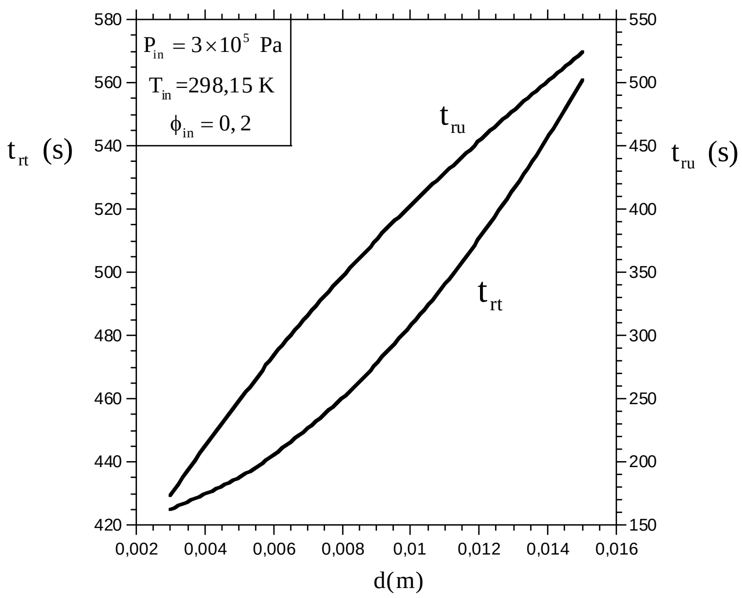


Figure 8. Effects of sprinkler nozzle diameter on transient time.

Figure 9 below shows the analysis of the impact and effects of the dimensions of the control volume (CV1) on the transient regime times of temperature and relative humidity. As expected, with larger dimensions, the transient regime time is longer. In other words, the larger the box is, the greater the thermal capacity of the system is required due to the greater quantity of matter to be heated, and, therefore, the longer it takes to reach the desired temperature and relative humidity. Likewise, more water vapor (diffusion process) must be transferred to a greater quantity of air, which also causes the transition times $t_{rt}(s)$ $t_{ru}(s)$ to be greater.

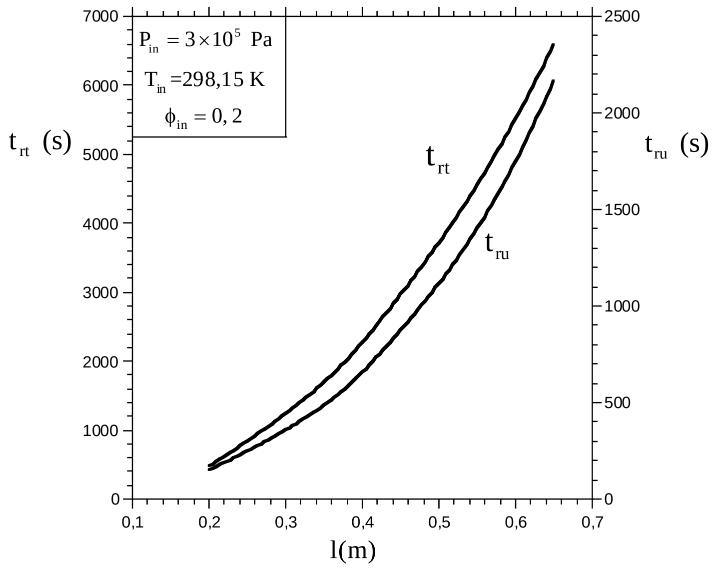


Figure 9. Effects of box dimensions on temperature transient time and relative air humidity.

4. CONCLUSION

This study presented the mathematical modeling and numerical simulation results of a mechanical ventilator with a heating and air humidification system for intensive care unit (ICU) patients. In this sense, the following concluded:

(i) A concept for a mechanical ventilation system was presented;
(ii) A mathematical model was developed that was capable of capturing the physical response expected for the system;
(iii) A computational code was created for numerical simulation and was able to generate satisfactory results based on the mathematical model;
(iiii) An effective control system for the temperature and relative humidity of the air entering the patient was proposed;
(iiiii) In the parametric analysis, all correct physical trends were captured by the mathematical model as shown in the results and discussion chapter.
In summary, the physical trends observed were as follows:

- For higher humid inlet temperatures, the system reaches the recommended temperature and relative humidity transition time more quickly;
- The water vapor density gradient is proportional to temperature, that is, the higher the humid air temperature is, the higher the water vapor density gradient;
- The influence of the humid air inlet temperature is more significant in the temperature transition time than the relative humidity transition time;
- The effect of the humid air inlet pressure on the transition time to reach the recommended temperature is negligible, while on the transition time of the recommended relative humidity it is significant and has a great impact (shorter time);
- By increasing the diameter of the spray nozzle, the transition time to reach the recommended temperature and relative humidity is increased;
- By increasing the size of the box (CV1 dimensions), the transition time to reach the recommended temperature and relative humidity is increased.

In this way, based on the conclusions listed and the physical trends mentioned above, it can be established that the proposed system is a possible simple equipment solution for the care of intensive care units and hospital emergencies.

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