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DESIGN, CONSTRUCTION AND PERFORMANCE EVALUATION OF A COMBINED REFRIGERATOR AND FOOD WARMER

this is a heat recovery system.

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ABSTRACT

The design, construction and performance evaluation of a combined refrigerator and food warmer based on vapour compression refrigeration cycle has been carried out in this work. The research has been conducted with the motive of recoverying the condenser heat for food warming. In this way the system has been used in performing the dual purpose of a refrigerator and a heat pump. The system comprises a conventional refrigerator utilizing HFC-134a as refrigerant and the warmer mounted on top of the refrigerator. The insulation consists of polyurethane foam for the cold chamber and fibreglass for the warmer. Relevant thermodynamic relations were used to design the important parameters of the system such as refrigeration effect and capacity, compressor size, lengths of evaporator, capillary and condenser tubes, and thicknesses of the walls of the refrigerator and warmer compartments. In the experimental tests conducted, temperatures of $-12^{0}C$ *and* $69^{0}C$ *were obtained in the unloaded refrigerator and warmer respectively. About two and a half litres of water at an initial temperature of* $27.8^{0}C$ *became frozen within a period of approximately 120 minutes in the refrigerator. The same quantity of water at the same initial temperature was heated to* $50^{0}C$ *in about 150 minutes.*

Keywords: Refrigerator, HFC-134a, Fibreglass, Warmer, Polyurethane Foam, Compressor.

INTRODUCTION

A device that continues to take the same substance, called refrigerant, through the same cycle of processes while exchanging heat with the environment and expending certain quantity of work is either a conventional refrigerator or a heat pump. In the refrigerator the useful heat exchange is the one at the evaporator, the heat rejected at the condenser is thrown away. And in the heat pump, the heat rejected at the condenser is the only useful heat exchange; the heat extracted from the environment at the evaporator equivalent is not applied usefully. In this way the full benefit of either of the systems is not realized. In cold climates refrigerators and heat pumps are in common use. In these places heat pumps are used for home heating, domestic hot water heating and possibly for drying, while the refrigerator provides cold space for food storage. In hot climates only the refrigerator is commonly used, home heating and domestic hot water heating are undesirable for the greater part of the year. If a device like the heat pump is to find use in tropical regions it may not be in the concept described or for the purposes mentioned earlier. The use of the combined refrigerator and the heat pump configured as a food warmer provides a platform for harnessing the full benefits of the refrigeration system, particularly in hot climates. In this arrangement the warmer is more of a heat recovery system which utilizes the heat rejected at the condenser that would, otherwise, have been thrown away into the environment. By this arrangement the conventional refrigerator is now made to perform the dual functions of cold and hot preservation of food Previous works executed on refrigeration are mostly on the design and fabrication of refrigerators, heat-pumps and air-conditioners. Only a handful of works have been done on the combination system. These include the work carried out by H. E. Berman & co. which discloses a “Crockpot” type combined refrigerator and cooker, for which they were issued with the U. S. Pat. No. 2,504,794. The U. S. Pat. No. 3,682,643 was also issued to L. H. Foster for devising a means for freezing and/or cooking food in the same compartment. D. D. Roads & co. were also issued with U. S. Patent No. 3,516,485 for their work on a food container for selectively freezing or heating food. However, these patents make use of microprocessors and thermoelectric refrigeration, and not a vapour compression refrigeration cycle employing just a single compressor (patentsonline.com, ©2008). Works on the combination system have also been carried out in which the heat from the condenser is used for hot water heating or for drying. The present work investigates the above combination system in which the heat from the condenser is trapped in a compartment, the warmer, for use in the moderate temperature preservation of cooked food items. The warmer unit functions like a heat recovery system, requiring no additional work input on the compressor side of the parent refrigerator as only a short length of the heating tube is required in the warmer. The advantage of this system over the ordinary warmer dishes is that the desired warmer temperature can be set and maintained for as long as the refrigerator runs. And when compared with the conventional domestic food ovens, this warmer does not impose additional power or energy consumption on the combination system.

Design Description, Analysis and Calculation

Design Description

The combined refrigerator and warmer system consists of a conventional vapour compression refrigerating unit, on top of which is mounted the warmer unit. The two are separated by a 45mm thick fibreglass insulation. The refrigerator enclosure is located below that of the warmer. The refrigerator wall is insulated with polyurethane foam while the wall of the warmer is well packed with fibreglass insulation. These insulation materials were chosen for their low thermal conductivities (Dossat, 1978) to suit the respective functions desired. The refrigerator interior is made of thin aluminium sheets while that of the oven is covered in galvanised steel sheet. The compressor is suspended just behind the oven chamber and mounted firmly on the base plate provided. This reduces the length of the condenser pipe needed to be run from the compressor discharge to the warmer. The condenser pipe is first laid into a few coils inside the warmer as the heating element before it is passed out and made into another coil for further heat discharge into the environment. A flow regulator is provided for the refrigerant before it is discharged into the warmer and a bypass pipe, is also provided running from the regulator joint, bypassing the warmer and rejoining the condenser pipe at the other side of the warmer. In this way regulation of warmer temperature is achieved by opening the regulator only to the extent of delivering the quantity of refrigerant through the warmer that would give the desired temperature inside the warmer.

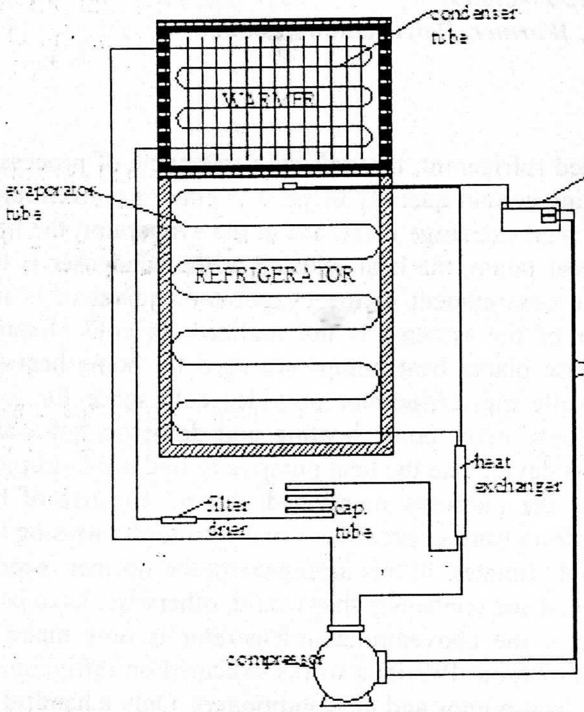

Figure 1: The Combined Refrigerator and Oven.

Design Analysis and Calculations

Design Specifications

Based on the choice of a domestic size combined refrigerator and warmer, the following parameters were selected for the design:

Refrigerator external dimensions: 450 × 460 × 750 mm
Warmer external dimensions: 450 × 460 × 250 mm
Thickness of polyurethane foam insulation wall lining for the refrigerator: 35 mm
Thickness of fibreglass insulation wall lining for warmer: 45 mm
Thickness of fibreglass insulation between refrigerator and warmer: 50 mm
Thickness of galvanised steel sheet: 2 mm

Thickness of aluminium sheet: 0.4 mm
Thermal conductivity of polyurethane foam: 0.025 W/m.K
Thermal conductivity of fibreglass: 0.036 W/m.K
Thermal conductivity of galvanised steel: 51.9 W/m.K
Thermal conductivity of aluminium: 250 W/m.K
Design Evaporator temperature: -15°C
Design warmer temperature: 60°C
Average ambient temperature (Minna): 34°C
Refrigerant used: HFC-134a

Analysis of Refrigeration and Heating Loads

The various components of the refrigeration loads are: the transmission load through the walls structure of the evaporator cabinet, the product load as a result of the items kept in the evaporator, infiltration load as a result of outside air escaping into the evaporator by various means, and miscellaneous. However, only the transmission load was calculated for, other loads were expressed as factors of the transmission load. The transmission load is via the various walls of the evaporator cabinet and for a typical composite wall is expressed by the heat equation (Osore, 2005),

$$Q = UA\Delta T \qquad 1$$

where, U is the overall heat transfer coefficient for the composite wall and ΔT is the temperature difference between the evaporator cabinet and the ambient air. The overall heat transfer coefficient is computed through the overall resistance and the individual resistances of the various components of the wall (Eastop and McConkey, 1995). Approximate values were selected for the convective coefficients of the inside and external walls of the cabinet. Similar procedure was followed for the warmer cabinet. The total heat gain for the evaporator cabinet was obtained as, Q = 61W, while the heat loss was obtained for the warmer as, Q = 23W.

Analysis of Refrigeration Cycle

Using the saturation properties of R134a at the design evaporator and condenser temperatures and assuming that the refrigerant leaves the condenser as a saturated liquid, and enters the compressor as a saturated gas, refrigeration cycle analysis was performed (Rogers and Mayhew, 1957) and (Osore, 2000) to yield the following values for the refrigeration effect, mass flow rate of refrigerant, and compressor power input:

Ref. effect, R.E = 99.51 kJ/kg; Mass flow rate, □ = 0.00061 kg/s; Power input, P = 29.1 W.

Size of Evaporator, Condenser, and Capillary Tubes

Noting that the heat gain into the evaporator expressed by equation (1) is also the heat transfer rate across the walls of the evaporator tube, the heat equation is expressed via the various thermal resistances for the evaporator tube walls to obtain an expression for the length of the tube as

$$L_{Evap} = \frac{Q\left\{\frac{1}{r_o h_o} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{1}{r_i h_i}\right\}}{2\pi\Delta T} \qquad 2$$

ΔT in this case is the temperature difference between the evaporator cabinet and the refrigerant fluid flowing inside the tube. For an effective heat transfer process a temperature difference of 7°C is adequate. The convective heat transfer coefficient for the internal walls, h_i, was calculated from the expression

$$h_i = Nu_D \frac{k}{d_i} \qquad 3$$

where, Nu_D is the Nusselt no. based on the pipe diameter, and k is the thermal conductivity of the fluid.

The regime of flow of refrigerant in the evaporator pipe was determined to be laminar and the appropriate expression for the Nusselt no. expressed as follows was used to evaluate Nu_D as

$$Nu_D = 0.664\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3} \qquad 4$$

The other parameters relating to R-134a were selected from the table of thermo-physical properties of refrigerants (Lienhard and Lienhard. 2006).

For the condenser the flow was determined to be turbulent and Nu_D was evaluated using

$$Nu_D = \frac{\frac{f}{8}(\mathrm{Re}_D - 1000)\mathrm{Pr}}{\left\{1 + 12.7\sqrt{\left(\frac{f}{8}\right)}\left(\mathrm{Pr}^{2/3} - 1\right)\right\}} \qquad 5$$

where, the friction factor, f, assuming a smooth pipe surface, is expressed as

$$f = \frac{1}{[1.82\log_{10}\mathrm{Re}_D - 1.64]^2} \qquad 6$$

For the capillary tube the compressible flow equation expressed as follows was used in determining the tube length:

$$\tfrac{1}{2}\left(P^2 - P_1^2\right) - \frac{V_1^2 P_1^2 \log_e \frac{P}{P_1}}{RT} = -\frac{fP_1^2 V_1^2 (x - x_1)}{2mRT} \qquad 7$$

where, $(x - x_1)$ is the length of the tube, P & P_1 are the pressures at the inlet and outlet of the tube respectively, and m = d/4 is the hydraulic mean depth. Again property values were selected from the table of thermo-physical properties of R-134a.

Contribution of Friction Losses to Compressor Power Input

The contribution of friction losses in the various pipes to the power requirement at the compressor was determined by first determining the friction loss using the Darcy-Weisback equation (Douglas et al, 2001) as

$$h_f = \frac{4fl.V^2}{D\,2g} \qquad 8$$

For the laminar flow in the evaporator the friction factor, f, is expressed as

$$f = \frac{64}{\mathrm{Re}_D} \qquad 9$$

The friction power requirement was evaluated using the expression

$$P = \rho g h_f Q \qquad 10$$

The evaluated friction power was determined to be negligible compared to the compressor power input determined by the cycle analysis.

Allowing for product, infiltration and other miscellaneous loads, a factor of 2.75 was applied to the calculated compressor power to obtain, Compressor Power, P = 80.05 W, or $\frac{1}{8}hp$.

Performance Testing of the Combined Refrigerator and Warmer

The performance of the combined refrigerator and warmer system was examined to determine the following:

- the actual cooling and heating rates in the refrigerator and warmer compartment respectively in the unloaded and loaded mode.
- the actual minimum temperature attainable in the evaporator and the actual maximum attainable in the warmer.

Test Procedures

Evaporator and Warmer Compartment Unloaded

The temperature of the ambient air and the initial temperature of the two compartments, evaporator and warmer, were measured using a digital thermometer. Two such thermometers were each placed in the evaporator and the warmer. The system was powered on and allowed to stabilize. The temperature readings in each of the compartment were recorded after every 15 minutes until the temperature becomes constant over a period of time.

Evaporator and Warmer in the Loaded Condition

The same preliminary procedure was used to measure the initial temperatures of the two compartments and that of the surroundings. The same preliminary procedure for measuring the performance of the unloaded system was followed for the system in the loaded mode. Thereafter, two separate volumes of water, each

235cm^3, were measured in a container using a weighing balance. Two similar mercury-in-glass thermometers were each immersed into the samples of water to record initial temperature values before they were placed in the warmer and refrigerator separately. The system was then powered on and the temperature readings of the water samples were taken after every 30 minutes, until the water in the warmer attains a constant temperature value and the water in the refrigerator freezes.

RESULTS AND DISCUSSION

The graphical illustration of the results obtained are as presented in figures 2 and 3. Figure 2 presents results of the experimental testing of the combined system when it was unloaded and figure 3 presents same for the combined system in the loaded condition.

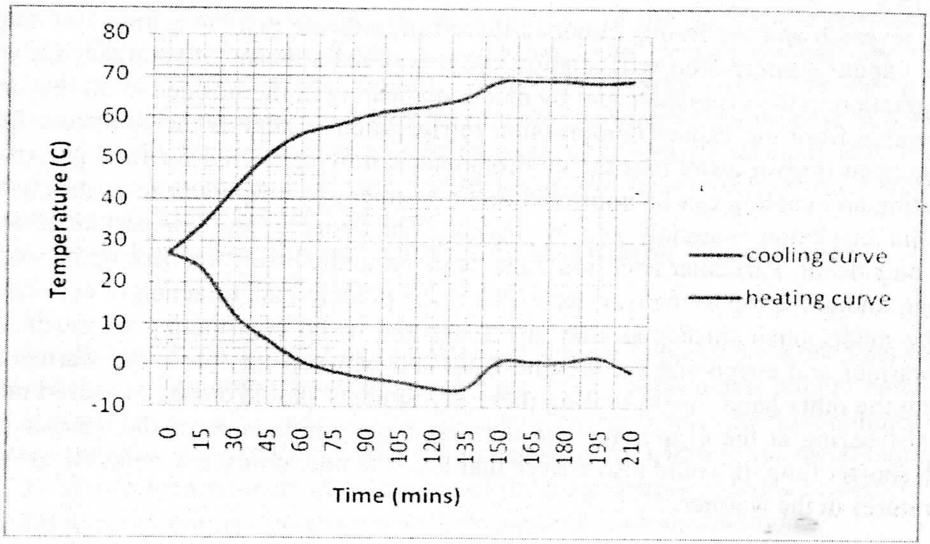

Fig. 2: Temperature-time history for the unloaded system

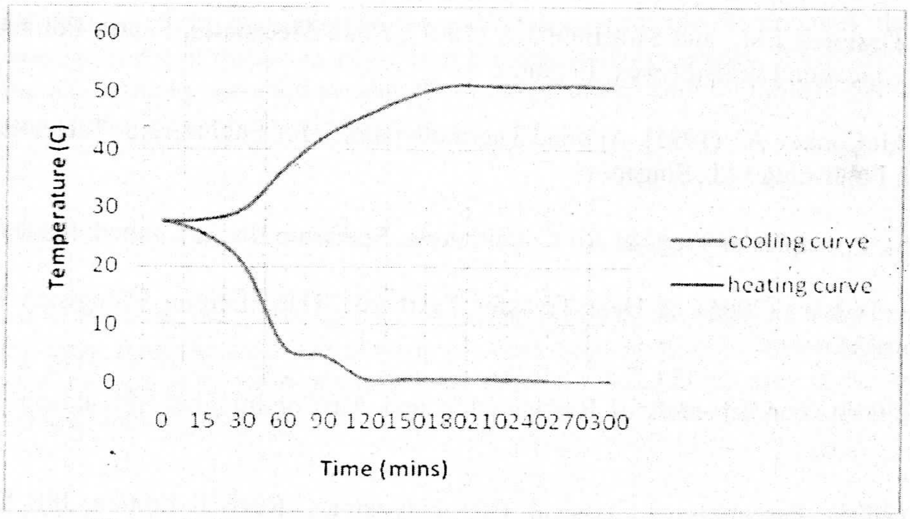

Figure 3: Temperature-time history for the loaded system

It would be seen from figure 2 that the temperature of the unloaded evaporator reduced drastically from 27^{0}C to below the freezing point (4°C) to about 0^{0}C in the first 60 minutes of operation. After about 135 minutes a temperature of -5.5^{0}C was recorded. However, for the first 60 minutes the warmer temperature increases from 27^{0}C to 56^{0}C, and at 150 minutes a temperature of 69^{0}C was attained within the heating enclosure. The temperature in the warmer remains constant at around 69^{0}C up to 210 minutes for which the experiment lasted. At 150 minutes, the temperature in the refrigerator rises back to 1^{0}C due to the increase in the temperature of the ambient environment in which the system was operating before finally coming down again to -1.5^{0}C at the end of 210 minutes. When about 2 litres of water was kept inside the refrigerator, the temperature of the water decreased from 27.8^{0}C to 6^{0}C in 60 minutes (see figure 3); and after about 75 minutes freezing began. In the warmer, the temperature of the same quantity of water rose from 27.8^{0}C to

37^{0}C in the first 60 minutes and to about 50°C in 150 minutes. The water temperature rose to about 52°C after about 180 minutes of heating and remained constant at the temperature till the end of the experiment after 300 minutes. However, full frost water condition at -0.5°C was attained in the evaporator at the end of 240 minutes. This condition was maintained till the end of the experiment after 300 minutes. The length of time elapsed before freezing occurred at the evaporator and before the attainment of the maximum water temperature in the warmer could be attributed to the following reasons. First it may be due to inadequate thickness of insulation materials around the compartments, particularly that between the evaporator and the warmer, and that at the top of the warmer. There might also have been infiltration of outside air into compartments through the doors.

CONCLUSIONS

The foregoing research and the results obtained therefrom indicate that the refrigerator and warmer system employing the vapour compression refrigeration cycle presents a means of simultaneously preserving food items by refrigeration in the evaporator and by moderate heating in the warmer. With this system, maximum benefit is derivable from the vapour compression refrigeration system as the condenser heat which would otherwise have been thrown away into the environment is now harnessed for food preservation. However, the rate of heating and cooling can be improved in the warmer and evaporator respectively by increasing the thickness of the insulation materials and by making the doors of the compartments airtight to reduce infiltration of outside air. Particular attention must be given to the insulation parking between the evaporator and the warmer, and at the top of the warmer which is the predominant direction of convective heat transfer. These must be made much thicker as they are suspected to be contributing so much to heat exchange between the warmer and evaporator on the one hand and between the top of the warmer and the ambient environment on the other hand. Incorporating these adjustments should ensure increased rate of cooling and heating so that freezing at the evaporator and the maximum temperature of the warmer could be attained within a much shorter time. It would also ensure that lower temperatures are attainable in the evaporator and higher temperatures in the warmer.

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