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Design of a simulator for the energy evaluation of cold rooms

Diseño de un simulador para la evaluación energética de cámaras frigoríficas

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Abstract

This paper presents the design of a simulator to evaluate the performance of cold rooms, which allows to determine the feasibility of its implementation. The design is based on the selection of construction materials and the dimensions of the cold rooms, to later determine the thermal loads that allow estimating the heat to be removed from the product. From the thermal loads, the modeling of the thermodynamic cycle, simple steam compression cycle, is obtained the work of the compressor and the coefficient of performance of the system are obtained. The evaluation of the performance of the simulator was carried out for climatic conditions of the municipality of Apan Hidalgo, where the storage of poultry meat in a range of 0°C to 4°C was considered. The proposed cold room can store a maximum capacity of 500 birds. As a result, the simulator obtains the thermal loads associated with the cooling process, the behavior of the cooling cycle, the heat removed by the system, the work of the compressor, the number of cycles per day, the behavior of the temperature of the product inside the cold rooms and the energy consumed by the system.

Cold room, Poultry conservation, Energy consumption

Resumen

En este trabajo se presenta el diseño de un simulador para evaluar el desempeño de cámaras frigoríficas, lo que permite determinar la viabilidad de su implementación. El diseño parte de la selección de materiales de construcción y las dimensiones de la cámara, para posteriormente determinar las cargas térmicas que permitan estimar el calor a retirar del producto. A partir de las cargas térmicas se obtiene el modelado del ciclo termodinámico, ciclo simple de compresión a vapor, y se obtiene el trabajo del compresor y el coeficiente de desempeño del sistema. La evaluación del desempeño del simulador se realizó para condiciones climáticas del municipio de Apan Hidalgo, en donde se consideró el almacenamiento de carne avícola en un rango de 0°C a 4°C. La cámara frigorífica propuesta alcanza a almacenar una capacidad máxima de 500 carcasas. Como resultados el simulador obtiene las cargas térmicas asociadas al proceso de refrigeración, el comportamiento del ciclo de refrigeración, el calor retirado por el sistema, el trabajo del compresor, el número de ciclados al día, el comportamiento de la temperatura del producto dentro de la cámara y la energía consumida por el sistema.

Cámara frigorífica, Conservación avícola, Consumo energético

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1. Introduction

Simulation is developing more and more in different fields of study due to technological advances in software and hardware. The uses of simulation are given in different areas, in design and operation of systems, application in industrial projects, in economic systems, in health services, among others.

This work presents the design of a simulator that allows evaluating the operating conditions of a cold room, the simulator is intended to be a tool to determine the feasibility of using a certain cold room depending on the product to be stored and the geographical region where it will be installed.

The design of the cold room, which will be simulated to determine its behavior, will depend both on the variable operating conditions and the climatic characteristics of the region where it will be installed, seeking to keep the refrigerated product in the temperature range that allows its optimum preservation.

The modeled refrigeration system uses a simple compression cycle. Propane (R-290) is selected as the refrigerant fluid, as it is a natural refrigerant with good energy efficiency (Gas Servei. n.d.). At present, hydrocarbons such as propane are replacing commonly used refrigerants that damage the ozone layer or contribute to global warming.

The design of the simulator consists of the mathematical modeling of the state variables, the thermodynamic analysis of the cycle and the calculation of thermal loads. For this design it is necessary to know the characteristics of the product, the climatic conditions of the site, the properties of the construction materials of the chamber and the number of individuals that will enter the enclosure.

The simulator solves the differential equations corresponding to the mathematical model and delivers as results: the heat discarded by the condenser, the work done by the compressor, the number of cycles, the coefficient of performance (COP), the product temperature and the amount of energy consumed throughout the day.

2. Methodology

The methodology to be developed in this work is based on the following points:

- Design of the cold room.
- Mathematical modeling of the thermal loads.
- Modeling of the cold room
- Modeling of the refrigeration cycle
- Modeling of the cold room with refrigeration cycle
- Simulator integration
- Simulator evaluation

3. Development

3.1 Design of the cold chamber

An important part of the cold room design is to know the characteristics of the materials to be used in order to obtain the required insulation.

The proposed sizing for the cold room is shown in Table 1, having a heat transfer area of 40.9 m^2 and a volume of 23.6 m^3.

Cold storage	
Height (m)	3.50
Length (m)	2.70
Width(m)	2.50
Aisle width (m)	0.90
Storage area width (m)	0.90
Door	
Height (m)	2.50
Length (m)	0.90

Table 1 Refrigerator dimensions
Source: Own Elaboration

Polyurethane as an insulating material is considered the optimum material for the design of the enclosure due to its low thermal conductivity, since it allows less heat to pass into the interior of the cold room. Table 2 shows the materials of construction for cold rooms.

Wall material	Thermal conductivity ($\frac{W}{m°K}$)	Thickness (m)
Polyurethane	0.028	0.075
Brick	0.87	0.16
Mortar	1.40	0.04
Door material		
Polyurethane	0.028	0.02
Aluminum	205	0.05

Table 2 Construction Materials
Source: Own Elaboration

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The refrigeration system uses a simple compression cycle, using a hydrocarbon (HC) as refrigerant. It is proposed to use propane R-290 as it is a natural refrigerant, and is currently being used for industrial refrigeration replacing the R-134a. Table 3 shows the physical properties and Table 4 the flammability properties of the refrigerant.

Refrigerant R-290 (Propane)		
Lower flammability limit		Auto-ignition temperature (°C)
In volume (%)	In weight ($\frac{Kg}{m^3}$)	
2.10	0.038	460

Table 3 Physical properties
Source: (Gas Servei. n.d.).

Molecular weight	Boiling temperature (°C)	Critical temperature (°C)	Critical pressure (bar)	Calor latente ($\frac{KJ}{Kg}$)
44.1	-42	96.7	42.48	342

Table 4 Flammability Properties
Source: (Gas Servei. n.d.).

3.2 Calculation of thermal loads.

The thermal load is the amount of heat that must be removed from the site to be cooled, to reach the desired temperature, in the design of the simulator the following thermal loads are considered (Sanchez. M. & Pineda. I. 2001):

– Thermal load due to product cooling, is obtained by the following equation:

$$Q_P = \frac{m_p * Cp_p * \Delta T}{t} \tag{1}$$

Where:
m_p the mass of the product.
Cp_p is the specific heat of the product.
ΔT is the temperature difference, initial temperature minus the final temperature of the product.

– Thermal load due to transmission losses in walls, which is obtained from the following equation:

$$Q_{T1} = U_1 * A_1 * \Delta T \tag{2}$$

Where:
U_1 is the overall heat transfer coefficient in the walls.
A_1 is the area of the enclosure
ΔT is the temperature difference, outside air temperature minus inside air temperature.

The global coefficient in the walls, is obtained as:

$$U_1 = \frac{1}{\frac{1}{h_1}+\frac{e_1}{k_1}+\frac{e_2}{k_2}+\frac{e_3}{k_3}+\frac{1}{h_2}} \tag{3}$$

Where:
e_1 is the thickness of the insulation.
e_2 is the thickness of the brick.
e_3 is the thickness of the mortar.
h_1 is the convective coefficient inside the chamber.
h_2 is the convective coefficient on the outside of the cavity.
k_1 is the thermal conductivity of the insulation.
k_2 is the thermal conductivity of the brick.
k_3 is the thermal conductivity of the mortar

– Thermal load due to transmission losses at the door is obtained from the following equation:

$$Q_{T2} = U_2 * A_2 * \Delta T \tag{4}$$

Where:
U_2 is the overall heat transfer coefficient at the door.
$A_1$is the area of the door.

The overall coefficient at the door is obtained as:

$$U_1 = \frac{1}{\frac{1}{h_1}+\frac{e_1}{k_1}+\frac{e_4}{k_4}+\frac{1}{h_2}} \tag{5}$$

Where:
e_4 is the thickness of aluminum.
k_4 is the thermal conductivity of aluminum.

– Calculation of the sensitive load of people is obtained from the following equation:

$$Q_e = N_e * Qs_e \tag{6}$$

Where:
N_e is the number of people entering the enclosure.
Qs_P is the sensible heat per person.

– The thermal load produced by lamps is obtained from the following equation:

$$Q_l = N_l * P \tag{7}$$

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Where:
N_l is the total number of lamps
P is the lamp wattage

– Thermal load by infiltration of outside air is obtained from the following equation:

$$Q_I = Q_1 + Q_2 \quad (8)$$

Where:
Q_1 is the infiltrated heat due to technical air renewals.
Q_2 is the infiltrated heat due to equivalent air renewals

– The infiltrated heat due to technical air renewals is obtained from the following equations:

$$Q_1 = m * \Delta h \quad (9)$$

$$m = V * \rho * n \quad (10)$$

Where:
m is the mass of the air
Δh is the enthalpy difference of the air, inside and outside the enclosure
V is the volume of the enclosure
ρ is the air density
n is the number of technical air renewals

The infiltrated heat due to the equivalent air renewals is obtained by the following equations:

$$Q_2 = m * \Delta h$$
$$m = V * \rho * \theta \quad (11)$$
$$V = \frac{a*H}{4}\sqrt{0.072} * H * \Delta T \quad (12)$$

Where:
V is the air volume
θ is the door squeeze time
a is the width of the door
H is the height of the door
ΔT is the difference in outside air temperaturas

– Total thermal load.

In an area to be conditioned, the total thermal load results from the sum of the thermal loads involved in the cooling process.

$$Q_L = Q_e + Q_l + Q_p + Q_{T,1} + Q_{T,2} + Q_{\mathrm{I}} \quad (13)$$

3.3 Camera modeling

The modeling of the chamber allows predicting the behavior of the air temperature inside the enclosure without considering the cooling system. The change in the chamber temperature depends mainly on the climatic conditions of the site, from the thermal loads due to infiltration, and is obtained by the following equation:

$$\frac{dT}{dt} = \frac{U_1*A_1+U_2*A_2}{m_a*Cp_a}(T_e - T_i) \quad (14)$$

Where:
m_a is the mass of air
Cp_a is the specific heat of air.

Other important considerations for determining the temperature inside the chamber are the heat generated by the people entering the chamber and the lamps inside, which are determined with equations (6) and (7). In addition to these loads, the heat of the product must be considered, which can be estimated from equation:

$$Q_P = h * A * \Delta T \quad (15)$$

Where:
h is the convective coefficient of the product.
A is the average total area of the product.
ΔT is the difference in product temperatures, air temperature inside the chamber minus average product temperature.

The change in temperature of the product entering the chamber is obtained as:

$$\frac{dT_P}{dt} = \frac{Q_P}{m_P*Cp_P} \quad (16)$$

Where:
Q_P is the heat of the product.

The temperature of all the product inside the chamber, incoming product plus stored product, is obtained from the following equation:

$$T = \left(\frac{P_t-P_e}{P_t} * T_c\right) + \left(\frac{P_e}{P_t} * T_P\right) \quad (17)$$

Where:
P_t is the total product quantity.
P_e is the amount of product entering.
T_c is the temperature of the air inside the chamber.

T_P is the temperature of the entering product.

3.4 Analysis of the refrigeration cycle

The analysis of the refrigeration cycle allows to know the behavior of the system, considering a vapor compression cycle using R-290 as refrigerant (Webbook. n.d.). The thermodynamic states of the refrigeration cycle are shown in Figure 1, which are (Cengel, & Boles, M. 2001):

a) State 1, saturated steam at the refrigeration temperature.
b) b) State 2, superheated steam with constant entropy with respect to state 1.
c) State 3, saturated liquid at temperature higher than ambient.
d) State 4, mixture with constant enthalpy with respect to state 3.

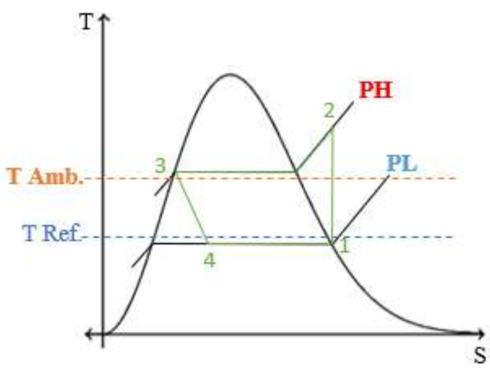

Figure 1 Diagram of the simple compression refrigeration cycle
Source: Own Elaboration

For the realization of the modeling of the cycle parameters such as refrigerant flow, the heat rejected by the condenser, the compressor work and the Coefficient of Performance (COP) are required.

The mass flow of refrigerant acts as a means of transporting heat from one point to another, its function is to absorb heat from the heat loads through the evaporator to subsequently discard this heat to the environment through the condenser, after passing through the compression stage. The mass flow is obtained as:

$$m_R = \frac{Q_L}{h_1 - h_4} \tag{18}$$

Where:
h_1 is the enthalpy at the evaporator outlet
h_4 is the enthalpy at the evaporator inlet

The heat discharged by the condenser is obtained as:

$$Q_H = m_R \ (h_2 - h_3) \tag{19}$$

Where:
h_2 is the enthalpy at the compressor outlet.
h_3 is the enthalpy at the condenser outlet.

The work required by the compressor to increase the pressure of the refrigerant fluid is obtained from the following equation:

$$W = m_R \,(h_2 - h_1) \tag{20}$$

If the thermodynamic states lie within the mixing region in the T-S diagram, the quality X, which varies from 0 to 1, must be determined. The calculation of the quality is obtained from the following equation:

$$X = \frac{h_g}{h_f + h_g} \tag{21}$$

And the enthalpy in the mixed state (h_m), is obtained from the following equations:

$$h_m = \mathrm{X}\,(h_g - h_f) + h_f \tag{22}$$

Where:
h_g is the enthalpy of saturated vapor
h_f is the enthalpy of saturated liquid

The coefficient of performance (COP) for a conventional refrigeration system is obtained from the following equation:

$$COP = \frac{Q_L}{W} \tag{23}$$

Where:
Q_L is the heat or load to be cooled
W is the compressor work

To determine the electrical consumption of the refrigeration system, the efficiency of the compressor must be considered, in this case a hermetic reciprocating compressor suitable for the refrigerant R-290 was selected, having an efficiency that is generally between 75% to 90%. For this analysis an efficiency of 85% is considered, with an operation of 14 hours/day. The electrical energy consumed throughout the day by the compressor is obtained from the following equation:

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$$EE_C = \frac{w*N}{\eta_c} \quad (24)$$

Where:
W is the work required by the compressor
N is the operating time/day
η_c is the efficiency of the compressor

3.5 Integration of the cold room with the refrigeration cycle

Once the calculation of thermal loads, the modeling of the cold room and the refrigeration cycle have been completed, the simulator is integrated. The structure of the simulator design is shown in Figure 2, where the input variables are the dimensions of the chamber, the climatic conditions of the place, the characteristics of the product, the number of people entering and the type of lamps used for lighting the room.

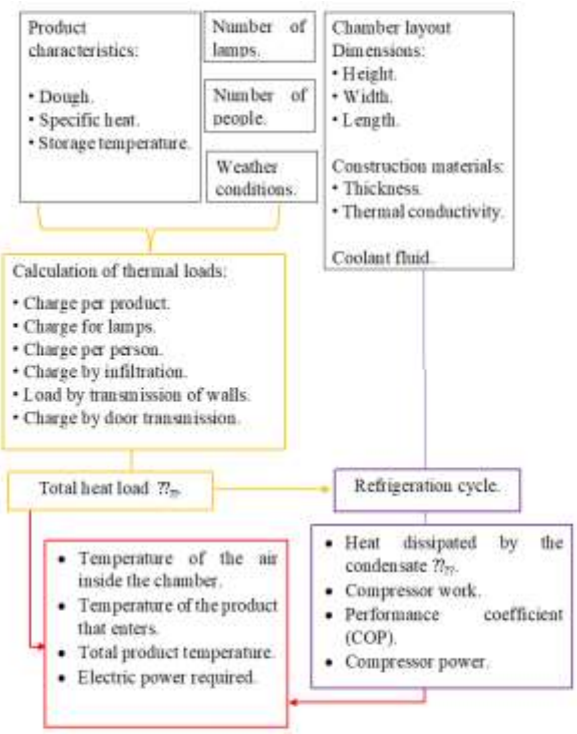


Figure 2 Simulator implementation
Source: Own Elaboration

Subsequently, the simulator calculates the thermal loads, the compressor work and the coefficient of performance (COP) of the system. Additionally, the simulator shows as outputs the temperature of the product entering the chamber, the temperature of the product remaining in the chamber and the electrical energy required.

3.6 Simulator evaluation

For the evaluation of the simulator, a case study must be proposed. In this case, it is proposed to evaluate the preservation of poultry meat, taking into account that the preservation temperature of poultry meat ranges from -1° C to 4° C, with a specific heat of $3.31\frac{kJ}{kg°C}$, and a shelf life of between 8 and 10 days.

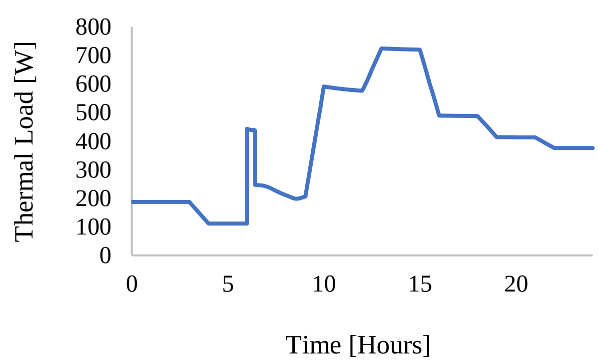
Based on the size of the chamber defined above, which can vary according to needs, a maximum capacity of 500 carcasses was considered. Considering that the storage facility will be located in the municipality of Apan Hidalgo, the climatic conditions of the region are used; atmospheric pressure of 101,325 kPa, with a maximum temperature of 35° C and a minimum of 3° C.

4. Results

The results obtained from the methodology described above are as follows:

4.1 Simulation of thermal loads

Graph 1 shows the behavior of the total thermal loads throughout an average day, where a maximum load of 724 W at 12:30 a.m. and a minimum of 111 W at approximately 5:00 a.m. is observed.



Graph 1 Total thermal load
Source: Own Elaboration

Graph 2 shows the behavior of the thermal loads separately.

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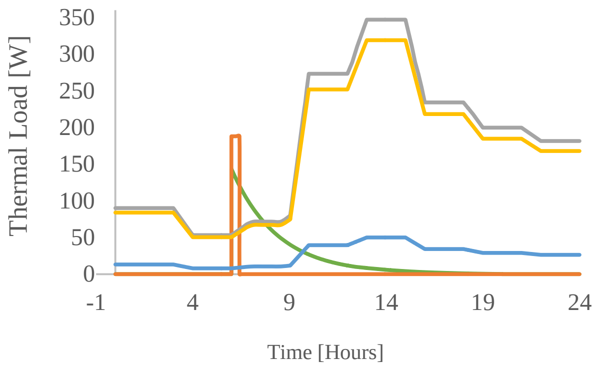


Graph 2 Thermal Loads
Source: Own Elaboration

The green line represents the thermal load of the product, which enters the chamber at 6:00 hours with a pre-cooling temperature of 10° C, with a cooling time of 8.5 hours and a maximum thermal load of 150 W. The orange line represents the thermal load of the people and lamps.

The orange line represents the thermal load of people and lamps, it is considered that people enter at 6:00 hours and stay 25 minutes and the lamps are kept on during that time, obtaining a thermal load of 188 W.

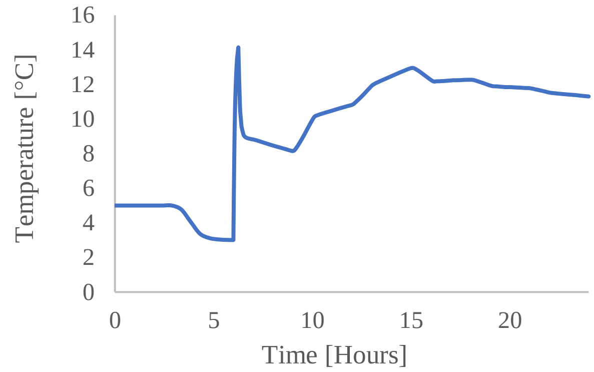
The gray line shows the thermal load due to infiltration, obtaining 53.3 W as minimum and 347 W maximum.

The yellow line shows the thermal load transmission through the walls, obtaining a minimum of 50.3 W and a maximum of 320 W. The blue line shows the thermal transmission load through the door, obtaining a minimum of 7.8 W and a maximum of 50 W.

4.2 Simulation of the interior of the chamber

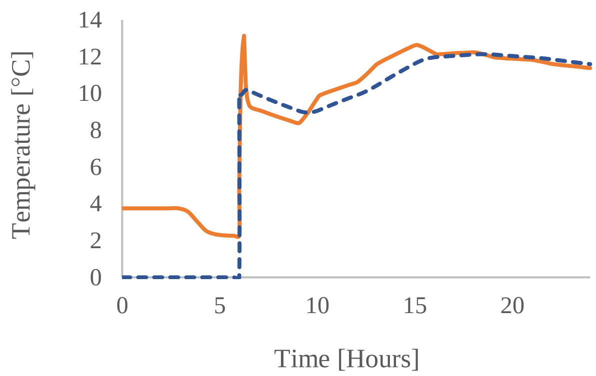
Graphs 3a and 3b show the behavior of the air and product temperature inside the chamber without refrigeration.

Graph 3a shows the variation of air temperature throughout an average day. A maximum temperature of 14°C is reached at 6:00 a.m., due to the entry of people and product, although this is for a short period of time. The temperature that has the greatest impact on thermal loads is the one obtained due to infiltration through walls, which occurs between 12:00 and 15:00 hours.



Graph 3a. Air temperature in the chamber
Source: Own Elaboration

Graph 3b shows the temperature of the product throughout the same average day. The solid line represents the variation in temperature of the total meat inside the chamber, which reaches a maximum temperature of 14° C at the time the product is received. The dotted line represents the temperature of the meat entering the chamber at 10° C, increasing its temperature until it stabilizes at approximately 16:00 hours.

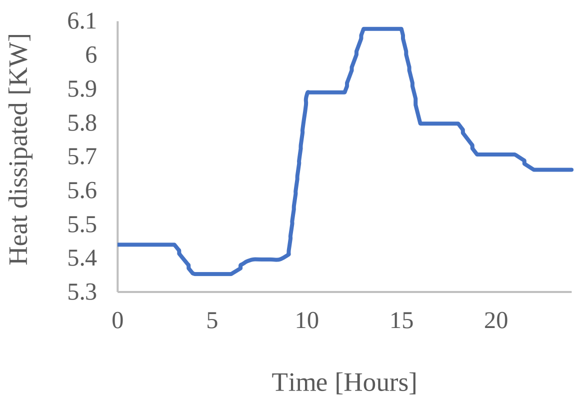


Graph 3b Product temperatura
Source: Own Elaboration

4.3 Refrigeration cycle simulation

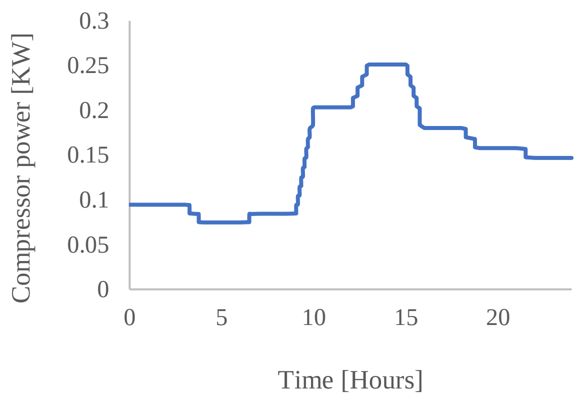
The refrigeration cycle works from the thermal loads involved in the system. Setting the product storage temperature at 0°C, with a refrigerant mass flow rate of $0.0065\ \frac{Kg}{s}$, the simulator calculates the work required to maintain the storage conditions. A maximum work of 448.1 W is obtained for the extreme conditions, with a coefficient of performance (COP) of 5.39.

Graph 4 shows the heat dissipated by the condensation process Q_H , which reaches a maximum value of 6077 W and a minimum of 5353 W.



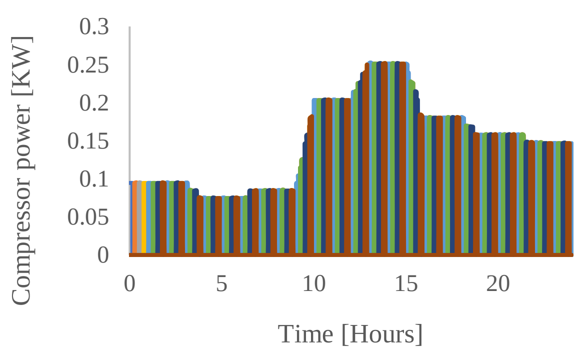
Graph 4 Heat dissipated by the condensation process.
Source: Own Elaboration

Graph 5 shows the work of the compressor during an average day, with a maximum work of 251.2 W at 15:00 hours, and a minimum of 74.77 W at 6:00 hours.



Graph 5. *Compressor work*
Source: Own Elaboration

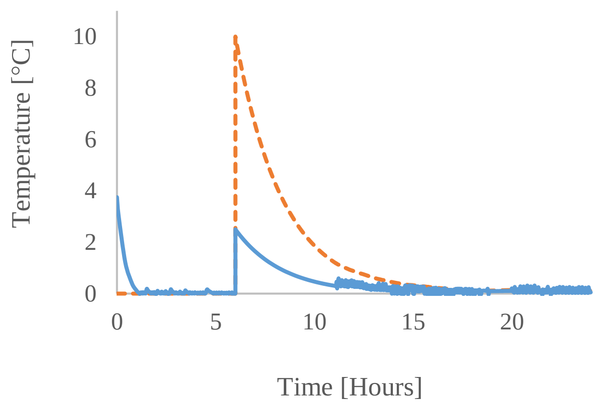
Graph 6 shows the behavior of the refrigeration system cycles in an average day, knowing that the compressor works 14 hours with 25 minutes for every 24 hours of operation, there is a total of 96 cycles in a day.



Graph 6 Cycles throughout the day
Source: Own Elaboration

4.5 Simulation of refrigerated product

The integration of the models of thermal loads, refrigeration cycle and temperature in the chamber, allows us to know the behavior of the temperature of the product to be refrigerated. Graph 7 shows the behavior of the temperature of the meat entering the chamber; the dotted line shows the behavior of the temperature of the product entering the chamber, where it can be observed that it reaches the refrigeration temperature in approximately 8.5 hours. The continuous line shows the behavior of the temperature of the total meat inside the chamber, where an increase in temperature is observed at the time of reception. This increase does not exceed the refrigeration range, which goes from 0° C to 4° C, remaining below 3° C.



Graph 7 Product temperature inside the chamber
Source: Own Elaboration

5. Conclusions

The simulator meets the expectations of evaluating the performance of cold rooms from their operating conditions throughout the day, although for this work an average day of February was considered, the simulator allows evaluating any day of the year, generating as results the behavior of the variables of interest; thermal load, product temperature, waste heat, compressor work, operating time and performance coefficient.

The simulator design considers variable inputs, such as the properties of the construction materials of the chamber, the dimensions of the chamber, and the climatic characteristics of the installation region. It also allows changing the type and quantity of product, the number of people entering the chamber and the lighting. These features make the simulator very flexible, allowing the evaluation of different types of chambers.

The evaluation of the simulator's performance was carried out for the case study referred to in this work, which describes the design of a cold storage chamber for the preservation of chicken installed in the municipality of Apan, Hidalgo. In this evaluation it was verified that the behavior of the conservation temperature shown by the simulator is within the operating range of a real chamber, so it can be said that the design of the proposed chamber is feasible from the thermodynamic point of view.

Regarding energy consumption, considering a hermetic reciprocating compressor with an efficiency of 85% and an operating time of 14.4 hours/day, a maximum power of 448.1 W and an average energy consumption of 3614 Whr/day were obtained, which indicates that the chamber design is also feasible from the energy point of view.

Currently it is intended that any energy system is sustainable, and a cold room is no exception, so it would be desirable that the simulator in addition to assessing the energy feasibility also verify whether the design of the chamber can be sustainable. Therefore, it is considered as future work to integrate a module to the simulator to determine the economic feasibility of generating the work of the compressor through solar energy.

It is worth mentioning that the purpose of this simulator is to evaluate the energy feasibility of a cold room design, using theoretical models obtained from general physical laws, so the results generated should be considered ideal. To improve the performance of the simulator, the input variables should be as close to reality as possible.

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