Performance of an Improved Household Refrigerator/Freezer

D. O. Ariyo
Department of Mechanical Engineering, School of Engineering, Federal Polytechnic, Offa, Nigeria

Y. L. Shuaib-Babata
Department of Mechanical Engineering, School of Engineering, The Federal Polytechnic, Ado-Ekiti, Ekiti State, Nigeria

M. A. Azeez
Department of Mechanical Engineering, School of Engineering, Federal Polytechnic, Offa, Kwara State, Nigeria

Abstract--**A household refrigerator/freezer based on modified two-stage cycle is presented. The freezer cycle operating temperatures are –20°C for the evaporator and 40°C for the condenser while refrigerator (fresh food) evaporator temperature is –5°C with the same condenser temperature of 40°C. The degree of subcooling is 5K while degree of superheating is 7K and the refrigerating capacity of the system is 380W. A pipe with sudden enlargement was incorporated between the freezer evaporator and the flash chamber, which eliminates the use of compressor in the first stage. The performance of this modified refrigeration system was compared with household refrigeration systems based on single and dual cycles. Maximum energy savings over single cycle refrigeration system is 49.9% while that over dual cycle is 39.6%. The maximum coefficient of performance computed for the modified two-stage cycle is 6.5 while those of single and dual cycles are 3.3 and 5.0 respectively.**

Keywords: Modified Two-Stage Cycle, Single Cycle, Dual Cycle, Energy Savings, Coefficient of Performance.

I. INTRODUCTION

A household single stage vapour compression refrigeration system has one hermetic compressor, a condenser, one throttling device, and two evaporators: one in the freezer compartment and another in the refrigerator (fresh food) compartment. Refrigeration for both fresh food and freezer compartments is provided by a single vapour compression cycle, which operates at the freezer evaporating saturation temperature.

A dual vapour compression refrigeration system has two refrigeration cycles; one for the refrigerator which provides refrigeration for fresh food compartment and the other for the freezer. These cycles operate independently to meet the load requirements of the refrigerator and freezer respectively.

The main functions of a refrigerator/freezer are provision of storage space maintained at a low temperature (probably 0°C to 4°C) for the preservation of food items such as vegetables; also formation of ice cubes and storage of meat products below their freezing temperatures.

The refrigerator/freezer under consideration is a top-mount combination, which uses mechanical vapour compression cycle. The reasons being its compactness and efficient use of electrical energy. The need for large storage space and better preservation of frozen food items is another reason for this type of design. Although one compressor operates both the freezer and refrigerator (fresh food compartment), each is provided with its own door so that they can be opened independently. From experience, the service usage of fresh food compartment is higher, hence with this arrangement; freezer temperature is expected to be relatively stable. The two evaporators, which are separated by a partition, are connected to the same compressor. There are no fans connected to the system so that additional power that could be required is eliminated.

Gan et al (2000) in a related study presented energy savings of up to 30% for a dual cycle with a condensing temperature of 35°C and freezer and refrigerator evaporator temperatures of –20°C and –5°C respectively over a single cycle. Higher energy savings can be expected from a modified two-stage cycle with a suddenly enlarged pipe replacing the compressor in the first stage.

In a vapour compression refrigeration cycle, power consumption by the compressor is a function of operating conditions of evaporator and condenser. When the pressure ratio is large, the power needed for compression increases. In a conventional two-stage vapour compression refrigeration system, two compressors are used such that the pressure ratio over which each of them operates is reduced. The compressors are arranged in series so that the first compressor is responsible for compressing the refrigerant to an intermediate pressure and the second compressor performs the remaining work of raising refrigerant to its final pressure (ASHRAE, 2013, Arora, 2004). Although two-stage vapour compression refrigeration system may be more expensive, its energy efficiency is better than single and dual systems.

Gan (1999) presented a modified two-stage vapour compression system in which the compressor of the first stage of compression is expected to operate at a sub-zero temperature. It is doubtful if such a compressor is in the market now. In this present work, a pipe with sudden enlargement replaces the compressor in the first stage of compression. It is expected that the pressure difference created as a result of the sudden enlargement will be adequate for the second stage of compression. The pressure increased as a result of the sudden enlargement to an intermediate pressure, which corresponds to the evaporating pressure of the refrigerator.

II. ANALYSIS OF RESULTS

Steady state operation is assumed in analyzing the performance of modified two-stage, dual and single cycles. The cooling capacity of the refrigerator/freezer in this study is 380W (Ariyo, 2007). It is a top-mount combination with freezer temperature of –20°C and intermediate (refrigerator) temperature of –5°C while the condenser temperature is 40°C. The refrigerant used in the cycle is R134a. Intermediate pressure is limited by the set point temperature of the cabinet air because of the need for adequate heat transfer; hence evaporator temperature is maintained at –5°C. This affords comparison with single and dual cycles maintained at same operating conditions. The beauty of the cycle in this study is that no power is involved in raising the pressure from freezer pressure to intermediate (refrigerator evaporator) pressure, yet the process is isentropic. The only power expended is that of refrigerator compressor. Figure 1 shows the schematic diagram of the modified two-stage vapour compression refrigeration system while Figure 2 shows the cycle on the p-h diagram. The enlarged pipe used in the first stage of the compression is shown in Figure 3.

The following equations could be used to calculate the relevant parameters:

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2 \qquad \text{(Continuity Equation)} \qquad (1)$$

$$P = \rho R T \qquad \text{(Equation of State)} \qquad (2)$$

$$\left(\frac{\gamma}{\gamma-1}\right)\frac{P_1}{\rho_1}\left[1 - \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}\right] = \frac{V_2^2 - V_1^2}{2} \qquad (3)$$

(Bernoulli's Equation)

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} \qquad \text{(Isentropic Process)} \qquad (4)$$

(Bansal, 2003; Rajput, 2005)

Where P = Pressure, T = Temperature, A = Area, V= Velocity, ρ= Density, R = Gas Constant,

γ = Isentropic index.

Depending on the enlargement of the pipe downstream, pressure, temperature and density are expected to increase while velocity decreases. The process in this arrangement is assumed to be isentropic and no external power is supplied to the freezer compressor to raise its pressure to intermediate pressure. The refrigerant flow in the entire cycle is as a result of the refrigerator compressor; hence this arrangement is considered more energy efficient.

It was established by Gan (1999) that suction line heat exchanger decreases the performance of the freezer cycle, probably because of low intermediate pressure, therefore it is discarded in this modification and another reason is because compressor is not used in the freezer cycle. The performance parameters were calculated using R134a tables and p-h chart.

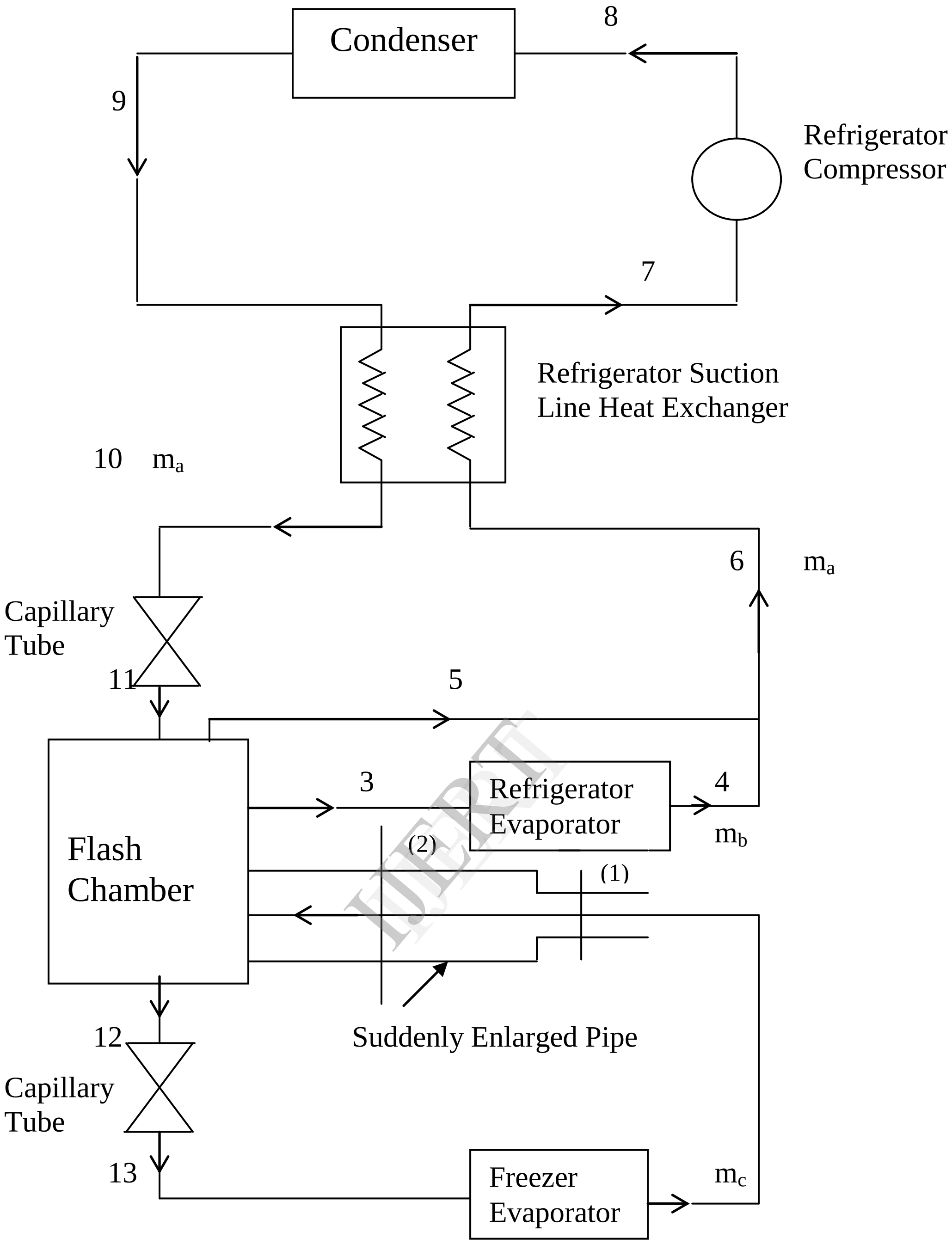

Fig 1: Schematic Diagram of the proposed Two-Stage Vapor Compression system.

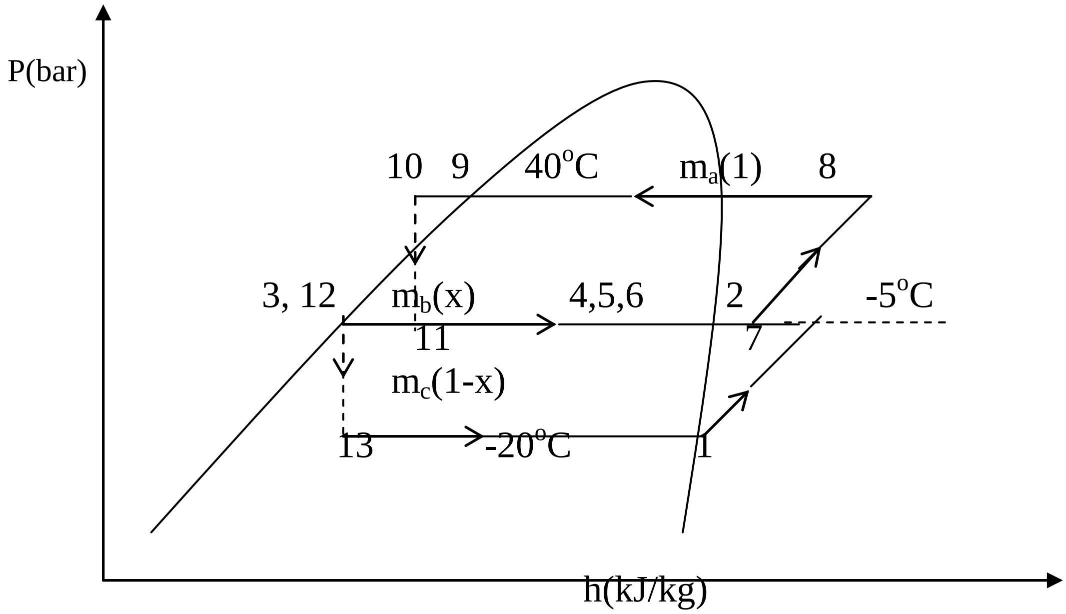

Fig 2: P-h Diagram of the proposed Two-Stage Vapour compression system

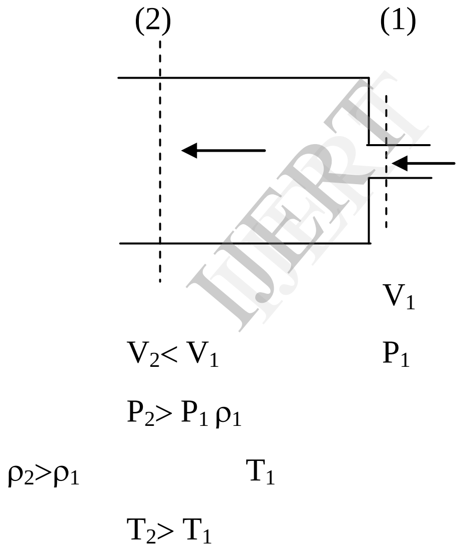

Fig. 3 Pipe With Sudden Enlargement

Domestic refrigeration units are usually small in size with compressor power between 35 and 400W, hence the cycle is designed for 380W (Ananthanarayanan, 2005; Prasad, 2005). The proposed refrigerator-freezer is a twin door type with the freezer cabinet mounted on the top.

Energy balance on the flash chamber yields the following expression:

$$m_c(h_2-h_{12}) = m_b h_3 - m_a h_{11} + (m_a - m_b)\, h_5 \quad (5)$$

where m_a, m_b and m_c, are masses of refrigerant flowing. If m_a, m_b and m_c, are translated into fractions of refrigerant flowing, then:

$$m_a = 1,\ m_b = x \text{ and } m_c = 1 - x$$

Equation (5) becomes

$$(1 - x)(h_2-h_{12}) = xh_3 - h_{11} + (1 - x)\, h_5 \quad (6)$$

The flash chamber is a small vessel which stores refrigerant at low temperature therefore the most appropriate environment to place it is in the refrigerator. The flash chamber stores refrigerant at the intermediate pressure for the use of the freezer cycle. The freezer and refrigerator cycles run simultaneously and there is continuous flow of refrigerant, therefore sizing of the flash chamber is also not crucial in this modified design. A cylindrical volume of 1.3 x 10^{-4} m^3 may be appropriate. The following parameters relate to the proposed cycle:

Refrigerant

R134a

Freezer saturation temperature

-20°C

Fresh food evaporator saturation temperature

-5°C

Condenser saturation temperature

40°C

For Dual cycle:

(i) Freezer cycle.

Evaporator saturation temperature

-20°C

Condenser saturation temperature

40°C

(ii) Refrigerator cycle

Evaporator saturation temperature

-5°C

Condenser saturation temperature

40°C

For single cycle:

Evaporator saturation temperature

-20°C

Condenser saturation temperature

40°C

If the temperature of the refrigerant is below its saturation temperature, it is said to be subcooled. This takes place at constant pressure. Practically speaking subcooling of the condensate is desirable before throttling because this reduces flashing and refrigerating effect is increased, hence degree of subcooling of 5K is chosen. Superheating is also desirable to prevent wet refrigerant from entering the compressor, which could result in severe mechanical problems, therefore degree of superheating of 7K is considered appropriate (Ariyo, 2007).

Four cases of refrigeration capacity sharing which also resulted in compression power sharing were considered for the dual cycle. These are: (i) 1:1 (ii) 2:1 (iii) 3:1 (iv) 4:1. In all four cases, compression power for single and two-stage cycles remain the same. The results of the analysis are shown in Tables 1 and 2.

Table 1 Computed Performance Data of Single, Dual and Modified Two-Stage Cycles

Single Cycle | Dual Cycle

				Ref. Cycle			Freezer Cycle		
Cases	Compression Power (W)	Ref. Cap. (W)	COP	Comp. Power (W)	Ref. Cap. (W)	COP	Comp. Power (W)	Ref. Cap. (W)	COP
I	116.3	380.0	3.3	38.3	190.0	5.0	58.2	190.0	3.3
II	116.3	380.0	3.3	51.0	253.3	5.0	38.8	126.7	3.3
III	116.3	380.0	3.3	57.4	285.0	5.0	29.1	95.0	3.3
IV	116.3	380.0	3.3	61.2	304.0	5.0	23.3	76.0	3.3

Modified Two- Stage Cycle

Comp. Power (W)	Ref. Cap. (W)	COP
58.3	380.0	6.5
58.3	380.0	6.5
58.3	380.0	6.5
58.3	380.0	6.5

Table 2 Comparison of Energy Savings of Single, Dual and Modified Two-Stage Cycles

		Power Consumption (W)		Energy Savings (%)		
Cases	Single cycle	Dual cycle	Modified Two-Stage cycle	Dual over single cycle	Two stage over single cycle	Two-stage over Dual cycle
I	116.3	96.5	58.3	17.0	49.9	39.6
II	116.3	89.8	58.3	22.8	49.9	35.1
III	116.3	86.5	58.3	25.6	49.9	32.6
IV	116.3	84.5	58.3	27.3	49.9	31.0

DISCUSSION OF RESULTS

A pipe with sudden enlargement introduced between the freezer evaporator and flash chamber replaces the compressor in the first stage. Isentropic process is assumed and intermediate temperature (evaporator temperature) is expected to be $-5^{o}C$. Because of this modification, flow of refrigerant in the entire cycle is as a result of the compressor in the refrigerator cycle and pressure and temperature rise occurs in the suddenly enlarged pipe. Energy savings of 22.7% could be recorded in this modified cycle over Gan's (1999) two-stage cycle.

For the cases considered, the modified two-stage cycle showed considerable energy savings over single and dual cycles. For example in the first case, energy consumption for single, dual and two-stage cycles are 116.3W, 96.5W and 58.3W respectively. Dual cycle also showed considerable reduction in energy consumption over single cycle, but this may not be feasible after the second case as can be seen in Table 1 because it is doubtful if compressors of capacities less than 30W are readily available in the market. This could be a limitation for dual cycle. Maximum energy savings of 49.9% and 39.6% of modified two-stage cycle were recorded over single and dual cycles respectively, while maximum energy saving of 27.3% of dual cycle over single cycle was recorded in case four. The coefficients of performance of the proposed two-stage cycle and single cycle are 6.5 and 3.3 while the COP of the refrigeration cycle of the dual cycle is 5.0 and that of its freezer cycle is 3.3.

The actual refrigeration capacity takes into consideration volumetric efficiency of the compressor since the actual amount of refrigerant vapour pumped is less than the theoretical. Therefore, the actual refrigeration capacity is less than the theoretical refrigeration capacity. Also, isentropic, compression and mechanical efficiencies will influence the power input needed for compression. These factors were not taken into consideration in the computation and could influence the results.

CONCLUSION

Energy savings of the proposed two-stage cycle are better than those of single and dual cycles. The beauty of this design is that one compressor is required which makes it more attractive than the dual cycle, which requires two compressors for its operation. The coefficient of performance for the two-stage cycle is the highest and compression power for the same refrigerating capacity is the least. Unlike the dual cycle no additional space is required for the second compressor, hence initial cost and space requirement are minimized. Additional piping and refrigerant charge are needed; these are offset by the energy savings. The pipe with sudden enlargement needs to be properly designed to achieve the required temperature and pressure at the intermediate stage.

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