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Design and Performance Evaluation of an Ice Block Making Machine

[1] A. Nasir, [2] A. Mohammed, [3] A. S. Adegoke, [4] H. T. Abdulkarim
[1,2,3,] Department of Mechanical Engineering, Federal University of Technology, Minna, Nigeria
[4] Department of Electrical/Electronic Technology, College of Education, Minna, Nigeria

ABSTRACT

The ambient temperature of tropical countries like Nigeria is as high as 40^{o}C during the dry season. This ultimately gives rise to increase in demand for ice which is used to reduce the temperature of water, soft-drinks as well as other uses. This increase in demand for ice block makes the design and construction of a machine which can be used for the production of ice within a short period of time in order to save energy and time imperative and a worthwhile venture. This paper presents the design and performance evaluation of a model ice block making machine that can freeze water quickly. In order to achieve quick freezing, certain design consideration, such as the quantity of water to be frozen, the choice of cooling system and the methodology of attaining the desired result. The methodology adopted involves increasing the area of heat transfer and employing a vapour compression refrigerating system to generate the refrigerating effect. The design analysis of the evaporator, compressor and condenser are presented in details. This paper also reports on the material selection, fabrication methods and the experimental procedures and results. A temperature of -14^{0}C was achieved in the freezing chamber and a freezing duration of 70 minutes for 1kg of ice block as test results.

Keywords: *Design, Refrigeration, Quick Freezing, Ice-Block.*

1. INTRODUCTION

The process of transforming water to ice by cooling below the freezing point of water 0^{0}Cis termed ice formation. Since ice formation is a cooling process, it could be said that ice formation is a heat transfer process that result to a phase change. For cooling to occur, the temperature gradient must be brought down by expediting the heat in the water and achieving a temperature in the range of -5^{0}C to -1^{0}C) in a process called refrigeration [1].The temperature changes in water as a result of reduction in enthalpy due to cooling are shown in the cooling curve in Figure 1.

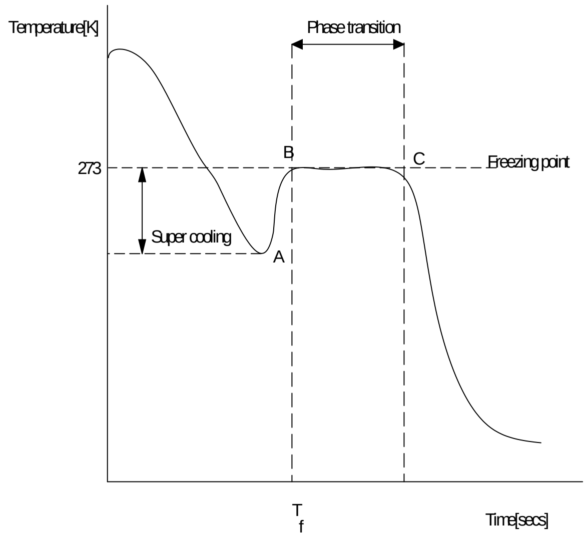


Fig 1: Cooling rate curve [2]

Point 'A' is the nucleation temperature, where a critical nucleus is formed in the liquid phase producing a localized warm up or negative cooling rate as shown on the cooling rate curve in Figure 1. Ice crystal growth will then propagate from this nucleus during small time interval 'A' – 'B'–'C'.Meryman1966[2,3] further explains that at point C, the cooling rate and accompanying heat flux in this small region will rapidly increase to point D to match the surrounding global heat flux called the critical cooling rate, which controls the extent or limit for ice crystals growth. The cooling rate which controls ice crystals size is some average value over a specific temperature change at any location. And when the crystal starts forming, it is certain that it will grow as long as the temperature of the entire system is below freezing [4]. However, Bald1990 [1] suggested that this approach will not produce an explicit relationship between cooling rate and ice crystals size. In some cases liquid is cooled sufficiently quicker so that nucleation does not occur and avoid ice formation. This process is called nitrification and results in an amorphous solid or glass. Muldrew1997 [4] explains that the liquid is in a detestable unit, it gets below a characteristic temperature, the glass transition temperature which is indicated by a sharp exotherm. Once below this temperature, the system is not merely a viscous liquid, but is a solid and it is in a stable thermodynamic state. Achieving nitrification with pure water requires very small water and incredibly fast cooling. It has been proposed experimentally that the heat conducted through the wall of a convectively cooled cavity equals to the heat given off by water for it to transform to ice. However, Weityet at, 1976[5] proposed that the rate of heat given off by a control volume immersed in a coolant equals to the convective heat entering the coolant. Freezing of water could be said to be a two stage process that is cooling the water to the freezing point and freezing the water for phase transformation to occur. However, analysis shows that the heat evolved in cooling is comparatively small to the heat evolved by the cooling and freezing process, as a result of latent heat of fusion, due to the energy that goes into hydrogen bond formation in the crystal. Muldew 1997 [4] analytically states that each water molecule is hydrogen bounded to four other neighboring molecules, each bond having an energy between 10 and 40 kJ/mol and concluded that 80 calories (350J) of energy is released by the transformation of 1gram of water at 0°Cto 80°C.

The applications of refrigeration include household (domestic) refrigerators, industrial freezers, cryogenics, air conditioning and heat pumps [3]. In developing tropical countries like Nigeria, the use of refrigeration as a domestic refrigerators are the most prevalent due to very high temperatures. Domestic refrigerators are used for chilling beverages at homes, offices, seminars, cocktail parties, general meetings, canteens and conferences. Oladunjoye [6] designed and constructed a two compartment freezing unit to enhance the availability of ice block. The need to have cold beverages during such meetings necessitated the design of an innovative refrigerated cooling table to be used for conference services [7, 8]. The demand for ice block has increased due to high temperature experienced in these tropical countries. Ice block making machine can be accomplished by variety of refrigerating effect generating methods. The time of freezing of the water is of considerable importance when designing an ice block making machine. The coefficient of performance (COP) and the energy used per system is also a vital factor in ice block machine design. In order to increase the heat transfer area, the coil was wound round the evaporator. Several designs were made at different time by different designers using adsorption system but they usually have a drawback of low C.O.P (0.12 to 0.23) [9].This paper presents the design and evaluation of an ice-block making machine using the principle of a vapour compression system.

2. VAPOUR COMPRESSION SYSTEM

The ice-making 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 a volatile liquid. Vapour compression refrigeration cycles specifically have two advantages which made it best suitable for the design of the ice making machine. First, it exploits the large thermal energy required to change a liquid to a vapour thus this will enhance the removal of high quantity of heat from water to cause a phase change. Second, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to the temperature of whatever is being cooled, this is a benefit because the farther the cooling fluids temperature is from that of the body being cooled, the higher the rate of heat transfer. Figure 2 shows the illustration of a typical vapour compression ice block machine.

Figure 3 shows the temperature-entropy and pressure-enthalpy diagram of the ice block making machine. Process 5-6 is the low temperature, low pressure heat source where heat is extracted from the water and achieved a refrigeration effect. In process 1-2, the low temperature, low pressure refrigerant vapour is then compressed isentropic ally in the compressor. Process 2-3 takes place at the condenser where the high pressure, high temperature vapour refrigerant undergoes the process of condensation isobarically and rejects the heat of condensation to an external heat sink. The high pressure liquid refrigerant passes through the filter/dryer and moisture and solid particles are trapped. In process 3-4, the high pressure refrigerant then passes through the counter flow heat exchanger where the warm refrigerant liquid from the condenser isobarically exchanges heat with the cool refrigerant vapour from the evaporator. Process 4-5 is the expansion valve where the high pressure liquid refrigerant undergoes isenthalpic expansion and regulates the flow of refrigerant to the evaporator. Due to the drop in pressure, some amount of liquid refrigerant flashes into vapour and the exit condition lies in the two-phase region as it enters into the ice making chamber (evaporator). At the evaporator, because of the low pressure of refrigerant, it boils at low temperature extracting the heat of vaporization isobarically and isothermally from the water as shown in process 5-6. Process 6-1 is observed before the refrigerant enters into the compressor. The low pressure, low temperature refrigerant passes through the counter flow heat exchanger where the cool refrigerant vapour from the evaporator isobarically exchanges heat with the warm refrigerant liquid from the condenser [7, 8].

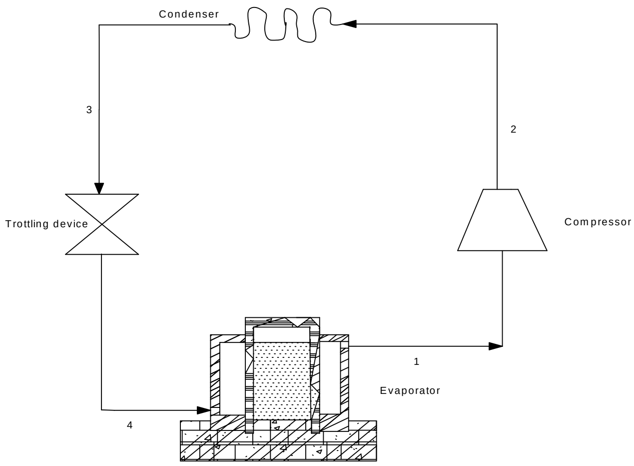
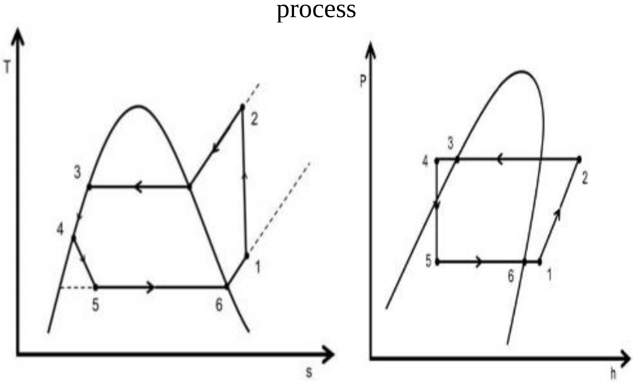


Fig 2: Schematic diagrams for the vapour compression process



(a) (b)

Fig 3: Vapour compression Cycle (a): Temperature-Entropy and (b): Pressure-Enthalpy

3. COMPONENT DESIGN AND CALCULATIONS

3.1 Evaporator Design Analysis

The purpose of the evaporator is to receive low pressure; low temperature fluid from the throttling device and to bring it in close thermal contact with load. The refrigerant takes up its latent heat of vaporization from the load and leaves the evaporator as a dry gas. The model evaporator for the project is to serve as the freezing chamber. The evaporator will be like a container in which the refrigerant is evaporated in copper tube closely coiled round the peripherals of the container. The form of the container is model to house water for sometimes before freezing will occur. Also, the material to be used for the tube must enhance a high rate of heat transfer so as to influence the rate of freezing. The evaporator was modeled based on the principle of heat exchanger, so for freezing to occur a certain quantity of heat must be evacuated from the water for change of phase from liquid to solid to occur and this heat evacuated must be able to pass through a heat transfer medium by conduction before the refrigerant can convectively sweep away the evacuated heat.

The mass flow rate of refrigerant circulating in the single stage system is the ratio of the refrigeration capacity to the refrigeration effect of the system[10].

$$\dot{m} = \frac{Q}{h_1 - h_4} \qquad (1)$$

The Coefficient of performance (COP) is given as:

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \qquad (2)$$

Where, $\dot{m}$ is the mass flow rate (kg/s), Q is the refrigeration capacity (kw), $h_1 - h_4$ is the refrigeration effect of refrigerant (kJ/kg).

The geometric form of the container is defined by its bore diameter, assuming that the volume of water to be frozen is V_o and is given as:

$$V_0 = \frac{\text{Mass of water}}{\text{Density of water}} \qquad (3)$$

Since the water being a liquid must fill the freezing cavity of a definite shape for the ice to take the form of the cavity. Then, the length and height of the cavity is equal to the proposed height of the ice block. Thus, the volume of the cavity is V_c

$$V_c = \frac{\text{®} d_1^2 h}{4} \qquad (4)$$

Equating equation (3) and (4), the diameter of the tube is given as;

$$d_1 = \left[\frac{4m}{\text{®}{}^\circ h}\right]^{1/2} \qquad (5)$$

Where, $d_1$is the diameter of the tube, his height of cavity, m is the mass of ice-block and ρ is the density of the ice-block.

For the production of ice, the total energy ($\dot{Q}_{ice}$) to be evacuated from water for it to be cooled down from approximately ambient temperature to water at zero degree, then cool down the ice to its final temperature is given as:

$$\dot{Q}_{ice} = \left[Cp_{water}(T_a - T_o) + h + Cp_{ice}(T_0 - T_1)\right]m \qquad (6)$$

Where, Cp_{water} is the heat capacity of water, Cp_{ice} is the heat capacity of ice, T_a is ambient temperature, T_o is the freezing temperature, T_1 is the final temperature, h is the Enthalpy to freeze water.

The total heat load to be evaporated ($\dot{Q}_E$), is given as [7, 8]:

$$\dot{Q}_E = \dot{Q}_L + \dot{Q}_{SP} + \dot{Q}_U \qquad (7)$$

$$\dot{Q}_L = \frac{A\Delta T}{R_T} \qquad (8)$$

$$\dot{Q}_S = 15\%\ \dot{Q}_U \qquad (9)$$

$$\dot{Q}_U = \dot{Q}_{ice} \qquad (10)$$

Where, $\dot{Q}_S$ is the usage load as a result of freezing, $\dot{Q}_L$ is the load due to leakages, $\dot{Q}_S$ is the supplementary load incorporated to account for factor of safety, A is the proposed total surface area of evaporation, ΔT (T_3–T_1) is the temperature difference between inlet and exit, R_T is the total resistance provided for leakages.

The material resistance is given as:

$$R = \frac{\text{Thickness (x)}}{\text{thermal conductivity of material (K)}} \qquad (11)$$

The length of the evaporator pipe coiled round the freezing tube can be obtained using

$$Q_{Evap} = AU\Delta T \qquad (12)$$

Where, A is the surface area of the evaporator, U is the overall heat transfer coefficient and ΔT is the temperature difference. Since the evaporator is well lagged, the tendency of heat escaping to the surroundings could be neglected. Thus, the overall heat transfer coefficient can be expressed based on the inside area as

$$A_i = 2 \text{®} r_i L \quad (13)$$

$$U^{-1} = \frac{r_i}{\bar{h} r_o} + \frac{r_i}{K_{Copp}} In\left({}^{r_o}\!/_{r_i} \right) \quad (14)$$

Thermal conductivity of copper is obtainable from property table. However, the inside heat transfer coefficient $\bar{h}$ is obtainable as

$$\bar{h} = \frac{NuK_R}{d_i} \quad (15)$$

$$Nu = 0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{0.4} \quad (16)$$

Where, Nu is the Nusselt number, Pr is the Prandtl number, Re is Reynolds number. The value of K_R is obtainable from property table at the evaporating temperature.

The length of evaporating pipe is given as;

$$L = \frac{Q_{evap}}{\text{®} d_i U \Delta T} \quad (17)$$

The Figure below shows the cross section of the freezing chamber.

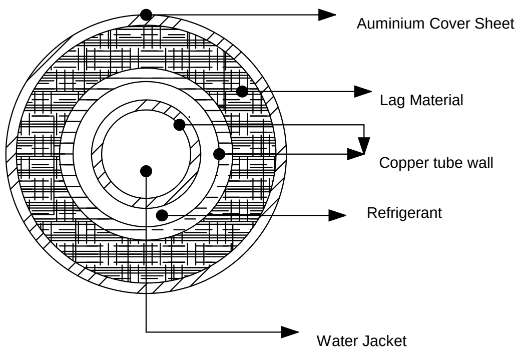


Fig 4: Cross section of the freezing chamber

The total resistance was offered by the copper tube walls, the lag material and the sheet cover is given as;

$$R_T = R_{AS} + R_{lag} + Rcu \quad (18)$$

$$R_{AS} = \frac{X_{AS}}{K_{AS}} \quad (19)$$

$$R_{lag} = \frac{X_{lag}}{K_{lag}} \quad (20)$$

$$Rcu = \frac{X_{AL}}{K_{AL}} \quad (21)$$

The values obtained from calculations of equation (1) to equation (21) are tabulated in Table 1 below.

Table 1: Evaporator design values

Name	Values/Units
Freezing chamber's height, h	0.25m
Freezing chamber's diameter, d	0.3m
Thickness of sheet, t	0.0015m
Length of evaporating pipe, L	1.57m
Mass flow rate, ṁ	0.00194Kg/s
Reynolds number, Re	1409.27
Prandtl number, Pr	4
Nusselt number, Nu	13.24
Overall heat transfer coefficient, U	196.48 W/m^2K
Leakage load, Q_L	86.15W
Usage load, Q_U	123W
Supplementary load, Q_{SP}	18.5W
Coefficient of performance, COP	3.36

3.2 Compressor Design Analysis

As the refrigerant leaves the evaporator either as a saturated or weak super saturated vapour, it enters the compressor where it becomes compressed. The compressor requires energy and carries out work. This work is transferred to the refrigerant vapour and is called the compression input. Several assumptions are incorporated into the modeling procedure: the modeled compressor cycle is only an approximation of the real compressor cycle; the compression and expansion in the compressor cycle are isentropic processes with equal and constant isentropic exponents; the isentropic exponent is dependent on the refrigerant type; the oil has negligible effects on refrigerant properties and compressor operation; there are isenthalpic pressure drops at the suction and discharge valves.

Due to the re-expansion of the refrigerant vapor in the clearance volume, the mass flow rate of the compressor refrigerant is a decreasing function of the pressure ratio:

$$\dot{m}_r = \frac{PD}{v_{suc}}\left[1 + C - C\left(\frac{P_{dis}}{P_{suc}}\right)^{\frac{1}{\gamma}}\right] \quad (22)$$

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Where, $\dot{m}_r$ is the refrigerant mass flow rate (kg/s), PD is piston displacement (m^3/s), v_{suc} is the specific volume at suction state (m^3/kg), C is clearance factor, γ is the isentropic exponent, P_{suc} and P_{dis} are the suction and discharge pressure respectively (kPa).

The compressor has a power requirement expressed mathematically as;

$$P = \dot{m}(h_2 - h_1) \tag{23}$$

Theoretical volume of vapour to be a handle

$$V = mV_g \tag{24}$$

Where, P is the compressor power and V_g is the specific volume at compressor suction.

The compressor values obtained from calculations of equation (22) to equation (24) are tabulated in Table 2 below.

Table 2: Compressor design values

Design	Values/Units
Power capacity, P	67.9W
Theoretical volume, v	1.49 x 10^{-4} m^3

3.3 Condenser Design Analysis

The condenser is a heat exchanger in which the compressed refrigerant vapour is liquefied due to extraction of its latent heat by a cooling medium. The condenser is being cooled by natural air. The natural convection is not sufficient to attain the heat transfer rate required on their-side of the condenser used in vapour compression systems. Therefore, a fan is employed with the limit of air velocity range as 0.91m/s<Vac<5.3 m/s. The design of the condenser required for the refrigerant heat rejection is described with respect to the required heat duty of condenser (Q_{cond}) using copper pipe with inner diameter "di" and outer diameter "do".

The heat head of condenser is the heat amount to be removed during condensation and it could be expressed mathematically is

$$Q_{cond} = \dot{m}(h_2 - h_3) \tag{25}$$

Where, $\dot{m}$ is the massflow rate of refrigerant entering the condenser(kg/s), h_2 is the enthalpy of compressed vapour entering condenser (kg/kj) and h_3 is the enthalpy of liquid leaving condenser (kg/kj) [10].

For an air–cooled condenser, the quantity of heat is given out is express as:

$$Q = AU\Delta t_m \tag{26}$$

$$U^{-1} = \left[\frac{d_0}{d_i}\right]\left[\frac{1}{h_i}\right] + \left[\frac{1}{2k}\right] d_o \times In\left[\frac{d_o}{d_1}\right] + \frac{1}{h_o} \tag{27}$$

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{In\left(\Delta t_1 / \Delta t_2\right)} \tag{28}$$

Where, U is the overall heat transfer coefficient of the condenser (W/m^2K), Δt_1 and Δt_2 is the maximum and minimum temperature difference respectively between the cooling fluid and condenser refrigerant(^{0}C) and K is the thermal conductivity of copper at 35^{0}C.

The fin coefficient for Refrigerant side is calculated as

$$h_i = \frac{N_{u_i} \times K}{d_i} \tag{29}$$

$$R_e = \frac{\text{°} v d_i}{\text{«}} \tag{30}$$

But

$$N_u = \frac{\left(f/8\right)\left(R_e - 1000\right)\Pr}{1 + 12.7\sqrt{f/8}\left(\Pr^{\frac{2}{3}} - 1\right)} \tag{31}$$

For $2300 \leq R_e \leq 5x10^6$

$$f = \frac{1}{\left(1.82\log_{10}\mathrm{Re} - 1.64\right)^2} \tag{32}$$

Where, V is the velocity of fluid flow, d_i is the inner diameter of pipe, ° is density or refrigerant at 40^{0}C, « is the viscosity of refrigerant at 40^{0}C, K_r is the thermal conductivity of refrigerant at mean temperature of 40^{0}C, N_u is the Nusselt number [10].

At the air side of the condenser, the following empirical formula for staggered tube rows in tube bundles.

$$N_{u_{do}} = \frac{\bar{h} \times d_o}{K} \tag{33}$$

$$R_e = \frac{\text{°} \times v_{air\,max} \times d_o}{\text{«}} \tag{34}$$

$$N_{u_{do}} = \Pr^{0.36}\left(\Pr/\Pr w\right)^n fn\left(R_{edo}\right) \tag{35}$$

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

$$\overline{N}_{u_{do}} = \Pr^{0.36} . fn(\mathrm{Re}_{do}) \quad (36)$$

$$fn(\mathrm{Re}_{do}) = 0.4\,\mathrm{Re}_{do}^{\;0.6} \quad (37)$$

$$\overline{N}_{u_{do}} = \Pr^{0.36} .0.4\,\mathrm{Re}_{do}^{\;0.6} \quad (38)$$

$$\overline{h}_o = \frac{\overline{N}_{u_{do}} \times K}{d_o} \quad (39)$$

Where, V_{max} is the air velocity, K is the thermal conductivity of air, d_o is the outside diameter and ³ is the kinematic viscosity of air.

The dimensions of the condenser is obtainable from

$$Q = AU\Delta t_m \quad (40)$$

The total condensing area is given as

$$A = \frac{Q}{U\Delta t_m} \quad (41)$$

The length is given as:

$$L = \frac{A}{®d_i} \quad (42)$$

The condenser values obtained from calculations of equation (23) to equation (42) are tabulated in Table 3 below.

Table 3: Condenser design values

Name	Values/Units
Length of condenser pipe, L	10m
Mass flow rate, ṁ	0.00194Kg/s
Reynolds number, Re	3432.16
Prandtl number, Pr	3.5
Nusselt number, Nu	21
Overall heat transfer coefficient, U	272.73 W/m^2K
Heat head of condenser, $\dot{Q}_{cond}$	0.296 KW
Heat rejection ratio	1.3
Coefficient of performance, COP	3.36

3.4 Capillary Tube

The principle of operation of the capillary tube is the flow resistance caused by a long, narrow tube, throttling the refrigerant pressure. Pressure falls gradually as the liquid flows through the tube, until it starts to evaporate in the tube. The capillary tube is made of seamless tubing and has a small accurate inside diameter. The amount of refrigerant in the system was carefully fixed, since all the liquid will travel to the evaporator during the off-cycle and will cause a liquid stroke when starting the compressor, if that amount is over estimated.

3.5 Accumulators

The accumulator serves as the absorber and act like an initial surge of liquid from the evaporator when compressor starts to work. The liquid vaporize into the accumulator and return to the compressor vapour. It also serves the function of expending the return of oil to the compressor.

3.6 Strainer Drier

To eliminate impurities from refrigerants, drier traps small quantity of dirt from the machine. The drier used in this work is the capsule charge with solid desiccant such as silica gel, activated charcoal or zeolite(molecular sieve) and is located in the liquid line ahead of the capillary tube. The capsules must have strainers to prevent loss of the drying agent into the circuit and to form an effective strainer drier to also prevent the valve orifice from damages by fine debris.

4. FABRICATION

4.1 Material Selection

The material selections are based on the material properties, availability, cost and ease of fabrication. The components to which material selection process are required include: the freezing chamber, the refrigeration system, the ice conveying mechanism, the stands and support frame.

The materials selected for the freezing chamber is aluminium. Aluminum has a great level of heat transfer across its walls, high thermal conductivity, low cost, ease of fabrication and availability when compared to other materials such as copper and stainless steel. In selecting the refrigeration system, the absorption and vapor compression systems were considered. The absorption cycle requires heat input from a gas burner and a liquid pump with an electrical power input to circulate the working fluid. Vapor compression cycle was chosen for the present study due to fewer components required and lower complexity. This is because the key components in the system serve as the driving force for the entire cycle. Though R-12 has an adverse effect on the depletion of ozone layers when compared toR134a, R-12 was selected due to low cost. The ice conveying mechanism, stands and support frames materials was constrained by weld ability, workability and strength. As a result mild steel was used due to its good weld ability and workability with its fair strength when compared to high carbon steels.

4.2 Detailed Construction

a. Freezing Chamber

Sand casting was used due to its simplicity and versatility in aluminum casting. The water jacket was then casted. The cast was then machined using a lathe machine. After machining to the required thickness, a0.25 inches pipe was coiled firmly round the machined aluminum water jacket. In order to facilitate easy removal of the iced block, the inside bore of the tube was tapered at 5^0C. However, for the water to be housed by the system prior to freezing a base was machined from aluminum cast in such a way to prevent leakages. Water inlet hole and the push rod passage hole were drilled on the tube. Hence, both the tube and jacket were then tight fitted and welded together with aluminum electrodes. The freezing chamber was then lagged with carbon dust cemented to the walls of the jacket with mortar. Aluminum sheet was then used in covering the lag by riveting the sheet cover to the fused lag.

b. Stands and Support Frame

The stands and support frame work comprises of angle iron (1.5 and 3 inches) and hollow pipe steel rigidly welded together to form solid and firm stand and support frame work. For ease of mobility, two rubber wheels were fitted to two of the legs.

c. Vapour Compression System

The vapour compression system was installed at the base with the condenser fan positioned in between the compressor and the condenser. 0.25 inches pipe was used as the refrigerant tubing. The compressor, condenser, capillary tube and the freezing chamber were connected together by brazing the 0.25 inches pipe to the inlet and exit points of the pipe network. The required amount of refrigerant was then introducing to the system by charging it. The amount of charge depends on the size of the system, the length of the line, type of refrigerant and the operating temperature.

d. Ice Conveying Mechanism

A power screw was constructed to convey the ice out of the freezing chamber. Four chains were attached to the screw arrangement and these chains were the attached to the base lid. A push rod was attached to the screw arrangement to create a push effect on the ice as the screw is been screwed down

5. PERFORMANCE EVALUATION

5.1 Tests for Leaks

The system was tested for leaks using soapy water solution. The soapy water was applied at all brazed nodes and along the entire condenser line. Detergents have the desired properties of foaming with bubbles in the presence of leaking gas. From the test conducted, the system was confirmed free from leaks.

5.2 Test for Freezing Chamber Temperature

The test was conducted using a digital thermometer and stop watch. The temperature variation with time in the freezing chamber was studied by measuring the temperature of the walls of the freezing chamber at regular intervals of 10 minutes. The time at which the freezing chamber will attain its stabilized temperature during operation and the minimum temperatures attained in the freezing chamber are noted and recorded. The experiment was repeated three times and in all cases the values were recorded. Figure 5 shows the plots of the average values of temperatures against time. The error represents the maximum and minimum values at each interval.

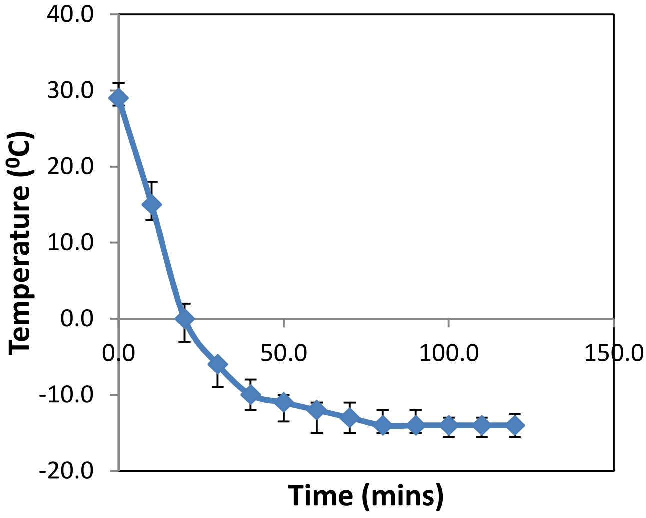

Fig 5: Temperature variations with Time in the Freezing Chamber

5.3 Discussions

The sequence of variation of the temperatures in the freezing chamber with time was observed to exhibit an initial drastic fall from 29^0C to 15^0C due to the sudden cooling effect as result of the vast temperature difference. However, as the cooling proceed further, it was noted that the temperature drop became reduced until a constant temperature was reached. Hence, no further reduction in temperature occurred. This indicates the trend towards the saturation temperature. The freezing chamber attains its stabilized temperature after 80 minutes of uninterrupted operation. This is the saturation temperature as there was no further decrease in the temperature as shown in Figure 5. However, the saturation temperature,-14^0C obtained in the freezing chamber was lower than the proposed temperature of -8^0C due to the heavy forced draught provided by the fan enhancing adequate cooling of the condenser. The freezing time was 1 hour 10 minutes which extends more than the proposed time of freezing by 10 minutes. This could have resulted from non-compactness of the lagging material as desired but the error is still within a correctable range. The result obtained can still be considered as quick freezing since a normal quick takes place in 2 hours or less [12].

6. CONCLUSION

An ice-block making machine has been designed, constructed and tested. The deigned was based on the principle of vapour compression, and the material for the construction was selected based on the material properties, availability, cost and ease of fabrication. The result from the experimental testing shows the production of 1kg of ice in 70 minutes which is a huge success. This shows that the machine is economically viable and increasing the surface area of heat transfer in a freezing compartment will reduce the freezing duration.

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