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Experimental Investigation and Monitoring for Location and Computation of Irreversibility in a Vapour Compression Cycle

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ABSTRACT

The Refrigerators and air conditioners based on vapour compression refrigeration cycle are one of the important elements of the society either for domestic or industrial purposes. it is essential in view of energy crisis to reduce the power consumed by these refrigerators. This paper presents a new dimension and or specification for monitoring the power consumed based on location and computation of irreversibility. COP of a reversible cycle is highest amongst all refrigerators operating between the same temperature limits, this characteristic of a reversible cycle is applied to an actual vapour compression refrigeration system setup. The result is that we can locate the component of the system which contributes most towards irreversibility and also, we can calculate the amount of irreversibility thereby knowing the extra power consumed by the component. If the amount of irreversibility is taken as a specification for either replacing or maintaining the system, we can save the extra power consumed. In view of energy crisis this becomes very important. This paper uses the Ist and IInd law thermodynamic analysis to locate and calculate the irreversibility in a vapour compression refrigeration system. The computations are made for an actual vapour compression refrigeration cycle setup considering up to 7 decimal places. The result gives the extra power required by compressor. The difference between actual power & Carnot power equals 1.1174206, which is equal to the product of irreversibility rate & ambient temperature. This analysis when applied to an actual system acts like a power saving mode, we have in a personal computer.

KEY WORDS: Irreversibility, Isentropic, COP, Adiabatic, Reversible.

INTRODUCTION

The basic Vapor Compression Refrigeration Cycle includes compressor, condenser, expansion valve, and evaporator for efficient heat transfer and cooling. The perfect cycle takes into account pressure-loss-free heat transfer in the condenser and evaporator, reversible adiabatic (isentropic) compression, and an adiabatic expansion valve coupled by pipework that experiences neither pressure loss nor external heat transfer. At point 1, the refrigerant exits the evaporator as a saturated vapour at low pressure and low temperature. It next enters the compressor, where it

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undergoes adiabatic, reversible compression (isentropic). As it exits the compressor at point 2, the vapour is superheated and at a high temperature. It then enters the condenser, where it is first desuperheated and then condensed at a steady pressure. At step 3, the refrigerant exits the condenser as a saturated liquid at high pressure and medium temperature. It then enters the expansion valve, where it expands adiabatically (constant enthalpy) and irreversibly. It enters the evaporator at point 4, where it exits the expansion valve as a low-quality, low-pressure, low-temperature vapour. There, it is reversibly evaporated at constant pressure to reach the saturated condition at point 1. With the exception of the condenser's desuperheating operation, heat transmission to and from the evaporator and condenser happens without a limited temperature differential between the fluids that produce and absorb heat [1].

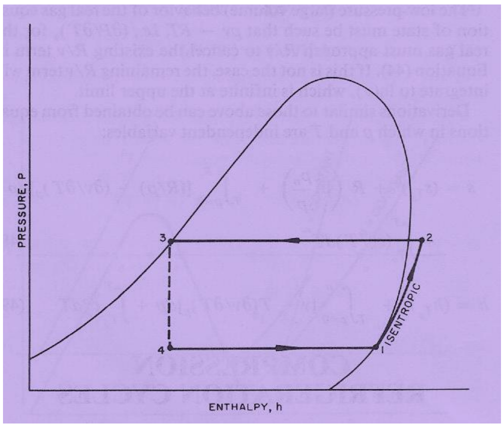

Fig 1.1 Pressure-Enthalpy Diagram [1]

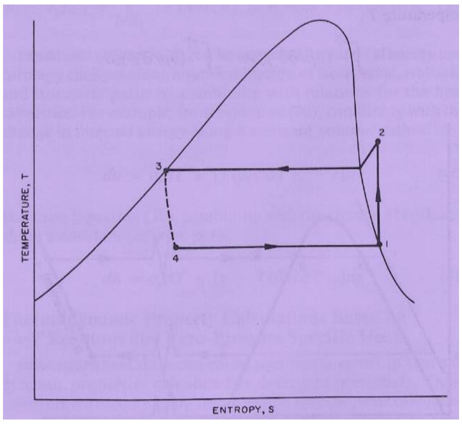

Fig 1.2 Temperature-Entropy Diagram [1]

EXPERIMENTATION

A. Second Law Thermodynamic Analysis

The notions of accessible energy and the reversible process are fundamental to applying the second law of thermodynamics to the analysis of compression refrigeration cycles. These fundamental ideas allow for the expression of a process or system's performance in terms of the irreversibility and deterioration of accessible energy. The following examples apply these concepts to compression refrigeration systems, relate the principles of irreversibility $I = T_0 \Delta S_{total}$ and emphasize use of the T-S coordinates to illustrate the effects of losses in the refrigeration system.[2]

B. Analysis of Vapour Compression Cycle Using R12

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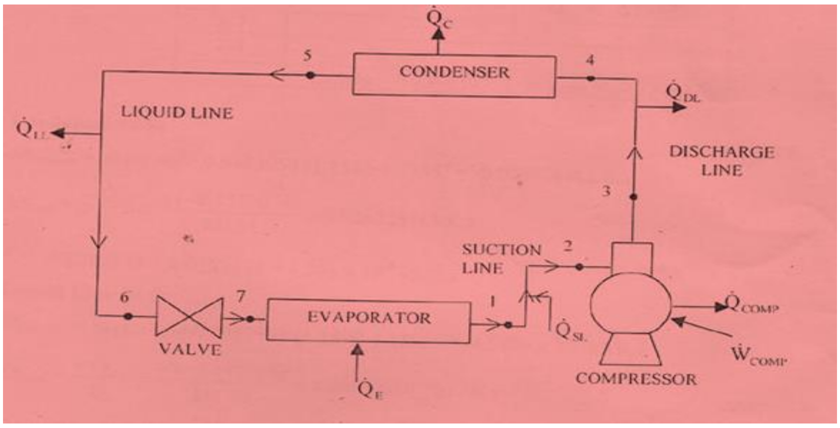

Fig 2.1: Vapour Compression Refrigeration System setup [1]

With reference to Fig 2.1 the actual data has been considered as Ambient air temp (t_0) = 30^0C, Refrigerated space temp (t_R) = -20^0C, Refrigeration load (Q_E) = 7 kW, compressor power input (W_{COMP}) = 2.5 kW.

We can now calculate the energy transferred to the refrigerant in each system component, as well as the irreversibility rate in each component. We can also demonstrate that the difference between the actual power input and the amount of power needed by a Carnot cycle functioning between t_R and T_0 with the same refrigeration load is equal to the total irreversibility rate multiplied by the ambient temperature.

Table 2.1 Properties of R12 [4]

State	Pressure (kPa)	Temperature (^{0} C)	Sp enthalpy (kJ/kg)	Sp. Entropy (kJ / Kg. K
1.	122.68	-18	388.04	1.7277
2.	117.68	-10	392.99	1.7607
3.	903	55	436.79	1.7782
4.	893	42	423.05	1.7386
5.	888	33	246.32	1.1585
6.	883	31	243.40	1.1490
7.	142.68	-22	243.40	1.1505

1) Analysis [3]

Mass flow rate (m)$\frac{Refrigeration\ load\ (Q_E)}{(h_1 - h_2)} = \frac{7}{388-04-243-40} = 0.0483959\ kg/s$

Energy transferred in suction line

$Q_{SL} = m\ (h_2 - h_1) = 0.0483959(392.99 - 388.04) = 0.2395597$ kW

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$Q_{COMP} = -W_{COMP} + m\ (h_3 - h_2)$

$Q_{COMP} = -2.5 + 0.0483959(436.79 - 392.99) = -0.3802596$ kW

$Q_{DL} = m\ (h_4 - h_3) = 0.0483959\ (423.05 - 436.79) = -0.6649596$ kW

$Q_C = m\ (h_5 - h_4) = 0.04833959(246.32 - 423.05) = -8.5530074$ kW

$Q_{LL} = m\ (h_6 - h_5) = 0.0483959\ (243.40 - 246.32) = -0.141316$ kW

2) Energy Balance Sheet [5]

Table 2.2: Energy Balance Sheet

PROCESS	GAIN (kW)	Loss (kW)
1-2	$Q_{SL} = 0.2395597$	-
2-3	$W_{COMP} = 2.5$	$Q_{COMP} = 0.3802596$ kW
3-4	-	$Q_{DL} = 0.6649596$ kW
4-5	-	$Q_C = -8.5530074$ kW
5-6	-	$Q_{LL} = -0.141316$ kW
6-7	-	-
7-1	$Q_E = 7$	-
	T. Gain = 9.7395597	**T. Loss = -9.7395426**

3) Condenser (4-5)

$\Delta S_{system} = m\ (S_5 - S_4) = 0.0483959\ (1.1585 - 1.7386) = -0.0280744$KJ/K. s

$\Delta S_{surr} = - Qc / T_O = - (8.5530074) / 303.15 = 0.0282137$ kJ/K. s

$\dot{I} = - 0.0280744 + 0.0282137 = 1.393 \times 10^{-4}$ kJ/K. s Eq 2.1

4) Liquid Line (5-6)

$\Delta S_{system} = m\ (S_6 - S_5) = 0.0483959\ (1.1490 - 1.1585) = - 4.5976 \times 10 -4$ kJ

$\Delta S_{surr} = - Q_{LL} / T_O = - (-0.141316) / 303.15 = 4.66158 \times 10^{-4}$ kJ/K. s

$\dot{I} = - 4.5976 \times 10^{-4} + 4.66158 \times 10^{-4} = 6.398 \times 10^{-6}$ kJ/K. s Eq 2.2

5) Discharge Line (3-4)

$\Delta S_{system} = m\ (S_4 - S_3) = 0.0483959\ (1.7386 - 1.7782) = - 1.9164 \times 10^{-3}$ kJ/K. s

$\Delta S_{surr} = - Q_{DL} / T_O = - (-0.6649596) / 303.15 = 2.1935 \times 10^{-3}$ kJ/K. s

$\dot{I} = - 1.9164 \times 10^{-3} + 2.1935 \times 10^{-3} = 2.771 \times 10^{-4}$ kJ/K. s Eq 2.3

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6) Suction Line (1-2)

$\Delta S_{system\,=}$ m $(S_2 - S_1)$ = 0.0483959 (1.7607 – 1.7277) = 1.59706 x 10 $^{-3}$ kJ/K. s

ΔS_{surr}= - Q $_{SL}$ / T_O = - 0.2395597 / 303.15 = - 7.90234 x 10 $^{-4}$ kJ/K. s

İ = 1.59706 x 10 $^{-3}$ – 7.90234 x 10 $^{-4}$ = 8.0683 x 10 $^{-4}$ kJ/K. s Eq 2.4

7) Evaporator Line (7-1)

$\Delta S_{system\,=}$ m $(S_1 - S_7)$ = 0.0483959 (1.7277 – 1.1505) = 0.0279341 kJ/K. s

ΔS_{surr} = - Q $_E$/ T_R = - 7 / 253.15 = - 0.0276515 kJ/K. s

İ = 0.0279341 – 0.0276515 = 2.826 x 10 $^{-4}$ kJ/K. s Eq 2.5

8) Throttling (6-7)

ΔS_{system}= m $(S_3 - S_2)$ = 0.0483959 (1.1505 – 1.1490) = 7.25938 x 10 $^{-5}$ kJ/K.s

ΔS_{surr} = 0 as $h_6 = h_7$

İ = 7.25938 x 10 $^{-5}$ + 0 = 7.25938 x 10 $^{-5}$ kJ/K. s Eq 2.6

9) Compressor (2-3)

ΔS_{system}= m $(S_3 - S_2)$ = 0.0483959 (1.7782 – 1.7607) = 8.46928 x 10 $^{-4}$ kJ/K. s

ΔS_{surr} = - Q $_{COMP}$/ T_O = - (-0.3802596) / 303.15 = 1.25436 x 10 $^{-3}$ kJ/K. s

İ = 8.46928 x 10 $^{-4}$ + 1.25436 x 10 $^{-3}$ = 2.10128 x 10 $^{-3}$ kJ/K. s Eq 2.7

İ = 8.46928 x 10 $^{-4}$ + 1.25436 x 10 $^{-3}$ = 2.10128 x 10 $^{-3}$ kJ/K.s Eq 2.7

Total change of irreversibility rate

(İ) = 1.393 x 10 $^{-4}$+ 6.398 x 10 $^{-6}$ + 2.771 x 10 $^{-4}$ + 8.0683 x 10 $^{-4}$ + 2.826 x 10 $^{-4}$ + 7.25938 x 10 $^{-5}$

+ 2.10128 x 10 $^{-3}$ = 3.6861 x 10 $^{-3}$ kJ/kg

İ x T_O = 3.6861 x 10 $^{-3}$ x 303.15 = 1.1174412 kW Eq 2.8

For Carnot cycle, COP = T_R / T_O - T_R = Refrigerating capacity / Work required

Work required = 7 (303.15 – 253.15) / 253.15 = 1.3825794 kW Eq 2.9

Hence, difference between actual power and Carnot cycle power

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= 2.5 – 1.3825794 = 1.1174206 kW

This is equal to the product of irreversibility rate and ambient temperature, from Eq 2.8 & Eq 2.9.

RESULTS OBTAINED

The Table3.1 below shows the amount of extra work required by an actual vapour compression refrigeration cycle using R134a. As per above analysis this extra work is equal to rate of irreversibility multiplied by ambient temperature. Experimental values have been taken at the end of each working month. The refrigerator was made to work 6 hrs. a day. It was found that after 5 months of working there is substantial increase in the extra work i.e. rate of irreversibility. Thus, a decision of maintaining the compressor or replacing a component can be based on this new dimension.

Table 3.1Work required by an actual vapour compression refrigeration cycle using R134a.

End of month	Extra work required = rate of irreversibility x ambient temperature	Remarks
1	1.1174206	Original irreversibility present in new
2	1.1174310	No significant change
3	1.1175213	No significant change
4	1.1177234	Minor increase in extra work required
5	1.301226	Significant increase
6	1.321266	Significant increase

CONCLUSIONS

It is possible to apply this crucial examination of the first and second laws of thermodynamics to any real-world vapour compression refrigeration system. Actually, this analysis determines the additional compressor power needed to overcome each component's irreversibility. The compressor is the part that has lost the most. This loss results from irreversibility caused by pressure decreases, mixing, and transfers between the compressor and the environment, as well as motor inefficiencies and friction losses. Another significant loss is the expansion device's unrestricted expansion. The analysis includes all hear transfer irreversibility on both the condenser and evaporator sides. Included is also the refrigerant pressure drop. The components that need to be maintained, replaced, or modified in order to increase performance are identified with the aid of this examination. Nevertheless, this kind of study is unable to determine the specifics of the losses. Applying this approach to big refrigeration systems will undoubtedly result in a rise in the additional power needed and assist in making critical decisions about the maintenance and replacement of individual parts or the entire system, thereby keeping an eye on the power requirements.

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