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CONTROL TEMPERATURE OF THE AIR CONDITIONING SYSTEM OF A VESSEL FROM EXERGOECONOMIC ANALYSIS

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Abstract

In this work the air conditioning system of a 93 m long and 14 m wide vessel with 4081 refrigerated m3 distributed in 51 premises is studied. In which energy, exergetic and exergoeconomic analyzes were carried out for control temperatures between 20 and 27 ° C and 50% relative humidity, with outdoor air conditions of 35 ° C and 70% relative humidity. For the vessel, the thermal load is calculated with an adaptation of the ASHRAE CLDT / SCL / CLF (cooling load temperature difference/cooling load factor/solar cooling load factor) methodology and the ISO 7547 standard. Thermal load contributors taken into account for the study were heat transfer through walls, ceilings, and glass in addition to gains from people, lighting, and Appliances. Transmission through walls and ceilings represents 33% of the thermal load, followed by glass with 18% and power equipment with 15%, the last three sources of thermal load generation are Appliances (12%), people (12%) and lighting (10%). For each degree centigrade of the control temperature, the thermal load is reduced by 2.4 and 1.1%, respectively, as determined by the ASHRAE and ISO methodologies. Similarly, the destruction of exergy is reduced by 4.16% for each degree Celsius that the control temperature is increased. An indicator is proposed to calculate the cost of generation of cooling load per unit volume and exergy of the thermal load from which it is obtained that the higher the control temperature, the lower the value of the cost of generation of the cooling load. From the exergoeconomic analysis, it is highlighted that the destruction of exergy is the main factor in the increase in system costs. Increases in exergy destruction increase the value of the indicator of cooling load generation costs per volume and unit of heat load exergy. For each degree Celsius that the control temperature is reduced, the cost of generation of cooling load per volume and unit of exergy of thermal load increases on average 0.35%.

Keywords: Thermal load, Air conditioning, Control temperature, Vessels, exergy, Thermoeconomic.

1. INTRODUCTION

Air conditioning systems are one of the great energy consumers, which is reflected in the high electrical energy requirements in buildings and boats. Therefore, it is important to design efficient and profitable systems that are friendly to the environment [1]. For this, it is essential to recognize unnecessary energy costs and understand the mechanisms that cause degradation, to improve systems and thus reduce environmental impact [2].

Air conditioning systems present drawbacks in the design, selection and / or use in buildings and vessels mainly due to inadequate control temperatures [3], oversizing or undersizing of the cooling capacity of the refrigeration machine and fan coil units, which they depend directly on the passive design and the calculation of the thermal load of the buildings and / or vessels [4] [5].

In buildings, different alternatives have been analyzed to reduce energy consumption by air conditioning systems, such as the implementation of renewable energies [3] [4], use of flow variation technologies in fans and pumps [5] [6] , changes of the construction materials or passive design measures [4] [10] and the appropriate selection of the control temperature [3] [11], the last two are the most important for the reduction of thermal load [12]. With the application of these measures, it would be possible to reduce the oversizing of air conditioning systems in ships. Additionally, the effects of changes in construction materials could be studied and as this work focuses, the appropriate selection of the control temperature.

Few studies have been carried out on the evaluation and increase of energy efficiency in boats as was done by Lugo & others [1], of which an energy diagnosis of a river boat is highlighted, from which it was concluded that: 79 % of the total electrical energy consumption of the ship is concentrated in three main services: refrigeration and air conditioning with 38%, lighting with 26% and cooking with 15% of the total consumption (Figure 1). In addition, inadequacies were shown such as: oversizing in the cold-water chiller units for the air conditioning system (20.6% higher than real capacity), in the fan coil type units (they can work at load factors of up to 15%) and the condensing unit installed for the refrigeration system (works at 46.5% of its capacity). Regarding the cold-water circulation system, it was detected that it consumes 22% of the electrical energy generated in said vessel. 64.5% of the losses generated in said system are located in pipes and accessories, while 22.2% are located in fan coils. Additionally, it was found that the cold-water circulation pump installed in said system is oversized and operating outside the best efficiency point.

For the calculation of the thermal load necessary for the selection of the capacity of the air conditioning systems in the design phases in ships, methodologies such as the one proposed by SNAME (Society of Naval Architects and Marine Engineers) [7] have been used, these are based on the use of traditional equations of heat transfer through walls. In the studies carried out by Hart, Fulton and Cox [8] and Fajardo & others [9] it was determined that the global heat transfer coefficient values established by SNAME in its Thermal Insulation Report are not adequate since they do not have in consideration of changes in configurations and modern insulating materials used in shipbuilding. In addition, they do not take into account the effects of heat storage, as recommended by the ASHRAE (American Society of Heating, Refrigerating and Air Conditioning) in their manuals. Likewise, for the determination of lighting loads, heat by people and by equipment, calculation equations are used that do not correspond [10]. Likewise, the ISO standard (International Organization for Standardization) 7547: 2002 has been implemented, it corresponds to the needs of vessels today [11], it still does not have corrections regarding the orientation and therefore all walls are assumed outside as sunny, which leads to oversizing of the air conditioning and refrigeration system. Despite this, there are not enough studies that allow the comparison, precision and flexibility of these air conditioning design standards in different types of boats.

The constructive changes regarding the configuration of walls and partitions in ships have been taken into account by Fajardo & others [12], they studied the performance of the air conditioning system of a river ship, from exergoeconomic indicators, using fiberglass, polyurethane or rock wool as thermal insulation on the walls, ceilings and floors of the boat. From which they concluded that: (i) as the insulation thickness increased, irreversibilities decreased, (ii) the increases in the exergy destroyed increased the costs of generating the cooling load and (iii) the costs of thermal load and insulation investment by unit area and unit cooling load were lower for polyurethane. The determination of the appropriate control temperature has been studied in a space-scale prototype of an air-conditioned vessel by Fajardo & others [9], where it was found that for the temperatures recommended by ASHRAE of 22 to 24 ° C [10], the highest exergetic efficiencies and lowest exergy destructions are obtained.

Sakulpipatsin & others, developed exergetic methods in the analysis of conditioned spaces in buildings and their air conditioning systems [13], it has been found that the emission of thermal energy, the control system and the energy conversion systems are the main causes of the exergetic inefficiencies in them. In addition, they revealed that the thermal load generated by solar radiation is the main exergy loss. Du et al., Through exergetic analysis, evaluated six control alternatives to improve the performance of the air conditioning system of a heated space and found that the best of them reduced the exergetic losses by 52% [14]. Goncalves & others [15] studied a hotel located in Coimbra, Portugal, in addition to energy-based indicators, two indicators were used: the primary energy ratio and the exergy efficiency. The results show a total consumption of 446kWh / m2 year, with 49 and 17% for the primary energy ratio and exergetic efficiency indicators. Electrical equipment is located as the main consumers of primary energy in the hotel; however, they present higher exergy efficiency compared to the processes related to the hotel's air conditioning system.

Exergy, exergoeconomic analysis and their indicators have been widely used to know irreversibilities, evaluate performance, costs and benefits in air-conditioned spaces. However, its use has not been extended in the evaluation of cooling and air conditioning systems in vessels. Therefore, in this study, the thermal load calculation adapted from ASHRAE, exergy and exergoeconomic analyzes will be applied to know the effect of the control temperature in the air-cooling system of a vessel.

2. MATERIALS AND METHODS

In this work the air conditioning system of a 93 m long and 14 m wide vessel with a refrigerated volume of 4081 m3 distributed in 51 rooms is studied. In which an energetic, exergy and exergoeconomic analysis was carried out for the control temperatures between 20 and 27 ° C.

FIGURE 1: VESSEL OPV SECOND GENERATION [16]

2.1 Thermal load

Thermal load is calculated for outdoor air conditions of 35 ° C and 70% relative humidity and control temperatures of 20 to 27 ° C and 50% relative humidity. Equations 1, 2 and Table 1 shows the equations for the calculation of thermal load with the ISO 7547 standard [11]. Equations 3 and 11 shows the equations for the calculation of thermal load for the vessel based on the CLDT / SCL / CLF methodology of ASHRAE [10].

$$\dot{Q}_{Walls} = UA_v\Delta T + A_v k\Delta T_e \quad (1)$$

$$\dot{Q}_{Window} = UA_g\Delta T + A_g G_s \quad (2)$$

Where $U, A_v, \Delta T, k, \Delta T_e, A_v, y\ G_s$ are Heat transfer coefficient, Surface, excluding hatches and windows, Difference in air temperature between air-conditioned and non-air-conditioned indoor spaces, Heat transfer coefficient for the deck and exterior bulkhead of the surface, Excess temperature (above the outside temperature) caused by solar radiation on the surfaces, Surface area of hatches and windows and Heat gain of glass surfaces respectively

Table 1: THERMAL LOAD OF PEOPLE AND LIGHTS [11]

Load	Equations and Variables	
People	Sitting at rest	$q_{sensible} = 70\ W$ $q_{latent} = 50\ W$
	Medium/ Heavy Work	$q_{sensible} = 85\ W$ $q_{latent} = 150\ W$
Lights	Cabins	$q_{Incandescent} = 15\ W/m^2$ $q_{Fluorescent} = 8\ W/m^2$
	Dinning	$q_{Incandescent} = 20\ W/m^2$ $q_{Fluorescent} = 10\ W/m^2$
	Gym	$q_{Incandescent} = 40\ W/m^2$ $q_{Fluorescent} = 20\ W/m^2$

$$\dot{Q}_{tran} = U * A * CLTD_c \quad (3)$$

$$\dot{Q}_{tran} = UA(t_b - t_{rc}) \quad (4)$$

$$CLTD_c = [(\mathrm{CLTD} + (25.5 - T_r) + (T_m - 29.4)] \quad (5)$$

$$\dot{Q}_{rad} = A * SC * SCL \quad (6)$$

$$SC = \frac{SHGC}{0{,}87} \quad (7)$$

$$\dot{Q}_{Lights} = (W)(F_{ul})(F_{sa})(CLF) \quad (8)$$

$$q_{sensible} = q_{Input} * F_U * F_R \quad (9)$$

$$q_{sensible} = N(SHG_P)(CLF_P) \quad (10)$$

$$q_{latent} = N(LHG_P) \quad (11)$$

Where $\dot{Q}_{tran}$, A, t_b, t_r, $CLTD$, $CLTD_c$, y T_m are Heat transfer loading through walls, ceilings and floors, Surface area, Adjacent space temperature, Design internal temperature in the air-conditioned space, Cooling load temperature difference, roof, wall, or glass, $CLTD$ corrected y Average outdoor temperature respectively.

2.2 Cooling load

The calculation of the cooling load is carried out from knowing the energy of the cold stream that is supplied by the fan coils, for this the suggestion given in the ISO 7547 standard of 2002 was followed "The temperature of the air supplied to the space does not it must be more than 10 ° C lower than the average temperature of the space" [11]. Based on the percentages of recirculated and outdoor air and the ratio of sensible heat to latent heat of the system. The specific amounts of cold air (Air_{sum}) necessary to achieve comfort conditions in each compartment were determined with Equation 12.

$$Air_{sum,k} = \frac{\dot{Q}_{sensible}}{1206 \times (T_{ret} - T_{Avg})} \times 3600 \quad (12)$$

Where $\dot{Q}_{sensible}$ is the sensible heat given by the thermal load of each compartment and T_{ret} and T_{Avg} are the supply and mean temperatures of the air within the conditioned space [12].

The total cooling load for each room $\dot{Q}_{cooling,k}$, was obtained by Equation 13.

$$\dot{Q}_{cooling,k} = \rho_{Air} \times Air_{sum} \times (h_{ret} - h_{sum}) \quad (13)$$

Where ρ_{Air} represents the air density (kg / m^3), h_{ret} and h_{sum} are the enthalpies of the return air and air supplies in the fan coils (kJ / kg), respectively.

2.3 Energy analysis

The consumption of fuel oil N°4 $\dot{m}_f$ required to generate the electrical power for the air conditioning system is shown in Equation 14. The required electrical power $\dot{W}_e$ (Equation 15) and the specific fuel consumption SCF (Equation 16). The power consumed was obtained from the energy consumption of the fan coils installed in each location, the total cooling load and the chiller's Energy Efficiency Ratio $EER_{Chiller}$. For SCF, a mathematical model was generated from the data reported in the technical specifications of the Caterpillar C32 Generator [17] that provides the electrical energy for the study vessel.

$$\dot{m}_f = \dot{W}_e SCF \quad (14)$$

$$\dot{W}_e = \frac{\dot{Q}_{cooling,total}}{EER_{Chiller}} + \sum \dot{W}_{Fan\ Coil,j} \quad (15)$$

$$SCF = 5.475\text{x}10^{-12}\dot{W}_e^4 - 1.1853\text{x}10^{-8}\dot{W}_e^3 + 9.1217\text{x}10^{-6}\dot{W}_e^2 - 2.965\text{x}10^{-3}\dot{W}_e + 0.6289 \quad (16)$$

To calculate the thermal load of the vessel, it is based on the ASHRAE CLDT / SCL / CLF methodology [10] adapted to a vessel by Fajardo & others [12], which takes into account the effects of heat storage, the different orientations with respect to the cardinal directions that the vessel can take. The thermal loads of transmission through walls, floors and ceilings, radiation and conduction in glass, for lighting, for electrical equipment and Appliances were determined.

The calculation of the cooling load is carried out from knowing the energy of the cold stream that is supplied by the fan coils, for this the suggestion given in the ISO 7547 standard of 2002 was followed *"The temperature of the air supplied to the space does not it must be more than 10 ° C lower than the average temperature of the space"* [11]. The volumetric flow of cold air (Air_{sum}) necessary to achieve the comfort conditions in each compartment were determined with Equation 17.

$$\dot{V}_{Air,sup,j} = \frac{\dot{Q}_{sensible,j}}{1206 \times (T_{ret} - T_{Avg})} \quad (17)$$

The total cooling load for each room $\dot{Q}_{cooling,k}$, is obtained by Equation 18.

$$\dot{Q}_{cooling,j} = \rho_a \times \dot{V}_{Air,sup,j} \times (h_{ret} - h_{sup}) \quad (18)$$

2.4 Exergy analysis

The exergy balance is obtained using Equation 19 [2].

$$\dot{X}_F = \dot{X}_P + \dot{X}_D \quad (19)$$

Where $\dot{X}_F$ is the exergy of the cooling and electrical load of the fan coil units of each room, $\dot{X}_P$ the sum of the exergies of the thermal load contributors and $\dot{X}_D$ the exergy destroyed.

The exergy destruction was determined with Equation 20, its components are described in Table 3 by Equations 21, 22, 23, 24 and 25 adapted by Fajardo & others [12] for a vessel.

$$\dot{X}_D = \dot{X}_{Cooling} + \dot{X}_{Elect} - \dot{X}_{tran} - \dot{X}_{gain} - \dot{X}_{rad} \quad (20)$$

$$\dot{X}_{Cooling,j} = \left|\dot{Q}_{cooling} * \left(1 - \frac{T_0}{T_i}\right)\right| \quad (21)$$

$$\dot{X}_{Elect,j} = \dot{W}_{Fan\ Coil}\ (kW) \quad (22)$$

$$\dot{X}_{tran,j} = \sum_{i=1}^{n}\left(\dot{Q}_{tran,k} * \left(1 - \frac{T_0}{T_{sup,k}}\right)\right) \quad (23)$$

$$\dot{X}_{gain,j} = \sum_{i=1}^{n}\left(\dot{Q}_{gain,k} * \left(1 - \frac{T_0}{T_{Source,k}}\right)\right) \quad (24)$$

$$\dot{X}_{rad,j} = \dot{Q}_{rad} * \left(1 - \frac{T_0}{T_{Sun}}\right) \quad (25)$$

Where $\dot{X}_{Cooling,j}$, $\dot{X}_{Elect,j}$, $\dot{X}_{tran,j}$, $\dot{X}_{gain,j}$ and $\dot{X}_{rad,j}$ are Cooling load, Power fan coil, Transmission through walls roof and glass, People, Appliances, power and lights and Radiation. $T_{Sun} = 6000\ K$ [13]

The exergetic efficiency ε is obtained from Equation 26.

$$\varepsilon = \frac{\dot{x}_P}{\dot{x}_F} \times 100 \quad (26)$$

The exergy destruction ratio y_D is calculated with Equation 27.

$$y_D = \frac{\dot{x}_D}{\dot{x}_F} \times 100 \quad (27)$$

2.5 Exergoeconomic analysis

The cost balance was determined as expressed in Equation 28, where $\dot{C}_{P,k}$ represents the total rate of costs associated with satisfaction of the thermal load, $\dot{C}_{F,k}$the total rate of associated costs to the generation of cooling load, $\dot{Z}_k$ is the total rate of non-exegetical total costs, that is, the investment costs of equipment, and the rate of operation and maintenance costs [18].

$$\dot{C}_{P,k} = \dot{C}_{F,k} + \dot{Z}_k \quad (28)$$

The components of the cost balance are presented in Table 2 (Equations 29, 30, and 31).

TABLE 2: COMPONENTS OF THE COST BALANCE

Description	Equation	
Total rate of costs associated with satisfying the thermal load for each configuration	$\dot{C}_{P,k} = c_{P,k}\dot{X}_P$ (\$/s)	(29)
Total rate of costs associated with generating cooling load for each configuration	$\dot{C}_{F,k} = c_{F,k}\dot{X}_F$ (\$/s)	(30)
Total non-exergetic costs	$\dot{Z}_k = \frac{PEC_k\varphi\left[\frac{i_r(1+i_r)^{ny}}{(1+i_r)^{ny}-1}\right]}{3600(RTY)}$ (\$/s)	(31)

The exergoeconomic indicators used in the study were: The exergy destroyed costs, the exergoeconomic factor [18] and the cost of generating cooling load per unit volume and exergy of thermal load [12].

The *costs of exergy destroyed* were obtained from Equation 32, considering that the costs of the product were fixed.

$$\dot{C}_D = c_{F,k}\dot{X}_{D,k} \tag{32}$$

The *exergoeconomic factor* is the ratio of the contribution of non-exergetic costs to the increase in total cost, for the case studied compares the capital costs in investment and operation and maintenance with the costs of irreversibilities (Equation 33).

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{D,k}} \tag{33}$$

The cooling load generation cost per unit volume and thermal load exergy is also calculated, which was modified with respect to the one proposed by Fajardo [12] (Equation 34).

$$\frac{cooling\ load\ cost}{Volume \times total\ thermal\ load\ exergy} = \frac{\dot{C}_F}{(\dot{X}_{P,tot})(V)}\left(\frac{\$}{GJm^3}\right) \tag{34}$$

With the combination of these indicators, the effect on the cost of generating the cooling load was determined by varying the control temperatures.

3. RESULTS AND DISCUSSION

3.1 Energy analysis

For the calculation of thermal load following ASHRAE's CLDT / SCL / CLF methodology and ISO 7547 standard, the following considerations were taken into account:

- Design exterior temperature = 35 ° C and Design exterior humidity = 70%.
- design Interior temperature = control temperatures vary from 20 to 27 ° C and Interior design humidity = 50%.
- Variation in daily outdoor temperature: 8ºC
- Design Month: July
- South orientation
- Time: 13.00

The total thermal load for both thermal load calculation methodologies is presented in Figure 2, where it is highlighted that the thermal load obtained with the ASHRAE methodology is higher on average 24487 W for all control temperatures. This may be because the ISO methodology contemplates fixed temperature changes, standard solar incidence heat gains and ignores the thermal load for temporary electrical appliances (Appliances). While the ASHRAE methodology takes into account the effects of heat storage and the different orientations with respect to the cardinal points that the course of a boat can takes in addition to different corrections for materials and color. Therefore, the total thermal load obtained with ISO methodology can become underestimated and therefore incur a bad selection of the cooling system, unless broad safety factors are used. From this, the following analyzes will start from the thermal load obtained with the ASHRAE methodology, for which for each degree centigrade that the control temperature is increased, the thermal load decreases 1170.27 W (equivalent to 0.94%), of which the 80% are due to transmission through walls and ceilings and 20% through glass.

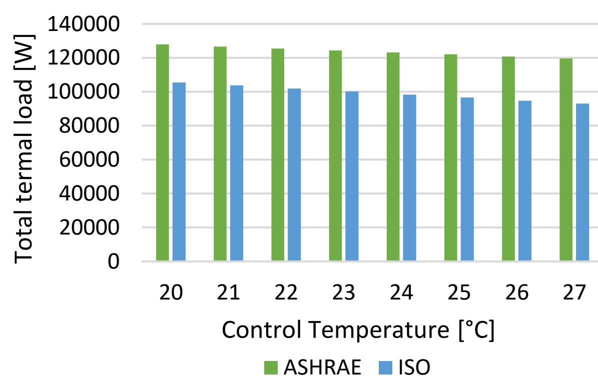

FIGURE 2: THERMAL LOAD FOR DIFFERENT CONTROL TEMPERATURES

Figure 3 shows the relationship of the contribution of the thermal load in the vessel. Transmission through walls and ceilings represents 33% of the thermal load, followed by glass with 18% and power equipment with 15%, the last three sources of thermal load generation are Appliances (12%) , people (12%) and lighting (10%).

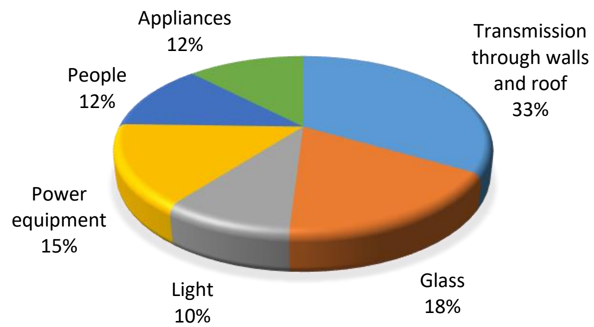

FIGURE 3: RATIO OF THE CONTRIBUTION OF THE THERMAL LOAD ON VESSEL

Table 3 shows the types of thermal load for control temperatures between 20 to 27 ° C, where it can be seen that the only sources of thermal load that are affected by the selection of the control temperature are thermal loads due to transmission through walls and ceilings, and glass. By increasing the control temperature of one degree centigrade, the thermal loads are reduced by 2.4 and 1.1%, respectively. La carga térmica latente no presenta variación al seleccionar diferentes temperaturas de control dado que esta solo se genera por appliances and people. Las infiltraciones no se contemplan por ser una embarcación militar.

TABLE 3: TYPES OF THERMAL LOAD FOR DIFFERENT CONTROL TEMPERATURES

T [°C]	Transmission through walls and roof [W]	Glass [W]	Power equipment [W]	Appliances [W]	Lights [W]	People [W]
20	41382	21782	11722	15160	18636	15214
21	40449	21546	11722	15160	18636	15214
22	39516	21309	11722	15160	18636	15214
23	38583	21072	11722	15160	18636	15214
24	37649	20835	11722	15160	18636	15214
25	36716	20598	11722	15160	18636	15214
26	35783	20361	11722	15160	18636	15214
27	34849	20124	11722	15160	18636	15214

The cooling load is shown in Figure 4, where it can be seen that for each degree Celsius that the control temperature is increased, the cooling load decreases an average of 20400 W (4.84%). To maintain comfort, the air conditioning system needs equipment that can supply 154 (138.5) Ton ref, in addition to the 79 fan coil units for the 51 premises (24 of 1 Ton ref, 11 of 1.5 Ton ref, 15 of 2 Ton ref, 7 of 2.5 Ton ref and 22 of 3 Ton ref).

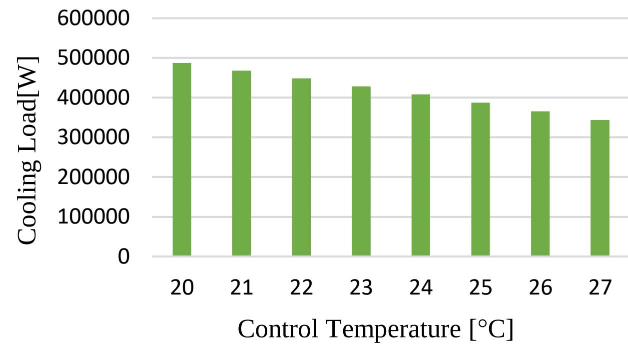

FIGURE 4: COOLING LOAD FOR DIFFERENT CONTROL TEMPERATURE

3.2 Exergy analysis

Table 4 shows the percentage change in the cooling load and power exergies of the fan coils ($\dot{X}_F$), the thermal load exergies ($\dot{X}_P$), the exergy destruction ($\dot{X}_D$) that accounts the degradation of energy and the exergy efficiency ε of the air conditioning system of the boat for different control temperatures, with respect to 23 ° C, where the values of 228058.81 W, 32416.78 W, 195642.04 W and 14.21%, respectively, were obtained. The positive and negative values indicate increases and decreases in the exergies of fuel, product, destroyed and exergetic efficiency. Increases for the values of $\dot{X}_F$, $\dot{X}_P$ and $\dot{X}_D$ are presented for control temperatures below 23 ° C, which causes a decrease in the exergetic efficiency of the air conditioning system, while for those above 23 ° C an increase in exergetic efficiency is obtained due to the decrease in the exergies of fuel, product and destroyed. This is consistent with the results obtained by Fajardo and others in a study of a room-scale air conditioning system on a vessel [9].

Table 4: EXERGY ANALYSIS FOR DIFFERENT CONTROL TEMPERATURES.

T [°C]	$\dot{X}_F$ [%]	$\dot{X}_P$ [%]	$\dot{X}_D$ [%]	ε [%]
20	13.29	2.22	15.12	-9.77
21	8.47	1.48	9.63	-6.44
22	5.04	0.74	5.75	-4.09
23	0.00	0.00	0.00	0.00
24	-5.31	-0.74	-6.07	4.83
25	-9.56	-1.48	-10.90	8.93
26	-13.42	-2.22	-15.27	12.93
27	-20.20	-2.96	-23.06	21.61

In the exergy analysis for the different control temperatures, the exergy destruction ratio (Equation 27) was determined from the exergy destroyed and the cooling load exergy. The results are presented in Figure 5 and show that at a higher control temperature, irreversibilities decrease. Esto se debe a la reducción de la carga térmica y por tanto una carga de enfriamiento cada vez menor, lo que ocasiona que el proceso de mantener el confort en la embarcación degrade menos energía.

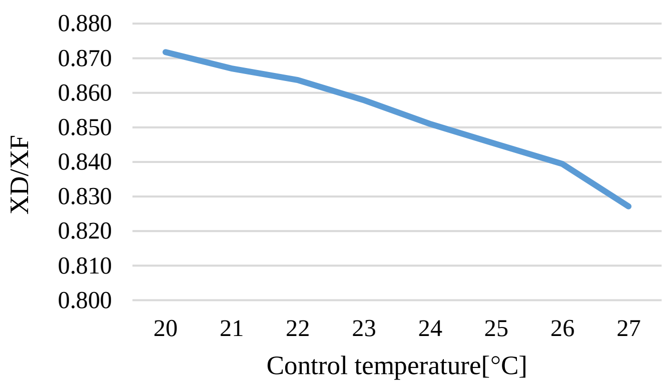

Figure 5: EXERGY DESTRUCTION RATIO FOR DIFFERENT CONTROL TEMPERATURES.

3.3 Exergoeconomic análisis

The values of the parameters used to develop the exergoeconomic analysis and indicators are presented in Table 5. The equipment acquisition costs include the refrigeration machines and the fan coil units necessary to satisfy the comfort needs of the boat. The cooling load generation costs were calculated from the equivalent costs of the fuel oil necessary to generate the electricity consumed by the cooling machine and the fan coils.

TABLE 5. PARAMETER VALUES FOR EXERGOECONOMIC ANALYSIS.

Parameter	Value	Reference
i_r (%)	7	[19]
n_y (years)	20	[12]
RTY (Hours)	5376	[12]
PEC ($)	1216192.71	-
Installation Factor (- PEC)	1.12	[20]
Maintenance factor (-PEC)	1.06	[21]
Marine diesel ($/Gal)	1.23	[22]

The exergy flows, cost flows and exergy unit costs associated with the control temperatures were calculated from Equations 29. and 30 and they are shown in Table 6. The exergy cost parameters of the vessel for the different control temperatures are shown in Table 7. These parameters indicate the performance of the air conditioning system on an exergy basis for the different comfort temperatures.

TABLE 6: EXERGY AND COST OF COOLING LOAD AND THERMAL LOAD.

T [°C]	$\dot{X}_F$ [kW]	$\dot{C}_F$ [$/h]	$\dot{X}_P$ [kW]	$\dot{C}_P$ [$/h]
20	**258.36**	28.95	33.14	51.01
21	247.37	28.66	32.90	50.60
22	239.54	28.36	32.66	50.31
23	228.06	28.06	32.42	50.01
24	215.95	27.76	32.18	49.72
25	206.26	27.45	31.94	49.42
26	197.46	27.14	31.70	49.11
27	181.98	26.83	31.46	48.81

The exergy costs and the highest thermal load exergies are presented when the lowest control temperatures are selected as shown in Table 6, this is due to the increase in the thermal load and cooling load exergies, even though at higher control temperatures, the unit exergy cost of fuel and product are higher, as shown in Table 7. Higher control temperatures present a higher exergoeconomic factor, this because they present a cost of destruction of exergy, mainly due to the reduction of the destruction of exergy since the unit exergetic fuel cost increases at high control temperatures. El incremento del costo de fuel por unidad de exergía se debe a que a mayor temperatura de control el SCF. Lo que equivale a un funcionamiento menos eficiente por el sobredimensionamiento de los equipos.

TABLE 7: EXERGOECONOMIC ANALYSIS FOR DIFFERENT CONTROL TEMPERATURES.

T [°C]	c_F [$/MWh]	c_P [$/MWh]	$\dot{Z}$ [$/h]	$\dot{C}_D$ [$/h]	f [−]
20	0.0311	0.4217	21.354	25.239	0.458
21	0.0322	0.4223	21.354	24.847	0.462
22	0.0329	0.4229	21.354	24.495	0.466
23	0.0342	0.4234	21.354	24.072	0.470
24	0.0357	0.4240	21.354	23.621	0.475
25	0.0370	0.4245	21.354	23.200	0.479
26	0.0382	0.4250	21.354	22.783	0.484
27	0.0409	0.4254	21.354	22.188	0.490

Figure 6 shows the results of the indicator proposed by the authors for the evaluation of the air conditioning system for the different comfort temperatures, the cost of generation of cooling load per volume and unit of exergy of thermal load. In which it can be seen that for study control temperatures this presents an inverse relationship since the higher control temperature presents the lowest costs. For each degree Celsius that the control temperature is reduced, the cost of generation of cooling load per volume and unit of exergy of thermal load increases on average 0.35%. Por tanto, seleccionar la temperatura de control más alta posible manteniendo los estándares de confort presenta una mejor degradación de la energía y un mejor funcionamiento del sistema.

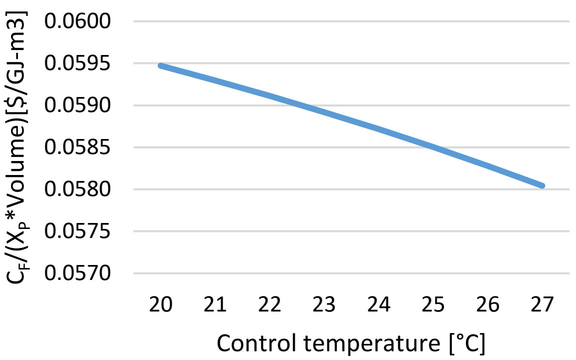

FIGURE 6: COOLING LOAD GENERATION COST PER UNIT VOLUME AND HEAT LOAD EXERGY FOR THE DIFFERENT CONTROL TEMPERATURES.

4. CONCLUSION

In this study, an exergoeconomic analysis was applied to the air conditioning system of a vessel to determine its exergy and economic performance when using different control temperatures. Some final observations obtained from the results of this study may be the following:

a) For the vessel, the major contributors of thermal load were the transfer of heat through walls and ceilings, and glass, which represent 33 and 18%, respectively. Which by an increase of one degree centigrade in the control temperature the thermal load is reduced by 2.4 and 1.1%, respectively.

b) Exergy destruction decreases when selecting a higher comfort temperature. For every degree Celsius that the comfort temperature is increased, exergy destruction is reduced by 4.16%.

c) Increases in exergy destruction increase the value of the indicator of cooling load generation costs per volume and unit of thermal load exergy. For the control temperature of 20 ° C the average generation cost is $ 0.0595 / GJ-m3 while for 27 ° C it is $ 0.0580 / GJ-m3.

d) Para mejorar la eficiencia del sistema es pertinente seleccionar la temperatura de control más alta posible, manteniendo los estándares de confort para los ocupantes. Igualmente se deben diseñar los sistemas partiendo de la temperatura de control seleccionada para reducir el sobredimensionamiento del sistema de acondicionamiento de aire.

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NOMENCLATURE

$\dot{C}$	Total Cost (USD/s)
c	Specific cost per unit of exergy (USD/KJ)
C_P	Specific Heat (KJ/kg°C)
$\dot{E}$	Exergy rate (KJ/s)
e	Specific exergy (KJ/kg)
h	Specific enthalpy (KJ/kg)
$\dot{m}$	Mass flow (kg/s)
P	Pressure (Kpa)
$\dot{Q}$	Heat transfer (kW)
R	Ideal gas constant (KJ/mol °K)
s	Specific Entropy (KJ/kg°K)
T	Temperature (°C)
$\dot{W}$	Work (kW)
Greek letters	
ε	Exergy efficiency
subscripts	
0	Condiciones del estado de referencia
D	Destruction
F	Fuel
k	*k*-th input
j	*k*-th local
P	Product
Abbreviations	
Avg	Average
CLDT	Cooling load temperatura difference
CLF	Cooling load factor
Elect	Electric
HR	Relative humidity
HVAC	Heating, Ventilation and Air Conditioning
SCL	Solar cooling factor
Sum	supplied
Ret	return
Trans	Transmission

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