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Development of Multi-Functional Water and Ice Dispensing Machine

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***Abstract*—The need for ice, especially for drinks and other purposes, at homes and in offices in small quantity cannot be underestimated. The available alternatives (refrigerators and iced block making machines) are economical and efficient only when medium or large quantity of ice is needed. Water dispenser has been a perfect substitute for refrigerators due to its ability to supply clean, chilled/hot water for drinking when needed in small quantity. To make small quantity of ice available at our disposal, there is the need for a system capable of performing these tasks. This paper presents the development and performance evaluation of a multi-functional water and ice dispensing machine that will dispense ice in addition to the normal cold and hot water. In order to achieve quick freezing, certain design considerations, such as: increasing the area of heat transfer of the heat exchangers and employing a multi-parallel compressors arrangement in the vapour compression refrigerating system to achieve the refrigerating effect. The results of the performance evaluation revealed that the machine dispensed warm water at temperature range of less than or equal to 90 °C at the rate of 5 liters per hour, cold water at temperature range of 10 to 13 °C at the rate of 2 liters per hour and produced 24 pieces of 20 x 20 x 20 mm iced-cubes in less than 90 minutes.**

Keywords—Water, Ice, Dispensing Machine, Development, performance evaluation

Nomenclatures

Q : Cooling load (kWh);
U : Thermal transmittance for the insulation panel;
T : Temperature (°C)
m: Mass (kg)
C_p : Specific heat of water (KJ/Kg°C)
$Q_{amb/zero}$: Cooling load required to cool water from ambient temperature to 0 °C (kWh)
$Q_{w/ice}$: Cooling load required to convert 0 °C water to 0 °C ice (kWh)
L_{ice} : Specific latent heat of fusion of ice (kJ/kg°C)
K_{ice} : Thermal conductivity of ice (kW/mK)
$Q_{ice/f}$: Cooling load required to cool ice further to -10 °C (Kwh)
$Q_{f/ice}$: Cooling load required to make ice in 15minutes (Kwh)
U_{air} : Convection heat transfer of air
T_{amb} : Ambient temperature of air (°C)
$Q_{m/ice}$: Cooling load required to prevent the ice from losing its coolness (Kwh)
N_p : Number of people inside
h_l : Heat loss per person per hour (W)
N_{lamp} : Number of lamps within the host room

W: Power rating of the lamps (W)
P_{hw} : Power required to heat water (W)
V_{hw} : Volume of water to be heated (m^3)
t_{hw} : Time required to heat water to desired temperature
l_{ice} : Length of ice (mm)
b_{ice}: Breadth of ice (mm)
h_{ice}: Height of ice (mm)
T: Temperature of ice (°C)
h_e: Heater element rating (W)
P: Pressure
h_l: Enthalpy of Saturated liquid
h_g: Enthalpy of Saturated liquid
s_l : Entropy of Saturated liquid
s_g : Entropy of Saturated vapour
$v_{i(ice)}$: Specific volume of refrigerant at suction (m^3/kg)
R_c: Refrigeration capacity (kW)
R_f : Refrigerant flow rate for ice (kg/s)
W: Work Done on refrigerant for ice (kJ/s)
$H_{rc(ice)}$: Heat rejected in the condenser for ice (kJ/s)
$H_{rc(cold\ water)}$: Refrigerant flow rate for cold water (kJ/s)
$H_{ae(ice)}$: Heat absorbed in the evaporator for ice (kJ/s)
$H_{ae(cold\ water)}$: Heat absorbed in the evaporator for cold water (kJ/s)
V: Volume of suction fluid (m^3/s)
A: Area (m^2)
$P_{td(ice)}$: Theoretical piston displacement

$L_{(pipe)}$: Length of condenser pipe
Nu: Nusselt number
Pr: Prandtl Number

Subscripts
til: Total product load
tl: Transmission load
c: Condenser
e: Evaporator
ae: Absorbed by evaporator
rc: rejected by condenser
sv: suction fluid
dr: work done on refrigerant
occ: Occupants
ex: Outside temperature
in: Inside temperature
d: Desired temperature
wr: Walls and roof
f: floor
cw: Cold water
ice: Ice
eq: Equipment

I. INTRODUCTION

Over time, several machines have been developed to make man comfortable. One of such is the water cooler which was first invented in the early 1906, whose invention was credited to Halsey Willard Taylor and Luther Haws. Haws patented the first drinking faucet in 1911. The main reason behind the invention was to provide potable water, as his father died of typhoid fever caused by drinking contaminated water [1].

As a result of improvement in standard of living of people, the use of modern and hygienic home appliances becomes a necessity in different households. Household refrigerators became popular in the early 1900s but only the wealthy could afford them at the time. Freezers did not become a staple part of the refrigerator until after World War II when frozen foods became popular [2] The invention of water dispensers is indeed a success as the machine has now become a major household appliance. Most time, people tend to sandwich wine or sip cups of coffee with certain cubes of ice block so as to make them chilled whilst drinking or drink cold water during meals, reduce the temperature of sausage dough during meat chopping in bowl cutters, retain the freshness of foods [3], reduce the number of carcass-associated bacteria [4, 5, 6, 7].

Cubes of ice are either obtained from freezer(s) or ice-block making machine(s), cold water obtained from refrigerating (fridge type) or water dispensing systems; this existing dispensing system can only provide cold and warm water. More so, the existing refrigerating system can only provide cold water and ice-block. In another words, none of the available system could jointly perform the tasks. However, with the way the machine is being used (a substitute for refrigerators), and need for cubes of ice to meet these purposes, either at home or in the offices, it becomes imperative to develop a device capable of performing these tasks (dispensing cold water, hot water and cubes of ice) when needed. Therefore, the objectives of this study are to develop a device capable of performing the three tasks and evaluate its performance. This machine was designed on the basis of vapour compression refrigeration cycle, which is simply a refrigeration cycle that absorbs heat from a cool environment and rejects it to a warm environment by the vacuum vaporization of the working fluid volatile liquid, called refrigerant.

Vapour compression refrigeration cycles, specifically, have two advantages which make it best suitable for this design; Firstly, it exploits the large thermal energy required to change a liquid to a vapour which enhances the removal of high quantity of heat from water to cause a phase change. Secondly, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to the temperature of item is being refrigerated, this is a benefit because the farther the cooling fluids temperature is from that of the body being cooled, the higher the heat transfer rate [8].

II DESIGN ANALYSIS

- Design Assumptions

According to [9, 10], assumptions represent the design intent derived from past experiences, industry knowledge or personal intuition; they serve as the setting, and provide foundation for making progress and testing ideas pertaining to research in question. Hence, for the purpose of making basis for the planning of this study, the following assumptions were utilized for developing and evaluating the performance of the machine; based on data obtained through participant observations, findings of scholars and relevant engineering standards, such as[11, 12, 13, 14]:

(i) Cooling unit dimension is 560 mm x 56 mm x 1010 mm; this dimension was derived from existing systems of similar capacity;
(ii) Ambient air is 30 ^oC at 50 % relative humidity;
(iii) Internal air is 1 ^oC at 95 % relative humidity;
(iv) Specific capacity of refrigerant (R600a) at constant pressure is 2.31 J/kgK;
(v) Specific capacity of refrigerant (R600a) at constant volume is 1.63 J/kgK;
(vi) Thermal conductivity of insulator is 0.28 W/m^2K,
(vii) Floor temperature is 25 ^oC;
(viii) Density of water is 997 kg/m^3 at 25 ^oC;
(ix) Volume of water to be cooled is 0.002 m^3 per hour for 16 hours of usage per day;
(x) Temperature of water entering (T_{win}) is 25 ^oC;
(xi) Desired temperature of cold water is ≤ 10 ^oC;
(xii) Volume of water to be converted to ice is 0.002 m^3;
(xiii) Density of ice is 900 kg/m^3 at 0 ^oC;
(xiv) Desired temperature of ice (T_{ice}) is -10 ^oC;
(xv) Time for converting water to ice is 15 minutes;

(xvi) Ambient temperature (T_{amb}) is 25 ^oC;
(xvii) Geometrical dimensions of ice cube size are 20 mm x 20 mm x 20 mm;
(xviii) Volume of water to be heated (V_{hw}) is 0.005 m^3;
(xix) Desired temperature of hot water (T_{hw}) is 60 ^oC;
(xx) Time required for heating water (t_{hw}) is 12 minutes;
(xxi) assumed that 5 people resided in the conditioned space and would be using the machine for 16 hours a day considering that each of the occupant gives off 270 W of heat per hour
(xxii) Host room has 3 lamps at 100 W each, running for 4 hours a day;
(xxiii) Speed of compressor ranges from 1450 – 2800 rpm (1450 rpm assumed);
(xxiv) Temperature difference (TD) ranges from 5 – 17 ^oC (average assumed, 11 ^oC)

II MATERIALS AND METHOD

- Design Procedure

For the purpose of developing the Ice chips-water dispensing machine, the underlisted sequence of approach was considered:
(i) The idea of designing hot water/cold water/ice dispenser was conceived and the functional requirements of the machine were established with reference to existing systems of this nature;
(ii) The cooling load of the proposed machine was estimated and the applicable parts sizes selected in line with engineering standards;
(iii) Different design sketches and assemblies for the machine and components were made. The best of the sketches was chosen, through ranking,
(iv) Standard parts, having specifications / dimensions in tandem with the calculated data, were selected/adopted and procured in order to allow for principle of interchangeability of parts;
(v) Individual components were constructed using detailed engineering drawing as a guide;
(vi) the components were assembled as detailed in the exploded drawings, and
(vii) The developed machine was tested, and measured data taken with the attached instrumentations documented for establishing its performance characteristics and the degree of conformity

- Design and Calculations

The basic components of the dispensing machine comprise the: evaporator, condenser, compressor, throttling device, strainer dryer and heating element. This section presents details of the design analyses of the basic components of the system in consonance with relevant engineering standards and findings from researches of relevant scholars.

- Evaporator design

It is the component on which the refrigerant vapourizes for the purpose of removing heat from the refrigerating item. The heat to be absorbed by the refrigerant circulating in the evaporator or total cooling load (Q_T) is a summation of the:

- Transmission load (Q_{tl})
- Product load (Q_{tpl})
- Internal load (Q_{til})
- Equipment load (Q_e)
- Infiltration load (Q_i)
- Safety factor
- Sensible heat gained from hot water reservoir (Q_{hw})

The transmission load was calculated using equation (1):

$$Q = [UA(T_{amb} - T_{in})]\ 24]/1000 \qquad (1)$$

The total product load (Q_{tpl}) is a summation of the cooling load for cold water (Q_{cw}) and ice ($Q_{tl/ice}$). The cooling load for cold water was calculated using the expression:

$$Q_{cw} = m_{np}\, C_p (T_{in} - T_d)\ /\ 3600 \qquad (2)$$

The refrigeration load required for the formation of ice ($Q_{tl/ice}$) is the summation of the thermal load required to cool the water from its inherited ambient temperature down to 0 ^oC, the load required to phase-changing it to ice 0 ^oC, the load required to cool the formed ice further to the designed temperature (-10 ^oC) and the load required to prevent the ice from losing its coolness.
Refrigeration load required to cool the water from ambient temperature down to 0 ^oC was derived with the equation (3):

$$Q_{amb/zero} = 4m_{ice}C_P(T_{in} - 0)/3600 \qquad (3)$$

The Refrigeration load required to cause phase changing was derived with the expression (4):

$$Q_{w/ice} = 4(m_{ice}\, L_{ice}) \qquad (4)$$

Refrigeration load required to cool ice further to -10 ^oC was calculated with equation (5):

$$Q_{ice/f} = 4m_{ice}s_{ice}(0 - T_{ice})/3600 \qquad (5)$$

Refrigeration load required to prevent the ice from losing its coolness was calculated using the equation (6):

$$Q_{m/ice} = 4Ulb(T_{amb} - T_{ice})/3600 \qquad (6)$$

where; U is the heat transfer coefficient of ice and air derived with equation (7):

$$U = 1/[\left(\frac{1}{U_{air}}\right) + \left(\frac{h}{K_{ice}}\right)] \qquad (7)$$

The internal load (Q_{til}) is the summation of the heat load generated by the occupants (Q_{occ}) and lighting (Q_l). The Cooling load required due to heat generated by occupants was derived with equation (8):

$$Q_{occ} = N_p t h_l/1000 \qquad (8)$$

and the cooling load due to lighting was derived with the expression described in (9):

$$Q_l = N_l t_{lamp} W/1000 \qquad (9)$$

The equipment load (Q_{eq}) denotes the load caused by the heat generated by the system equipment, like fan motors and defrosting of the evaporator.
Infiltration load (Q_i), according to [1] , connotes the heat gained due to uncontrolled leakage of air; as a

result of closing/opening of the doors, through cracks or joint between its walls, apertures created for inserting instrumentations, etc. Since the development of this machine has not been made, it is hard to get the exact value to account for it [15]; in order to cater for the loss that might emanate from this, a 10 % of the total cooling load was considered.
The sensible heat gained from heat radiated from the hot water reservoir (Q_{hw}) was calculated with equation (10):

$$Q_{hw} = 24[UA(T_{ex} - T_{ei})/1000 \quad (10).$$

The refrigeration effect produced over the entire run time of the refrigeration unit was obtained with equation (11):

$$R_c = Q_T/\mathrm{t} \quad (11)$$

Based on this capacity of the evaporators and the need to make the system compact, bare-tube type evaporators developed with copper tubing were chosen and utilized.

$$H_{ae} = \frac{R_f}{(h_1 - h_4)} \quad (12)$$

The mass flow rate of the refrigerant required for achieving the cooling in the evaporators was calculated using equation (13)

$$m_f = \frac{R_c}{(h_2 - h_1)} \quad (13)$$

The geometrical configurations (area and length of the evaporator were computed with equations (14) – (15):

$$A_e = \frac{H_{ae}}{U\Delta T_m} \quad (14)$$

$$l_{epipe} = \frac{H_{ae}}{\pi d_i U \Delta T} \quad (15)$$

Thus, the overall heat transfer coefficient, based on the inside area, was determined with modified equation (16) described by Rajput (2009):

$$U^{-1} = \frac{r_i}{\bar{h} r_0} + \frac{r_i}{Kcopp} In[^{\mathrm{r}_0}/_{r_i}] \quad (16)$$

The inside heat transfer coefficient $\bar{h}$ was deduced with equations (17) and (18):

$$\bar{\mathrm{h}} = \frac{\mathrm{NuK_r}}{\mathrm{d_i}} \quad (17)$$

$$Nu = 0.023(Re)^{0.8}(\mathrm{Pr})^{0.4} \quad (18)$$

- Power rating of heating element

The power required to heat water from the initial to the desired temperature (P_{hw}) was determined given by the relation described with equation (19):

$$h_e = \frac{4.2V_{hw}(T_{hw} - T_{win})}{3600.t_{hw}} \quad (19)$$

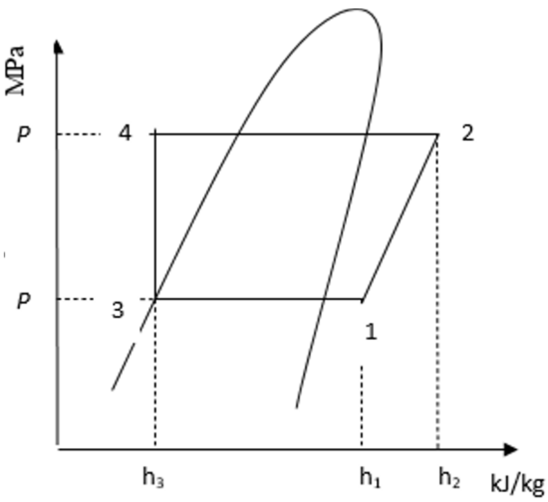
Table 1 presents details of the evaporator design parameters

Table 1: Evaporator design parameters

Parameter	Cold water chamber	Ice chamber
Heat absorbed by evaporator (kW),	1.089	0.992
capacity (Btu/h Tons)	3715.82 0.310	3384.84 0.282
Refrigerant mass flow rate (kg/s)	0.0033	0.0031
Coefficient of Performance (COP)	13.2	6.4
Evaporator area (m^2)	0.02213	0.0128
Thermal conductivity of copper (W/m^2k)	401	401
Reynolds number	1359.4	27.2
Length of evaporator pipe (m)	2.14	0.82
Tube internal diameter (mm)	3.26	5

- Compressor design

The compressor It is often considered as the heart of the refrigeration system. It performs dual functions as it pumps the refrigerant from the system so that desired pressure can be maintained in the evaporator and raises the pressure of refrigerant exiting the evaporator to the condenser pressure so that vapour refrigerant formed can be condensed at the corresponding saturation temperature by rejecting heat to the surroundings [1; 16], Figure 1 shows the P-h diagram of the refrigeration cycle.



The mass of refrigerant (m_{rf}) required to be circulated by the compressor, in both chambers (cold water and ice) was obtained with equation (20):

$$m_{rf} = \frac{R_f}{(h_2 - h_1)} \quad (20)$$

The volume of suction fluid (V_{sv}) in both chambers was determined using the relation (21):

$$V_{sv} = m_{rf} v_i \quad (21)$$

while the volumetric flow of refrigerant per minute (V_r) through the compressors were estimated with equation (22)

$$V_r = \frac{\pi D_p^2 L_s N_c n}{4} \qquad (22)$$

The properties of the compressor were estimated with equations (21) and (22), as described by [15, 16]. Table 2 presents the Compressors design parameters

Table 2 Compressors design parameters

Parameter	Unit	calculated		adopted	
		Cold water	ice	Cold water	ice
size	R_c (HP) Btu/h Tons	1.8	1.3	1.5 3816.7 0.3181	1.5 3816.7 0.3181
Operating cycle	Hr	16	16	16	16
efficiency	η_c (%)	85	85	85	85
power	W_{dr} (kW)	0. 0825	0.155	0.0688	0.179
suction volume	V_{sv} (m³)	0.000743	0.0012	0.000619	0.0014
piston displacement	V_r (cm³)	0.512	0.828	0.427	0.966

Based on the fact that the thermal requirements of the two evaporators are not the same, the two compressors were arranged in parallel (multi-parallel arranged).

- **Condenser Design**

It is the mirror image of the evaporator, according to [14], as it functions to reject the heat absorbs by the evaporator, and those emanating from suction line superheat, motor heat picked up by the suction vapour on its way through the compressor and work of compression.

In practice, condenser size is determined based on its total heat rejection (THR), which denotes the total heat load of the system and the energy added to its working fluid by the compressor. The heat rejected in condenser was obtained with equation (23):

$$H_{rc} = \frac{R_f}{h_2 - h_3} \qquad (23)$$

Area of the condenser was evaluated with equation (24):

$$A_c = \frac{H_{rc}}{U_{cop}(\Delta T_m)} \qquad (24)$$

$$U^{-1} = \left[\frac{d_o}{d_i}\right]\left[\frac{1}{h_i}\right] + \left[\frac{1}{2K}\right] d_o \, x \, In\left[\frac{d_o}{d_i}\right] + \frac{1}{h_o} \qquad (25)$$

The fin coefficient for Refrigerant side is calculated with equations (26) – (30) as:

$$h_i = \frac{N_u K}{d_i} \qquad (26)$$

$$R_e = \frac{\rho v d_i}{\mu} \qquad (27)$$

$$P_r = \frac{\mu C_p}{k} \qquad (28)$$

and

$$N_u = \pi \frac{\left(f/8\right)(R_e - 1000)P_r}{1+12.7\sqrt{f/8}\left(P_r^{\frac{2}{3}}-1\right)} \qquad (29)$$

For 2300 ≤ R_e ≤ 5 x 10^6

$$f = \frac{1}{(1.82\log_{10}\mathrm{Re}-1.64)^2} \qquad (30)$$

At the air side of the condenser, the heat transfer coefficient was determined with the empirical formulae described for staggered tube rows in tube bundles, according to [12, 17] .

$$N_{ud_0} = \frac{\bar{h} d_o}{K} \qquad (31)$$

$$h = \frac{N_{ud_0} K}{d_o} \qquad (32)$$

$$R_e = \frac{\rho v_{airmax} d_o}{\mu} \qquad (33)$$

$$N_{ud_0} = Pr^{0.36}\left(Pr/Pr_W\right)^n fn(Re_{do}) \qquad (34)$$

Since it is a compressed gas that flows in through the condenser then n = 0

$$N_{ud_0} = Pr^{0.36} fn(Re_{do}) \qquad (35)$$

The length of the condenser pipe was evaluated using:

$$l_{cpipe} = \frac{A_c}{\pi D} \qquad (36)$$

Table 3 presents summary of the condenser design analysis: **Table 3:** Condenser design for cold water

Variable	Symbol	Cold water	Ice
		chamber	
Hear rejected (kJ/s)	Hrc	1.1715	1.147
Condenser capacity	Btu/h Tons	3997.32 0.333	3913.73 0.326
Condensing temperature	ºC	11	11
Condenser area x 10^{-4}(m²)	A_c	4.69	5.73
Pipe diameter (mm)	d_{ic}	3.26	3.26
Length of pipe 10^{-2} (m)	lcpipe	2.99	3.65
Refrigerant flow rate 10^{-3} (kg/s)	Rf	3.3	3.1
Reynolds number	Re	858.9	13.4
Thermal conductivity (W/m²K)	U	401	401
Coefficient of performance	COP	13.2	6.4

- **Throttling device**

Based on the need to equalize the pressure developed at each of the two ends of the compressors when the system is switched off at the expense of reducing the starting torque that might promote the need for modulating metering device to be used to achieve this during start up, a factor for increasing costs (procurement and maintenance) of the compressors, a fixed metering device (e.g. capillary tube) was used; it was selected because the thermal load is relatively stable and is able to offer stable flow rate without the need of maintenance cost as it has no moving part [13]. Different sizes are available in the market, but based on the horsepower of the compressor, the appropriate size is 1 mm diameter capillary tube and was selected.

To determine the length of capillary tube needed to achieve the desired pressure drop, the equations (37) – (39) were employed.

$$A_{cap} = \frac{\pi D^2}{4} \qquad (37)$$

$$G = \frac{\dot{m}_r}{A_{cap}} \qquad (38)$$

They were employed based on an assumption that the flow is one-dimensional, incompressible and single-phase:

$$\Delta L = \frac{-\Delta P - G\Delta V}{\left(G/_{2D}\right)(fV)} \qquad (39)$$

Hence, the length of capillary tube for the ice chamber is 5.5 m and for the cold-water chamber is 3.9 m.

- **Strainer Drier**

The efficient operation of a refrigeration system depends greatly on the internal cleanliness of the unit. Since cleaning of refrigeration system involves (removing) removal of moisture and acid formed due to mixture of contaminant, as well as filtering out circulating solids, tasks that are not easily possible because the system is a close type. The device employed for these tasks is filter-drier; a Capsule charged with desiccant was used. The capsules have strainers to prevent loss of the drying agent into the circuit and the valve orifice from damages by fine debris.

- **Instrumentations**

The machine is equipped with two Giant DC4 analog thermometers for measuring the temperatures in the two evaporators. The pressure readings were obtained using oil pressure gauges. Helect infrared thermometer was used to measure the condenser temperature and a Multi Thermometer (Model: H-9283) was used to measure the ambient temperature

- **Fabrication**

The machine houses a common bottled water dispenser mounted on a frame. The frame, with dimensions of 610 mm x 750 mm, was constructed using 38.5 mm x 38.5 x 3 mm angle iron. Other components including the compressor for the ice compartment, condenser, dryer and capillary tubes and the electrical components were mounted on the frame. HDF plywood was used to cover the machine, considering its availability and cost compared to iron sheets. The machine was then painted ash because of its heat- repulsion property. The machine has an overall size of 750 x 610 x 940 mm. the fabrication was done at the Central Workshop of the Department of Mechanical Engineering, The Federal University of Technology, Akure (FUTA). Figure 4 shows the isometric view and Plate 1 the picture of the developed machine.

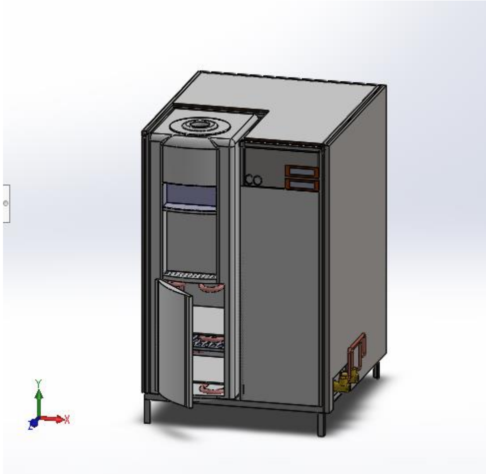


Figure 4: Isometric view of the developed machine.

Plate 1 the picture of the developed machine.

III RESULTS AND DISCUSSION

Figures 4, 5, 6 and 7 present the experimental results for the cold water and ice compartments taken with attached instrumentations.

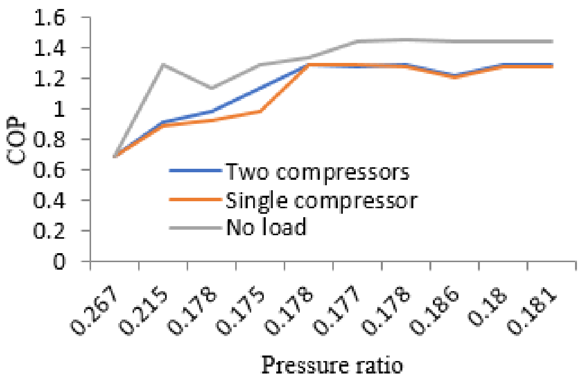


Figure 4: Variation of pressure ratio with COP

The performance of a refrigeration system is measured in terms of its Coefficient of Performance (C.O.P.), which is defined as the heat removed at a low temperature. The cooling effect, Q_c, is equal to the change in enthalpy of the working fluid as it passes through the evaporator, and the work input (W_{cp}) is equal to the increase in the working fluid's enthalpy as it passes through the compressor

Figure 4 presents the variation of COP with the pressure ratio. It can be seen that the COP increased with increase in pressure ratio (ratio of suction to discharge pressure). Initially during cold start, the system was noticed to exhibit increase in pressure ratio. As the system however approached the design conditions, the pressure ratio reduced. This is because at cold start, the compressor works assiduously to circulate the refrigerant so as to overcome the pressure created by the heat-laden refrigerating sample (with initial temperature of 25.2 ^{o}C) in the target box, a value higher than the evaporator temperature (24 ^{o}C), to achieve the needed cooling effect in the evaporator. There was 46 % increase in the COP of the system with a 32 % decrease in the pressure ratio suggesting that the system is performing optimally by creating adequate refrigerating effect per work of compression. It could also be noticed from the curve that the system had a higher efficiency when the two compressors were in operation (21% increase in COP). This effect could be attributed to the reduction in the work of the compression in spite of the progressive rise in the COP.

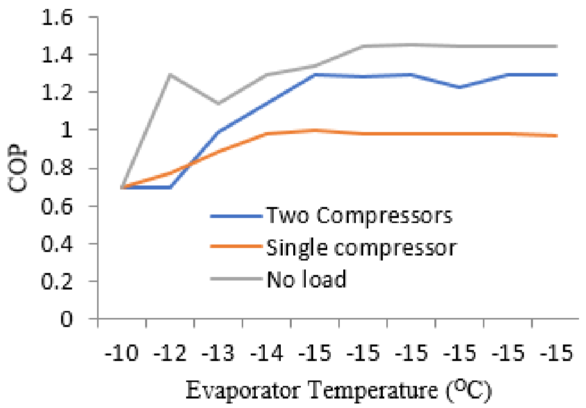


Figure 5: Variation of evaporator temperature with COP

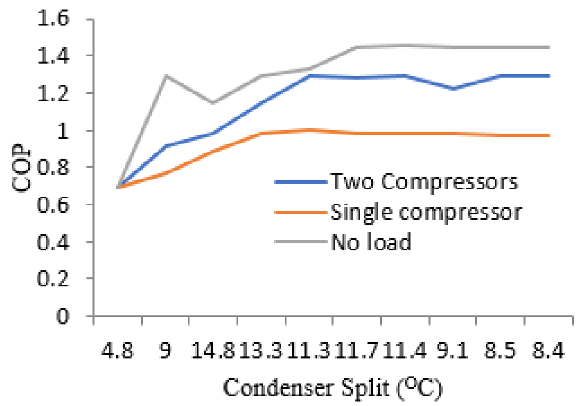


Figure 6: Variation of Condenser split with COP

Figures 5 and 6 present the variation of COP with evaporator temperature and Condenser split. From the plot, it was observed that the COP increased with increase in evaporator temperature at first instance, because the system was hot-pulled. However, after the COP of 1.289, it started to decline; the reason for this decline could be attributed to the gradual decrease in the thermal load of the target box which favoured more liquid refrigerant, instead of vapour, to be circulated by the compressor, which also resulted in the progressive increase in the pressure ratio as the experimentation proceeded. This agrees with the findings of [18, 19, 20] that the COP increases with increase in evaporator temperature. More so, the system is more efficient when there was no load. This is because of the reduction in thermal load which results to less heat being rejected to the surrounding air. Also, the condenser split (difference between the condenser and ambient temperatures) was seen to simultaneously increase with increase in COP; confirming that more heat was dissipated to the surrounding as the system tried to attain its design conditions. It was also observed that the COP increased as the condenser split decreased. This agrees with the findings of [14, 15, 16] that the performance of a heat exchanger improves when it is operated at low condenser split

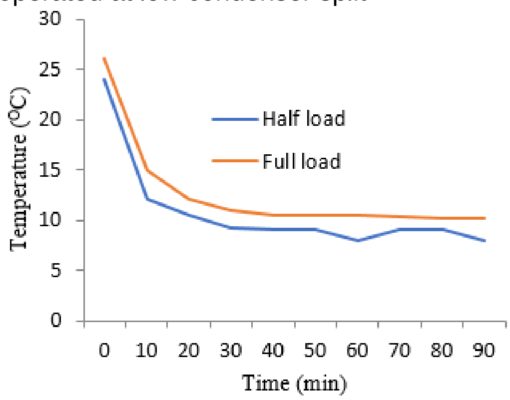


Figure 7: Cabinet Temperature variation with time in the cold-water chamber

Figure 7 shows the rate at which the Cabinet (target box) temperature varied with time in the cold-water chamber. It was noticed that the temperature of the cold water in the cold-water chamber reduced with time. The desired temperature (of 9 ^{o}C) was attained at about 65 minutes, as against the designed time of 60 minutes, when the machine is loaded with water for ice cubes production. This deviation could be attributed to the design assumptions used when procuring the compressors, as the distribution of design loads between the two compressors was done based on 50:50 sharing formula instead of the exact sizes of the thermal load distribution ratio of 55:45, obtained based on the calculated capacities of the compressors (1.8 kW and 1.3 kW), all at the expense of reducing the procurement costs.

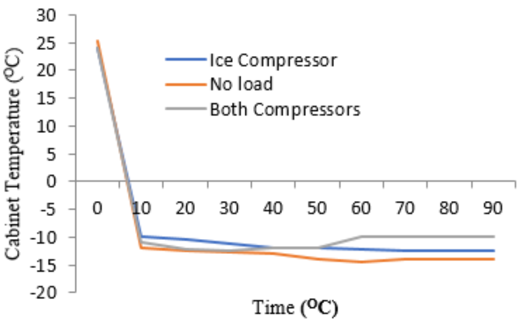


Figure 8: Cabinet Temperature variation with time in the ice water chamber

Figure 8 presents the variation of the cabinet temperature with time in the ice water chamber. The sequence of variation in temperature was observed to exhibit an initial drastic fall from 25.2 ^{o}C to -11 ^{o}C, within the first ten minutes of operation. This could be as a result of the combined refrigerating effect created by the two compressors due to the increased thermal loads of the refrigerating samples in the two target boxes. However, as the cooling proceeded further, it was noted that the rate at which the temperature drop became reduced until a constant temperature of -8 ^{o}C was reached; hence, no further reduction in temperature occurred. This indicates the trend towards subcooling. The freezing chamber attained its stabilized temperature after 100 minutes of uninterrupted operation. However, the subcooling temperature (of -8 ^{o}C) obtained in the freezing chamber was lower than the proposed temperature of -10 ^{o}C due to the heavy forced draught provided by the fan to enhance circulation of fresh air and reduce the prevailing ambient temperature around the condenser. It could be deduced from Figure 8 that the dedicated compressor for the ice section does most of the work of ice formation. However, significant work is also done by the second compressor as there was 20 % increase in the refrigerating effect compared to when only one compressor was in operation. Ice formation started at the mark of 30 minutes but complete freezing was achieved after 1 hour 30 minutes of refrigeration.

- **CONCLUSION**

The study focused on the development and performance evaluation of a multi-functional water and ice dispensing machine. It was developed to alleviate some of the challenges when the need for water in its existing three (3) phases in small quantity and within a short space of time arise.

The machine comprises three sections: cold water, hot water and ice chips. The design parameters for selecting, acquiring each component and assembling the machine were carried out with equations, findings of scholars, and relevant engineering standards to facilitate replacement of part(s) in case of any failure. Performance characteristics of the machine was determined using the thermodynamics properties taken with the attached instrumentations. Generally, the machine has a C.O.P of 1.15, a value viewed favourable, as it falls within the range of 1.00 - 2.00. The COP increased with increase in the condenser spilt but decrease in the work of compression, and had the highest performance when the two compressors were in operation.

The improvement in the COP and increase in the condenser split is beneficial as the machine would require little time to work to achieve reasonable refrigerating effect. Thus, the reduced work of compression would reduce the energy consumption costs, increase the machine reliability and reduce malfunctioning of the machine.

The acquisition of this device is a beneficial to both household and offices when these items are needed as it would reduce challenges associated with large sized appliances solely developed for the purpose.

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