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Research Article

3E Analysis of the Effect of Different Type of Fans on Cooling Performance Applied to an Industrial Deep Freezer

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ABSTRACT

In this study, two different industrial refrigerators were designed, manufactured and tested to analyze the impact of different types of fans (Type 1 and Type 2) used in industrial cooling systems on the performance of the cooling system. In order to test the fan performance and airflow effects, two axial fan configurations with different structures and different motor technology (EC and shaded-pole induction) were tested in two separate industrial refrigerator test rooms in accordance with TS EN ISO 23953-2 standards. R290 (Propane) was used as a refrigerant in the systems. The average temperature and relative humidity values of the environment where the experiment was conducted were measured as 25 °C and 60 % (Class 3), respectively. During the experiments, the total of 51.71 kWh energy was consumed in system 1, while the total of 54.22 kWh energy was consumed in system 2 and the difference between the energy consumption of the two systems was calculated as 4.85%. The average temperatures of the inlet and outlet of the evaporator of the system 1 and 2 were -21.57 °C, -18.97 °C and -23.43 °C, -20.94 °C, respectively. The average refrigerant temperatures for the system 1 and 2 were calculated as -24.65 °C, -26.44 °C, respectively. While the average coefficient of performance value of the type 1 system was 1.74, it was calculated as 1.54 for the type 2 cooling system. The average second-law efficiencies for the two cooling systems were calculated as 30.85 % and 29.81 %, respectively. In addition, the environmental economy analysis was carried out using the amount of CO_2 that was prevented from emitting and the CO_2 emission price calculated accordingly.

Keywords: *Refrigerator, Cooling, Axial Fan, Energy, Energy Efficiency*

Endüstriyel Derin Dondurucuya Uygulanan Farklı Tip Fanların Soğutma Performansına Etkisinin 3E Analizi

ÖZ

Bu çalışmada, endüstriyel soğutma sistemlerinde kullanılan farklı tip fanların (Tip 1 ve Tip 2) soğutma sisteminin performansına etkisini analiz etmek için iki farklı endüstriyel buzdolabı tasarlanmış, üretilmiş ve test edilmiştir. Fan performansını ve hava akış etkilerini test etmek için farklı yapılarda ve farklı motor

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teknolojilerine (EC ve gölge kutuplu indüksiyon) sahip iki adet aksiyal fan konfigürasyonu, TS EN ISO 23953-2 standartlarına göre iki ayrı endüstriyel buzdolabı test odasında test edilmiştir. Sistemlerde soğutucu akışkan olarak R290 (Propan) kullanılmıştır. Deneyin yapıldığı ortamın ortalama sıcaklık ve bağıl nem değerleri sırasıyla 25 ° C ve % 60 (Sınıf 3) olarak ölçülmüştür. Deneyler sırasında 1. sistemde toplam 51.71 kWh enerji tüketilirken 2. sistemde toplam 54.22 kWh enerji tüketilmiş ve iki sistemin enerji tüketimi arasındaki farkı % 4.85 olarak hesaplanmıştır. Sistem 1 ve 2'nin buharlaştırıcısının giriş ve çıkışının ortalama sıcaklıkları sırasıyla -21.57 °C, -18.97 °C ve -23.43 °C, -20.94 °C'dir.

Sistem 1 ve 2 için ortalama soğutucu akışkan sıcaklıkları sırasıyla -24.65 °C ve -26.44 °C olarak hesaplanmıştır. Tip 1 sistemin ortalama performans katsayısı değeri 1.74 iken, tip 2 soğutma sistemi için 1.54 olarak hesaplanmıştır. İki soğutma sistemi için ortalama ikinci kanun verimleri sırasıyla % 30.85 ve % 29.81 olarak hesaplanmıştır. Ayrıca emisyonu önlenen CO_2 miktarı ile çevre ekonomisi analizi yapılmış ve buna göre CO_2 emisyon fiyatı hesaplanmıştır.

Anahtar Kelimeler: *Buzdolabı, Soğutma, Aksiyal Fan, Enerji, Enerji Verimliliği*

I. INTRODUCTION

When it comes to global energy challenges, one of the essential things is to enhance energy efficiency considering energy saving [1]. Energy efficiency program has played a key role in reducing energy consumption in several countries. One of the most widely used households and industrial appliances in both developed and developing countries is the refrigerator [2]. With the improvement of expectations for everyday comforts, food quality and conservation issues have attracted increasing attention; therefore, the requirements of the refrigerator also increase rapidly. Industrial refrigerators mainly work cycling the cooled air within the refrigerator while the air is driven by the ventilation system. The fan directs the air through the evaporator and cools it. At that point, the cooled air is driven into the cooler box. Hence, the performance parameters of the system play a significant role for the cooling speed, cooling efficiency and total energy consumption of the refrigerator. High performance and increased efficiency can be carried out by the help of improved flow characteristics of the fans [3].

In industrial refrigerators, the air of the fresh and frozen food sections are cooled by evaporator to approximately +5 ºC and -18 ºC, respectively. Due to the migration of moisture from the products, the air in the room tends to have a high relative humidity mainly caused by door openings. Since the relative humidity of the internal air accumulates on the cold evaporator surface (usually below -18 ºC) the evaporator absorbs the moisture over time [4]. Most refrigerators in operation include fan-fed evaporator and condenser as heat exchangers, a capillary tube and a hermetic compressor. An internal heat exchanger and pre-condenser can also be included as additional parts. In this application, the evaporator usually runs at freezing temperature. Due to two simultaneous effects of the frost layer, which causes low thermal conductivity and reduction of air flow channels, the frosting conditions will reduce the cooling capacity which causes a longer compressor cycle requiring higher energy input [5].

In refrigerators, the evaporator and condenser fans are driven by electric motors, and only the highest power consumption is achieved after the compressor. The most common and low-cost motor type used for this purpose is the shaded pole induction motor. The fan structure and motor technology used in the evaporator and condenser enable the system to cool down and significantly reduce the waste heat generated. Reduced waste heat in the evaporator fan typically results in less operation of the compressor, which saves more energy. As a result, the fan structure and technology used in the evaporator and condenser are crucial, especially in terms of the energy efficiency of the cooling system. [6]

Several studies are conducted on the literature to achieve a higher freezing efficiency and reduce the energy consumption of the cooling system. It can be seen that only a few studies have been performed on the impact of different fans used in the evaporator and condenser on the entire cooling system. Similarly, it has been observed that few of these studies include experimental results, and most studies on energy efficiency only include expectations based on the efficiency of the motors used. Pioneering studies on the literature are evaluated within the scope of this topic and summarized below. Dong et al. studied an azeotropic mixture of R744/R170 (0.78/0.22, mass fraction) as a refrigerant instead of R134a for an air source heat pump water heater (ASHPWH) system. Exergy efficiency was analyzed for the system performance using both refrigerants based on the first and second law of thermodynamics. It was found that coefficient of heating performance (COP) and exergy efficiency of R744/R170 refrigerant were better than R134a and led to an incerase by 31.3 % and 30.6 %, respectively. In addition, 34.3–49.8 °C lower discharge temperature and reduced compression ratio (from 12.0417 to 3.2734) were obtained for R744/R170. It was also stated that R744/R170 mixture could be a good alternative to R134a in ASHPWH systems [7].

Acharya et al. investigated the impact of the condenser fan speed on vapor compression refrigeration system and found that as the CFM speed increases, the pressure head decreases and higher speed also contributes to low heat loss on the system. Cooling of the space is the critical parameter that causes higher CFM to cool at a faster rate [8]. Elsayed et al. analyzed the performance of split air conditioner driven by an inverter to obtain the effect of compressor speed, condenser fan speed, evaporator fan speed and electronic expansion valve. It is found that a 7 °C increase in inlet air temperature of condenser led the power consumption of the system increased by 18% and COP decreased by 15% [9]. Yang et al. used condensate water to advance the performance of air conditioner by reducing the airflow temperature around the condenser. Results shows that increased environment temperature led to the decrease of the refrigerating capacity and energy efficiency ratio by increasing power consumption [10]. Thakre et al. used R290 as a refrigerant to evaluate the energy efficiency of a variable speed dc compressor and result shows that system consumed 162.45 W in variable speed operation and 210.87 W in maximum speed operation while variable speed operation had lower cooling capacity in comparison to the maximum speed operation [11]. Hermosa et al. designed a controller to improve energy efficiency of household refrigerator driven by inverter and compared the performance of conventional and inverter-driven technology. It was found that the proposed controller and variable speed operation exhibited a reduction of 27 % in the power consumption [12].

Colorado and Rivera compared the performance of double stage, compression absorption single stage and conventional vapor compression systems for refrigeration. The results indicated that 45 % lower compression power was obtained with the cascade cycles [13]. Panigrahi and Mishra studied an appropriate aerofoil blade profile using computational fluid dynamics (CFD) for the fan blades to improve the efficiency of axial flow mine ventilation fans by varying angles of attack and various aerody-namic parameters. The study revealed that an appropriate aerofoil blade profile will increase energy efficiency of mine ventilation fans [14]. Zhao et al. investigated modification of fan geometry with the refined numerical and experimental approaches to reduce noise of outdoor unit. CFD and computational Aerodynamic Acoustics (CAA) was used to predict the noise behavior and it was found that modified fan geometries are effective to reduce the noise but the flanging outer-edge blade is more effective [15]. Dang et al. analyzed the thermal performance of a S-NDWCT equipped with an axial fan at different fan diameters and fan power to evaluate the effect of fan. Results showed that when the diameter of the fan exceeds 15.0 m, the air velocity in the central area of the tower increases significantly and the temperature distribution becomes more uniform [16]. Amin and Zulkifli compared conventional (on-off) and the variable speed drive (VSD) control of evaporator fan motor of split AC to reduce cooling time and electricity consumption. It was conducted that VSD control could increase the COP by 32 % compared to on-off control and reduce energy consumption up to 11% [17]. Angelini et al. presented a pitch and chord optimization study for increasing the efficiency and decreasing the trailing edge noise of an axial flow fan for air-cooled condensers and noise reduction of -0.5 dB achieved [18].

Effect of different type of fans on cooling performance applied to an industrial deep freezer is a rather complex issue to obtain without experimental studies. Not only energy consumption but also significant parameters such, inlet and outlet temperatures of the evaporator, refrigerant temperatures, coefficient of performance, second-law efficiencies should be taken into account in terms of optimum fan selection. This study aimed to investigate the effect of different fans on the performance of industrial deep freezer and to carry out the importance of proper fan selection based on conducted tests under TS EN ISO 23953-2 standards.

If the evaporator draws heat from the air and discharges heat to the air in the condenser, the airflow significantly affects the system performance and the thermal characterization of the cooling system in vapor compression refrigeration systems. Besides, energy consumption of the fan used in industrial refrigerators is also important in terms of energy efficiency. Industrial refrigerators are subjected to energy efficiency labelling according to their energy efficiency index. The effect of fan selection on the energy efficiency index value investigated within the scope of this study. In this study, fan selection has been realized by calculating the pressure losses in the system and the airflow rates of the fan were calculated considering the evaporator and condenser capacities. In this sense, this study is focused on investigating the effect of two different fan types on the coefficients of performance, total energy consumption and especially the effect of obtained airflow on the air temperatures in the industrial refrigerator. Results of the study are aimed to contribute into this scientific domain by investigating the effect of fan type and fan selection on industrial refrigerator systems. This study presents the investigation of the effect of two different types of fans used in an industrial plug-in vertical freezer system on the performance of the cooling system using R290 refrigerant. This study is organized as follows: material and method are given in the second section. Results and discussion are discussed in the third section, including those on temperatures, defrost intervals, coefficient of performance and total energy consumption of the tested systems. Conclusions are given and discussed in the fourth section.

II. MATERIAL AND METHOD

In this study, it is aimed to compare and analyze two different fans in an industrial plug-in vertical freezer system. L1 type vertical, open, vapor compression industrial refrigeration system operating with R290 refrigerant is designed with reference to -25 ℃ evaporation and 45 ℃ condensation temperatures. The P-h diagram of the cooling system is given in Fig. 1. The experiment set is designed, manufactured and tested. Tests in the cooling system were carried out in the test room under Class-3 at 25 ℃ temperature and 60 % relative humidity (% RH) conditions.

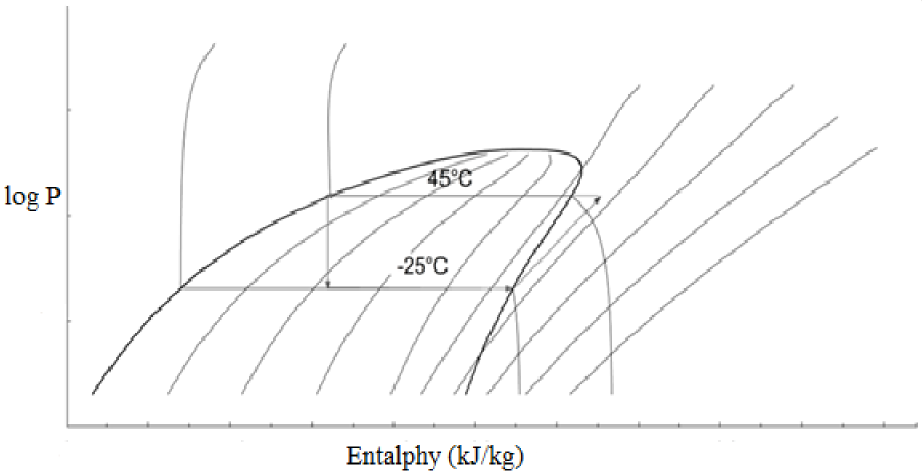
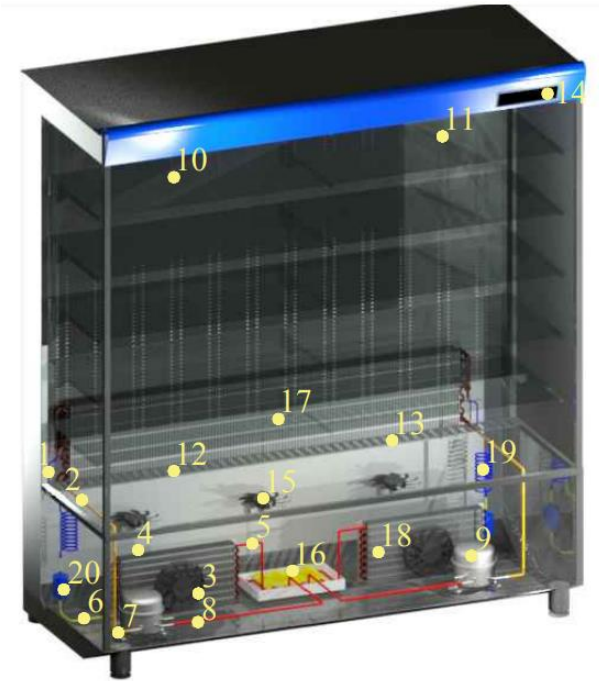


Figure 1. *The path followed by the cooling system in the P-h diagram*

The tests were performed equally and in accordance with the ISO 23953-2 standard using equipments with the same characteristics in both different fans. In this context, tests were carried out by recording the experimental data obtained using the fans whose characteristics are given in Table 1.

Table 1. *Technical characteristics of the fans tested in the experimental setup*

Fan	Number and model	Nominal voltage *(V)*	Output power range *(W)*	Speed range *(rpm)*	Air flow *(m^3/h)*	Operating temperature range *(℃)*	Fan Diameter *(mm)*
Axial Fan type 1	3 Electronically commutated motor	230	0-13	0-1800	200	-30/50	200
Axial Fan type 2	6 shaded pole induction motor	230	0-19	2650	160	-40/75	115



1	Evaporator inlet temperature value	**12**	Left side air off temperature value
2	Evaporator output temperature value	**13**	Right side air off temperature value
3	Condenser air inlet temperature value	**14**	Temperature and relative humidity sensor
4	Condenser air outlet temperature value	**15**	Fan
5	Condenser gas inlet temperature value	**16**	Defrost container
6	Condenser liquid output temperature value	**17**	Evaporator
7	Compressor suction temperature value	**18**	Condenser
8	Compressor discharge line temperature value	**19**	Capillary tube
9	Compressor body temperature value	**20**	Dryer
10	Left side air on temperature value	**HP**	High pressure value
11	Right side air on temperature value	**LP**	Low pressure value

Figure 2*. Deep freeze prototype and measuring points*

The experiments were carried out in the test room with calibrated test devices within the scope of TS EN ISO 23953-2 standard. Cooling performance of the system, energy consumption, temperature measurements, pressure measurements and energy efficiency tests were performed. While performing these tests, temperature measurements were taken every minute by means of thermocouples from the certain points of cooling systems equipment (compressor, condenser and evaporator inlet-outlet temperature values). Information on measurement equipments used in prototypes production and tests are given in Table 2.

Table 2. *Properties of measurement devices used in cooling systems and uncertainty values*

Device	Model	Unit	Measuring Range	Accuracy	Uncertainty
Thermocouple	Omega, CL-23A	°C	-40 / +150	± % 0.1 °C	± % 0.14 °C
Pressure transmitter	Eliwell, HP	bar	0-30	± 0.1	± 0.15
Pressure transmitter	Eliwell, LP	bar	0.5-8	± 0.01	± 0.02
Thermohygrometer	Rotronic, M23W2HT-1X	°C / %RH	0 / +50 / 0-100	± 0.03 / ± % 1.5	±0.06 °C ± 2.25 RH
Anemometer	E+E Electronic, EE66-VA3 EE660-V7	m/s	0-2	± 0.01	± 0.05
Digital scales	Value, VES-100B	g	0-100	± % 0.05	± % 0.065
Digital manifold	Testo, 550	bar	-50 / +150 -1 / +60	± 0.1 °C ± 0.01 bar	± 0.14 °C ± 0.02 bar
Energy analyzer	Janitza, UMG508	Ampere / Volt	-	Current ± % 0.2 / Voltage ± % 0.1	± % 0.25 Current ± % 0.14 Volt
Flowmeter	Siemens, SITRANS FC MASS 6000	kg/h	0-1000	± % 0.1	± % 0.2

III. THEORETICAL ANALYSIS

Evaporator capacity ($\dot{Q}_e$, kW) of the industrial cooler can be calculated by Eq. (1) [19]:

$$\dot{Q}_e = \dot{m}_r \,.\, (h_1 - h_4) \tag{1}$$

where $\dot{m}_r$ is refrigerant mass flow rate (kg/s) and h is enthalpy (kJ/kg). The compressor power ($\dot{W}_c$, kW) that will meet the cooling load in the industrial cooler and create the desired pressure difference in the cooling system can be determined by Eq. (2). In the design, the compressor power is determined experimentally with the energy analyzer:

$$\dot{W}_c = \dot{m}_r \,.\, (h_2 - h_1) = I.V.cos\varphi \tag{2}$$

where I is phase current (A), V is the voltage (V) applied to motor terminal and $cos\varphi$ is power factor. Cooling coefficient of the performance (COP) of the industrial refrigeration cycle is calculated by Eq. (3) [19,20]:

$$COP = \frac{\dot{Q}_e}{\dot{W}_c} \tag{3}$$

Reversible coefficient of performance of the industrial refrigeration cycle can be calculated Eq. (4):

$$COP_{car} = \frac{T_L}{T_H - T_L} \tag{4}$$

where T_H and T_L hot and cold sources temperatures (°C). The second law efficiency of the industrial cooling system can be found by the ratio of the cooling performance coefficient of the system to the Reversible performance coefficient under the same conditions. It can be defined as by following Eq. (5) [20]:

$$\eta = \frac{COP}{COP_{car}} \tag{5}$$

Fan power ($\dot{W}_f$, kW) to circulate the cool air blown through the evaporator is defined in Eq. (6) [21]:

$$\dot{W}_f = \frac{\dot{Q}\ .\ \Delta p}{\eta_f} \tag{6}$$

where Δp is air side pressure drop across an evaporator (Pa) and η_f is efficiency of fan (%). The logarithmic temperature difference (ΔT_{ln}) can be calculated with the Eq. (7) [21]:

$$\Delta T_{ln} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{7}$$

where ΔT_1 and ΔT_2temperature differences (°C). Mass flux (Γ,kg/m^{2}s) is calculated with the Eq. (8):

$$\Gamma = \frac{\dot{m}_r}{A_c} \tag{8}$$

where A_c is the area of compressor (m^2). Eq. (9) used in determining the amount of heat extracted from the air per unit time with the evaporator is given below [21]:

$$\dot{Q}_e = U.A.[\Delta T_{ln}] \tag{9}$$

Exergy destruction ($\dot{E}_x$) formulas and calculation has been added. In Eq. (10), s and 0 refer to the entropy (kJ/kgK) and the dead state, respectively. Formulas defined as follows [20,21]:

$$\dot{E}_x = \dot{m}\,[h - h_0 - T_0(s - s_0)] \tag{10}$$

Exergy destruction on compressor is calculated by Eq. (11):

$$\dot{E}_{x,dest,comp} = \dot{m}\,[(h_1 - T_0 s_1)] - [(h_2 - T_0 s_2)] + \dot{W}_{comp} \tag{11}$$

Exergy destruction on condenser is calculated by Eq. (12):

$$\dot{E}_{x,dest,con} = \dot{m}\,[(h_2 - T_0 s_2)] - [(h_3 - T_0 s_3)] - \left[\dot{Q}_{con}\left(1 - \frac{T_0}{T_{con}}\right)\right] \tag{12}$$

Exergy destruction on evaporator is given by Eq. (13):

$$\dot{E}_{x,dest,eva} = \dot{m}\,[(h_1 - T_0 s_1)] - [(h_4 - T_0 s_4)] \text{-} \left[\dot{Q}_{con}\left(1 - \frac{T_0}{T_{eva}}\right)\right] \quad (13)$$

It is important to perform energy and exergy analysis in refrigerant systems. Therefore, the exergy balance for the steady flow control volume in this study is given Eq. (14) and Eq. (15) [20]:

$$\eta_{ex}= \frac{\dot{E}_{x,dest,eva}}{\dot{W}_{comp}} \quad (14)$$

$$\eta_{ex}= \frac{\dot{m}\,[(h_1 - T_0 s_1)] - [(h_4 - T_0 s_4)]}{\dot{m}(h_2 - h_1)} = \frac{[(h_1 - T_0 s_1)] - [(h_4 - T_0 s_4)]}{(h_2 - h_1)} \quad (15)$$

Eq. (16) used to determine the amount of carbon dioxide (kgCO_2/h) prevented from being produced per hour is given below [22].

$$\emptyset CO_2 = \dot{W}_c.\ \psi CO_2 \quad (16)$$

where ψCO_2 is the amount of CO_2 released when energy is produced from coal (kgCO_2/kWh). Eq. (17) used to determine the gain per CO_2 blocked from production per hour is given below [22].

$$Z_{CO_2} = P_{CO_2}.\emptyset CO_2 \quad (17)$$

where P_{CO_2} is the earnings per CO_2 production blocked (¢/kgCO_2). The formulas used in fan efficiency calculations are as follows:

Input power of the mono-phase AC fan motor P_{input}, kW) can be calculated by Eq. (18) [23]:

$$P_{input} = U_{line} \cdot I_{line} \cdot cos\varphi \quad (18)$$

Shaft power of the motor or the power required by impeller is calculated by Eq. (19) [23]:

$$P_{shaft} = \eta_{motor} \cdot U_{line} \cdot I_{line} \cdot cos\varphi \quad (19)$$

where: η_{motor} is the efficiency of the motor, U_{line} and I_{line} are supply voltage (V) and current (A) and $cos\varphi$ is power factor.

Fan power P_{fan} can be calculated by Eq. (20) [23]:

$$P_{fan} = k \cdot Q \cdot P \quad (20)$$

where: Q is airflow (m^3/sec), P is static pressure (Pa) and k is compressibility factor approximated as unit. Impeller efficiency η_{imp} can be calculated by Eq. (21) [23]:

$$\eta_{imp} = \frac{P_{fan}}{P_{shaft}} \quad (21)$$

Overall efficiency of the fan η_{fan} can be calculated by Eq. (22) [23]:

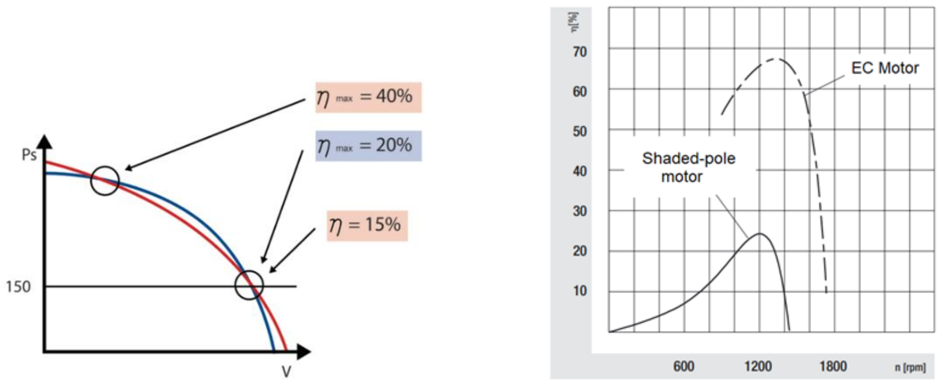
$$\eta_{fan} = \left[\frac{P_{fan}}{(U_{line} \cdot I_{line} \cdot cos\varphi)}\right] = \eta_{motor} \cdot \eta_{imp} \quad (22)$$

The equation developed by Kline et al. was used to define the total uncertainty (W_R) created by measuring the equipment [24]:

$$W_R = \left[\left(\frac{\partial R}{\partial x_1}.w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}.w_2\right)^2 + \left(\frac{\partial R}{\partial x_3}.w_3\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}.w_n\right)^2\right]^{1/2} \quad (23)$$

In Eq. (23), R is the size to measure, $x_1, x_2, x_3, \ldots, x_n$ are the independent variables affecting the measurement and $w_1, w_2, w_3, \ldots, w_n$ are the accuracies in the independent variables.

In addition to theoretical fan efficiency, fan should be operated at the point where optimum system efficiency is obtained. A typical example is given in Fig. 3 where the maximum efficiency of red fan is 40 % and maximum efficiency of blue fan is 20 %. It will be a critical mistake to select the fan according to it's maximum efficiency value. When it comes to efficiency value at the given operating point (150 Pa), it is clear that the blue fan is more efficient than the red fan. Therefore, different fan characteristics and operating points should be taken into account for overall system efficiency. A fan should be operated at its best efficiency point (BEP) to improve fan performance and increase intervals between repairs. The comparison of efficiency characteristic between EC and conventional shaded-pole induction motor is given in Fig. 3b. It is clear that EC motor has superior characteristics to shaded-pole induction motor in terms of partial load and speed efficiencies.



(a) ***(b)***

Figure 3. (a) *Effect of operating point on fan efficiency [25],* ***(b)*** *Comparison of efficiency characteristic between EC and conventional shaded-pole induction motor [26]*

IV. RESULTS AND DISCUSSIONS

The experimental setup designed, produced and tested is shown in Fig. 2. Experiments were carried out using the fans whose specifications are given in Table 2 for the designed and produced systems. No difference was observed in the equipment and cooling power used in the systems where both fans were tested, and the tests were conducted in the same environmental conditions. In the experimental setups, the total energy consumed in the first and second systems was measured as 51.71 kWh, 54.22 kWh, respectively. As a result of these data, the difference between the energy consumption of the two systems was calculated as 4.85 %. The energy-consuming device in heat pump systems are compressors and fans. Since only the fans are different in both systems, it has been observed that the axial fan type 1 used in the first system is more efficient.

For axial fan type 1, the evaporator inlet-outlet temperature of the air is given in Fig. 3. Considering the inlet and outlet air temperatures, defrosting was carried out in the system every 8 hours. However, the average temperatures of the air leaving and entering the evaporator were -21.57 ^{o}C, -18.97 ^{o}C, respectively. In this case, the temperature difference of the system air was calculated as 2.61 ^{o}C.

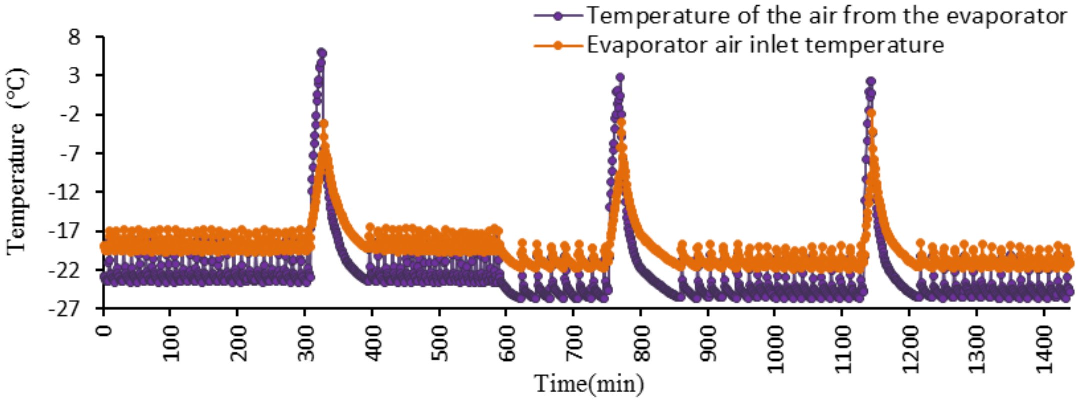


Figure 4. *Evaporator inlet-outlet temperature of the air for axial fan type1*

The evaporator inlet-outlet temperature of the air is given in Fig. 5 for axial fan type 2. In this experiment, the average temperatures of the air inlet to and outlet from the system were seen as -23.43 ºC, - 20.94 ºC, respectively. In this case, the temperature difference of the system air was calculated as 2.49 ºC. Considering the temperature difference between the first and second systems, it was seen that the fan in the first system was more efficient in terms of heat transfer as well as consuming less energy.

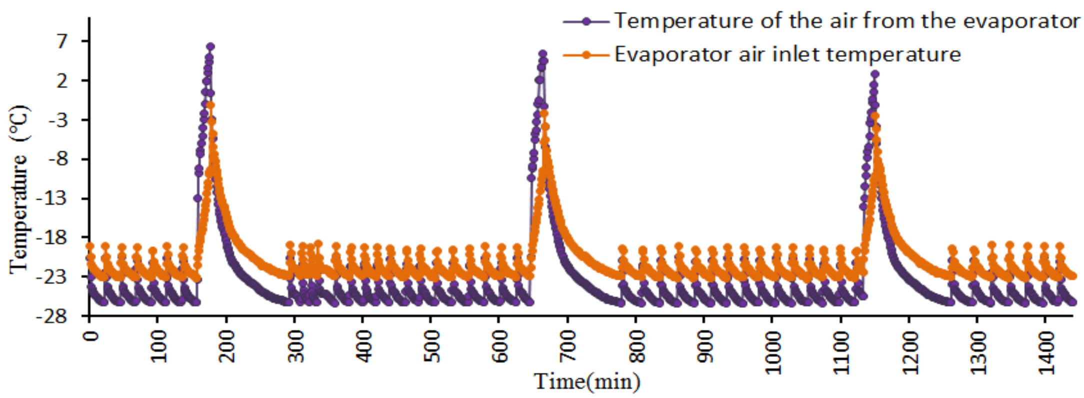


Figure 5. *Evaporator inlet-outlet temperature of the air for axial fan type 2*

As can be seen in Fig. 6, the time-dependent variation of refrigerant inlet temperatures in evaporators for fan type 1 and fan type 2 is shown. As required by the standard, the test setup was operated for 12 hours with lighting and cabinet doors open for 12 hours, and for the next 12 hours with lighting and doors closed. As can be seen in Fig. 5, the cooling system of type 2 must enter the defrost at longer intervals, whereas the cooling system type 1 goes into defrost more frequently. This situation caused a temperature drop on the evaporator surface as a result of the fan not operating at the proper speed; therefore, the desired heat transfer could not be achieved, while increasing the ice formation and defrosting the system in a shorter time. However, average refrigerant temperatures were calculated as -24.65 ºC, -26.44 ºC, respectively. Despite the measured values, the temperature difference was 1.79 ºC.

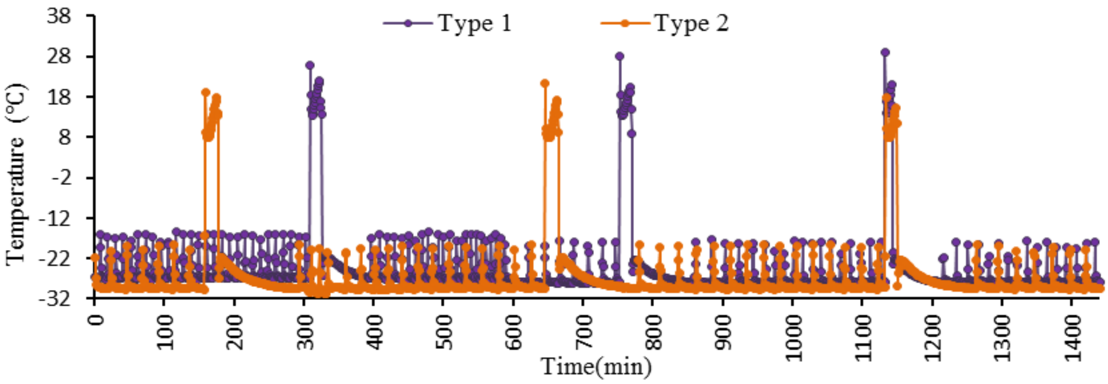


Figure 6. *The change of refrigerant temperature over time at the inlets of the evaporator for two different fans*

Fig. 7 shows the graph created by taking the refrigerant temperatures at the evaporator outlet every minute during the 24-hour test period in both systems. It was thought that the difference between the systems in which fan type 1 and fan type 2 were used in the first 12 hours between the two systems. The doors of the freezer and the lighting were turned on during the first 12 hours as required by the standard, and the cover and lighting are turned off within the next 12 hours. Here, it was observed that the thermal loads that could occur in the system of the type 1 fan affect the air temperature, while the type 2 fan was seen to work more stably despite the thermal loads. Calculated average refrigerant outlet temperatures were determined as -18.97 °C, -19.75 °C respectively. In this case, the difference between the evaporator inlet and outlet temperatures for the two systems is calculated as 6.33 °C, 6.69 °C respectively. In the systems where these two different fans were used, it was seen that the cooling system type 2 was 5.40 % more effective in terms of the evaporator refrigerant temperature difference.

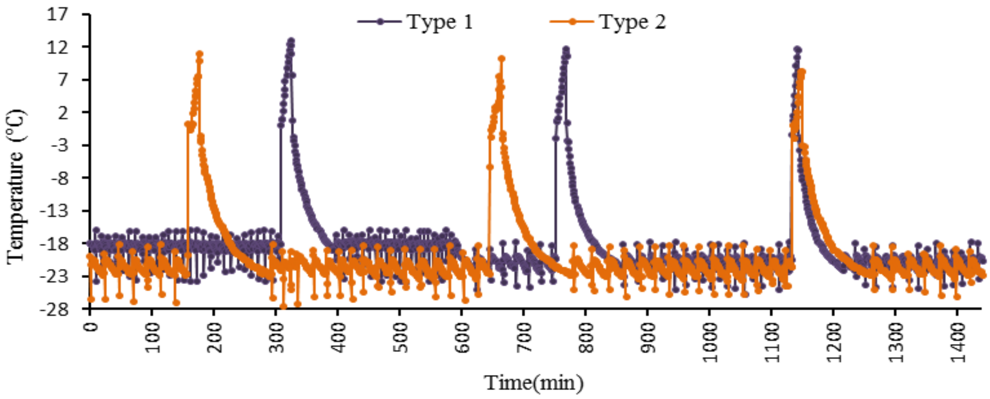


Figure 7. *Examination of the change in refrigerant temperature over time at the evaporator outlet in two different fans*

Fig. 8 shows the average power values calculated every 2 hours. Looking at the graph, it was seen that the power of the type 1 cooling system was lower than the type 2 cooling system. The average power values of the two cooling systems were calculated as 1.26 kW, 1.15 kW, respectively.

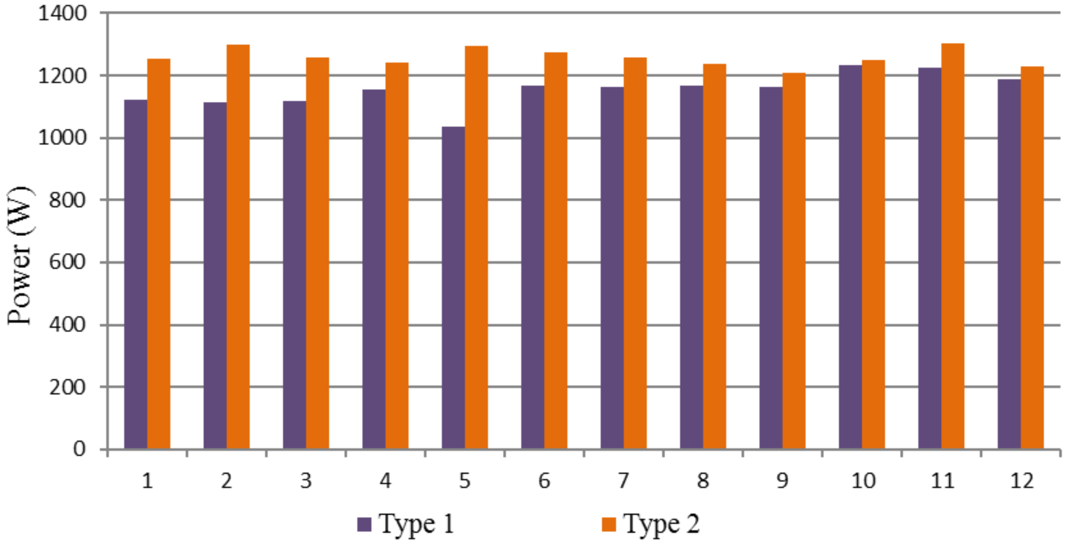


Figure 8. *Comparison of average power values calculated every 2 hours*

Fig. 9 shows the COP values calculated according to the determined power values. In COP values, while the average COP value of the type 1 system was 1.74, it was calculated as 1.54 for the type 2 cooling system. If it is associated with the differences in energy consumption seen in Fig. 7 and Fig. 8 and the results change in efficiency with fans. Important parameters determining the amount of heat absorbed from the surrounding in the evaporator are the resistance and heat transfer coefficient formed on the evaporator surface. The fans will lower the temperature of the evaporator surface depending on the air swept from the evaporator surface. This is desirable because it creates less ice and resistance on the evaporator surface. If the fans carry the water molecules suspended in the air to the evaporator surface together with the air they sweep. The point where these two situations balance each other will determine the operating speed of the fans and it is a very sensitive point. In this system, while the type 2 fan carries excess air, it has been observed that both consume energy and reduce the system performance by carrying the water molecules that will turn into ice on the evaporator surface.

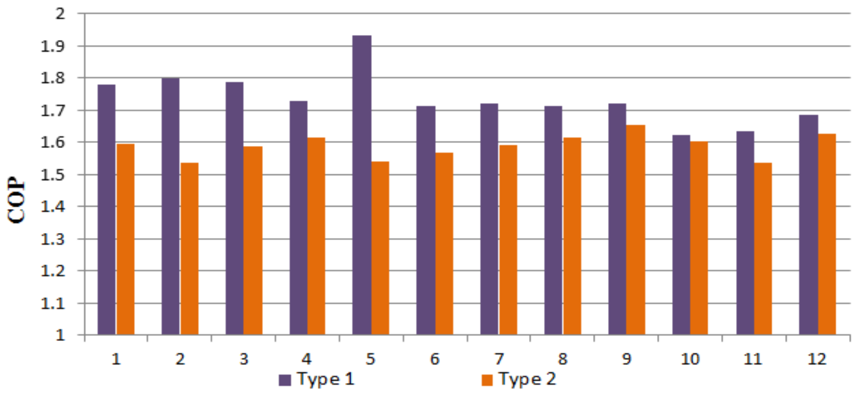


Figure 9. *Comparision of COP values calculated every 2 hours*

Exergy analysis expresses the second law of thermodynamics and makes it possible to evaluate and compare realistically and meaningfully in cooling systems. Exergy analysis for type 1 and type 2 has an important place in terms of determining their cooling performance. Figure 10 illustrates the average hourly exergy efficiency of the cooling systems for type 1 and type 2. The highest exergy efficiency was found as 34.4 % in the 5th hour in type 1 and the lowest exergy efficiency was calculated as 28.7 % in the 2th hour in type 2. The average exergy efficiency values were calculated as 30.85 % and 29.81 % for type 1 and type 2, respectively.

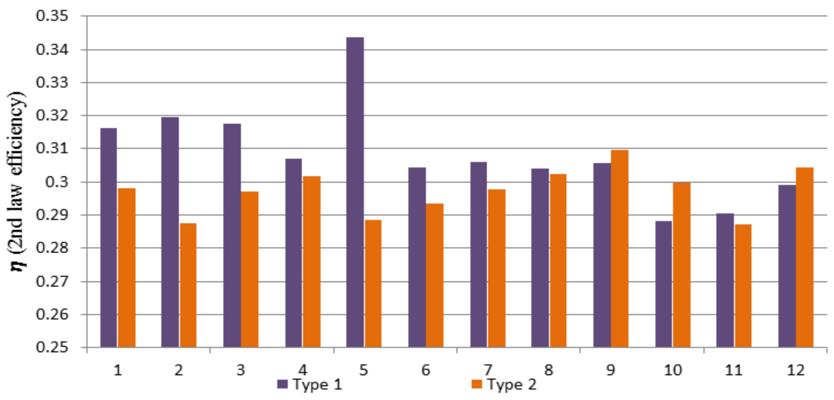


Figure 10. *Comparison of the second law efficiency values calculated every 2 hours*

Environmental economic analysis is carried out using the amount of CO_2 that is prevented from emitting and the CO_2 emission price calculated accordingly. Calculating the CO_2 emission that will occur when it meets the energy consumed by the cooling system from coal will help it determine the amount of CO_2 emission it prevents when it operates the cooling system. Sovacool et al. stated the average intensity of CO_2 equivalent for coal-fired electricity generation as 960 g CO_2/kWh [27]. Tripathi et al. determined that considering this ratio as well as 40 % transmission and distribution losses and 20 % losses due to inadequate electrical devices used, there would be 2.08 kg CO_2/kWh [28]. Arslan et al. explained CO_2 as 1.45 ¢/kg CO_2 in their experimental study and calculated the environmental cost numerically [29]. It can be seen how the $\emptyset CO_2$ value changes in time from Fig.11. According to the figure the maximum, minimum and average value of the $\emptyset CO_2$ for Fan type 1 is 2.981, 1.56 and 2.401 kg CO_2/h; and for Fan type 2, it is determined as 3.105, 1.797 and 2.619 kg CO_2/h. Enviro economic analysis provides information about CO_2, which is prevented from being released into the atmosphere. In the experiments, the amount of CO_2 was prevented from emission when comparing to fan type 2. With the use of fan type 1, 8.32 % more CO_2 emission was prevented. Accordingly, it is possible to say that the use of Fan type 1 is more advantageous than Fan type 2.

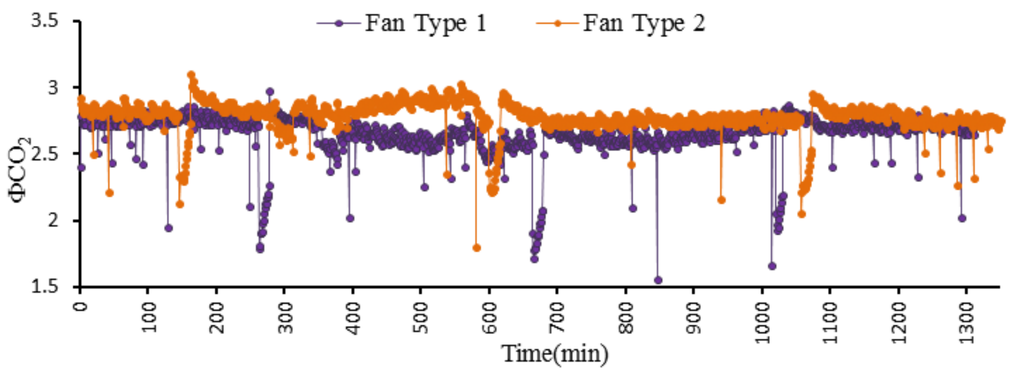


Figure 11. *Change of carbon dioxide, which is prevented from production per hour, during the experiment*

The enviroeconomic analysis relies on CO_2 emission price and determines carbon quantity. Determining a CO_2 price is one of the most powerful analyses revealing the decline of national greenhouse gas emissions. The CO_2 price is an approach imposing a cost on the emission of greenhouse gases which cause global warming. CO_2 price released into the atmosphere is a way of motivating people and countries to reduce carbon emissions. This situation encourages the use of renewable energy technologies that do not emit CO_2 into the atmosphere [22].

The going price of CO_2 (Z_{CO_2}, ¢/h) prevented from emitting into the atmosphere and; the cost of CO_2 per kilogram (PCO_2, 1.45 ¢ / kgCO_2) are equal to the product of the heat produced by the amount of

CO_2 released by the coal burned in 1 kWh electricity production. The going price of the CO_2 amount, which is prevented from being released into the atmosphere, is instantly shown in Fig. 12. Accordingly, the maximum, minimum and average values of ZCO_2 obtained in the experiment using fan type 1 were determined as 4.322, 2.262 and 3.482 ¢/kg, respectively. For fan type 2, these values were obtained as 4.503, 2.605 and 3.798 ¢/kg. The use of fan type 1 seems to be 2.19 % more profitable.

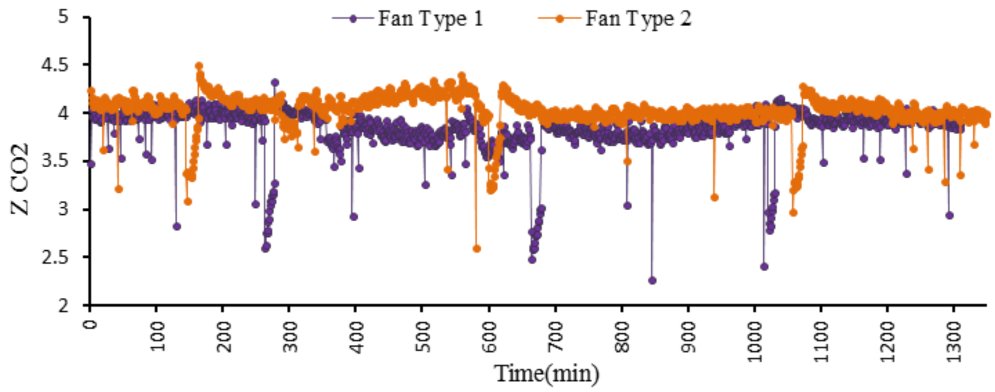


***Figure 12**. The gain per prevention of carbon dioxide product*

It is possible to say that the use of fan type 1 is more advantageous than fan type 2, since the going price is expected to be higher in this experiment, where the amount of CO_2 prevented from being released into the atmosphere is higher as a result of using fan type 1.

Average values of condenser and evaporator exergy destruction were calculated according to Eq. (12) and (13). Accordingly, the average exergy destructions of the condenser and evaporator are 0.178 W and 0.221 W for fan type1, and 0.198 W and 0.243 W for fan type 2. It was determined that the exergy destruction was high in type 2.

V. CONCLUSIONS

This study is on the investigation of the effects of fans on the heat flux and energy consumption of the cooling system in the case of using different types of fans in an industrial cooler. How the fan performance affects the performance of the industrial cooler has been tested experimentally according to the standards and the results have been analyzed. While the energy consumed in the systems designed as a result of the experiments was 51.71 kWh for the first system, this value was measured as 54.22 kWh for the second system. When the two systems were evaluated in terms of performance, COP values were calculated as 1.74 and 1.59, respectively. In the second law analysis, the second-law efficiency of industrial deep freezers was calculated as 30.85% for the first system and 29.81 % for the second system.

In the experiments, the results for different types of fans (Fan Type1 and Fan Type 2) were examined. According to enviro economic analysis, the amount of CO_2 that is prevented from being released into the atmosphere as a result of using Fan type 1 was determined as 2.4 kg CO_2/h, and for Fan type 2 this value was determined as 2.619 kgCO_2/h. As a result of using fan type 2, 8.32% less CO_2 is emitted to the atmosphere compared to Fan type 1.

While the amount of CO_2 prevented from being released into the atmosphere was determined as 3.482 ¢/kg in the experiment using the market Fan type 1, this value was obtained as 3.798 ¢/kg for Fan Type 2. It has been determined that using Fan Type 1 is 2.19% more profitable. In addition to evaporators and other equipment used in cooling systems, the selection of fans that provide air circulation in the system is of great importance. In this context, it is seen that the efficient selection of fans in the deep freezer or cooling system designs is an issue affecting the coefficient of performance and energy efficiency. Energy efficiency can be achieved by providing the desired heat flows at different flow rates with precision speed-controlled fans for heat extraction and removal in the evaporator and condenser. It has been investigated how the fan selection affects the system energy consumption, and according to the experimental results, the energy and exergy efficiency values for different fan types were determined according to the standard. This experimental information obtained will shed light on the selection of fans.

The effects of the fans used on the cooling system are quite complex. Experimental studies are needed to understand the behavior of fans on cooling systems. In this context, the expected results may not be obtained when the selection is made only according to the fan manufacturer data (airflow, pressure, etc.). In order to better understand the behavior and effects of different fans on the cooling system, it is necessary to compare the manufacturer's data and the configuration used with the test results. In this study, the relationship between different fan configurations with known technical specifications and test results was obtained by designing and manufacturing two sample deep freezers.

Suggestions for future works are given as follows;

1- To investigate the effect of partial fan speed and partial fan load on the performance of industrial refrigerator or deep freezer by implementing different fan diameters and blade angles.
2- To obtain mathematical models between used fan combination and cooling performance based on various experimental tests.
3- To implement smart control techniques between the compressor cycle and fan characteristics.

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VI. REFERENCES

[1] J. Wang, Y. Li, Y. Wang, L. Yang, X, Kong and B. Sundén, "Experimental investigation of heat transfer performance of a heat pipe combined with thermal energy storage materials of CuO-paraffin nanocomposites," *Solar Energy*, vol. 211, pp. 928-937, 2020.

[2] B. Utomo, Q. Lailiyah, P. Bakti and I. Paramudita, "Heat transfer characteristic of proposed heat transfer configurations of temperature chamber design for energy test refrigerator," in *American Institute of Physics (AIP) Conference Proceedings*, 2020, vol. 2248, pp. 050002-1-050002-10.

[3] J. Ye, X. Huang, Y. Cheng, J. Shao and Y. Zhang, "Air volume improvement in the duct system in frost-free refrigerators based on the CFD method," *Journal of Supercomputing*, vol. 76, no. 5, pp. 3749-3764, 2020.

[4] F.G. Modarres, M. Rasti, M.M. Joybari, M.R.F. Nasrabadi and O. Nematollahi, "Experimental investigation of energy consumption and environmental impact of adaptive defrost in domestic refrigerators," *Measurement*, vol. 92, pp. 391-399, 2016.

[5] R.S. Ribeiro, D.L. Silva and C.J.L. Hermes, "Optimal design of fan-supplied tube-fin evaporators subjected to frosting conditions aiming at minimum energy consumption," *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, vol. 40, no. 479, pp. 1-11, 2018.

[6] S. Erten, C. Ocak and M. Aktaş, "Experimental analysis of fan performance in industrial cooling systems," in *Proceeding of 9th International Conference on Advanced Technologies (ICAT'20)*, Bolu, Turkey, 2020, pp. 551-562.

[7] W. Dong, Y. Liu, Z. Kou, L. Yao, L. Tao and P. Xia, "Energy and exergy analysis of an air-source heat pump water heater system using CO_2/R170 mixture as an azeotropy refrigerant for sustainable development," *International Journal of Refrigeration*, vol. 106, pp. 628-638, 2019.

[8] P. Acharya, B.K. Choudhury, S.K. Rout, "Effect of speed of condenser fan motor on vapor compression refrigeration system," *Advances in Air Conditioning and Refrigeration*, Singapore: Springer, 2021, pp. 395-403.

[9] A.O. Elsayed, T.S. Kayed, "Dynamic performance analysis of inverter-driven split air conditioner," *International Journal of Refrigeration*, vol. 118, pp. 443-452, 2020.

[10] H. Yang, , N. Pei, , L. Liu, , M. Fan, , Y. Qin, "Experimental study on the effect of condensate water on the performance of split air conditioning system," *Energy Reports*, vol. 7, pp. 840-851, 2021.

[11] M. Thakre, C. Shinde, "Evaluation of a variable speed DC compressor for energy efficiency employing refrigerant-R290," in *International Conference on Power, Energy, Control and Transmission Systems (ICPECTS)*, Chennai, India, 2020, pp. 1-5.

[12] F. Hermosa, C. Tasiguano, M. Pozo, E. Acurio, "Controller design for high-energy-efficient performance of a household refrigerator using inverter technology," in *Proceedings of the 18th LACCEI International Multi-Conference for Engineering, Education and Technology*, Buenos Aires, Argentina, 2020, pp. 1-7.

[13] D. Colorado and W. Rivera, "Performance comparison between a conventional vapor compression and compressionabsorption single-stage and double-stage systems used for refrigeration," *Applied Thermal Engineering*, vol. 87, pp. 273-285, 2015.

[14] D.C. Panigrahi and D.P. Mishra, "CFD simulations for the selection of an appropriate blade profile for improving energy efficiency in axial flow mine ventilation fans," *Journal of Sustainable Mining*, vol. 13, no. 1, pp. 15-21, 2014.

[15] X. Zhao, J.Sun, Z. Zhang, "Prediction and measurement of axial flow fan aerodynamic and aeroacoustic performance in a split-type air-conditioner outdoor unit," *International Journal of Refrigeration*, vol. 36, no. 3, pp. 1098-1108, 2013.

[16] Z. Dang, Z. Zhang, M. Gao, S. He, "Numerical simulation of thermal performance for super large-scale wet cooling tower equipped with an axial fan," *International Journal of Heat and Mass Transfer*, vol. 135, pp. 220-234, 2019.

[17] A. Zulkifli and Zulfikri, "Optimize performance of split AC evaporator using variable speed drive in motor fan evaporator," in *American Institute of Physics (AIP) Conference Proceedings*, 2020, vol. 1983, pp. 020010-1 -020010-7.

[18] G. Angelini, T. Bonanni, A. Corsini, G. Delibra, L. Tieghi, D. Volponi, "Optimization of an axial fan for air cooled condensers," *Energy Procedia*, vol. 126, pp. 754-761, 2017.

[19] M. Caner, N. Duman, E. Buyruk and F. Kılınç, "Performance analysis of horizontal ground source heat pump system in Sivas," *Journal of Science and Technology of Dumlupinar University*, vol. 42, pp. 47-53, 2019.

[20] M. Aktaş, M. Koşan, E. Arslan and A.D. Tuncer, ”Designing a novel solar-assisted heat pump system with modification of a thermal energy storage unit,” *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy* , vol. 233, no. 5, pp. 588-603, 2019.

[21] Y.A. Çengel and M.A. Boles, *Thermodynamics: An Engineering Approach*, 8th ed., New York, USA: McGraw-Hill Higher Education, 2014.

[22] M.G. Elzen, A.F. Hof, A.M. Beltran, G. Grassi, M. Roelfsema, B. van Ruijven, J. van Vlietvan, and D.P. Vuuren, “The Copenhagen Accord: abatement costs and carbon prices resulting from the submissions,” *Environmental Science & Policy*, vol. 14, no. 1, pp. 28–39, 2011.

[23] C.D. Pitis, “Energy efficient single stage axial fan (ENEF),” in *IEEE Canada Electrical Power Conference*, Montreal, Canada, 2007, pp. 280-285.

[24] Kline, S. J., and McClintock, F. A., “Describing uncertainties in single sample experiments,” *Mechanical Engineering*, vol. 75, pp. 3-8, 1953.

[25] Systemair Technical Handbook. (Oct. 20, 2019). *Ventilation*, [Online]. Available: www.systemair.com/fileadmin/user_upload/systemair-b2b/Support/Technical_Handbook_EN_2019-10_E2029.pdf.

[26] Ebmpapst. (Nov. 4, 2018). *Comparison: Energy-saving motor (ESM) vs. Q-motor* [Online]. Available: www.ebmpapst.co.uk/media/content/news_media/ebmpapst_news/content_epnews/FirstSpirit_1421932187875Schulung_ESM_E~1.pdf.

[27] B.K. Sovacool, “Valuing the greenhouse gas emissions from nuclear power: a critical survey,” *Energy Policy*, vol. 36, no. 8, pp. 2950–2963, 2008.

[28] R. Tripathi, G.N. Tiwari and V.K. Dwivedi, “Overall energy, exergy and carbon credit analysis of N partially covered Photovoltaic Thermal (PVT) concentrating collector connected in series,” *Solar Energy*, vol. 136, pp. 260–267, 2016.

[29] E. Arslan, and M. Aktaş, “4E analysis of infrared-convective dryer powered solar photovoltaic thermal collector,” *Solar Energy*, vol. 208, pp. 46–57, 2020.