Laboratory Report **ES3B5 Engines and Heat Pumps**
Laboratory date 28/11/2017 Student Number: u1514864

1 Summary

This report details the analysis of the working cycle of a vapour-compression refrigeration system. The temperature of the supplied air to the system was varied and the resulting changes in pressure, temperature and enthalpy of the refrigerant measured. The p-h and h-x diagrams were then plotted and analysis of the system conducted. The factors of interest were the coefficients of performance (Carnot and real), the work done by the compressor, the heat flow from the refrigerant and the air in and out of the system. It was found that the Carnot COPs decreased when there was a greater difference between the evaporating and condensing temperatures. However, the real COPs were found to be higher than their Carnot equivalents. This discrepancy was attributed to various sources of error as highlighted in the analysis and conclusion sections of this report.

Contents

2 Introduction

The aim of the experiment was to analyse the performance and working cycle of a refrigeration unit. This was done with the use of p-h diagrams and measured data for certain set supply temperatures of the unit. Thermodynamic ratios were used for calculation of the heat flows, work outputs/inputs and coefficients of performance. The divergence between the ideal performance and real cylce was found to be due (in part) to the inefficiency in the mechanical compressor, with the isentropic enthalpy at the compressor interpolated from saturation data tables for the refrigerant R134a.

3 Experimental Procedure

The set-up used for the experiment was the ET 915.06, a training refrigeration unit produced by GUNT. Its associated software package was used to produce the p-h and h-x diagrams along with all the tabulated data in the results section of this report. The following procedure was carried out as part of the experiment:

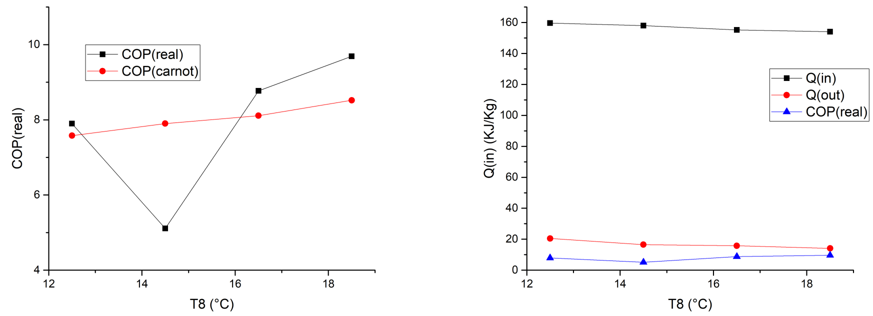
1. The software program was opened to display the data acquisition control panel. Alongside this, the main power switch to the unit was turned on.
2. The supply air temperature (T8) was set to an initial temperature of 18.5°C.
3. The auxiliary fans and compressor were turned on through their associated buttons in the control panel. This activated the refrigeration unit.
4. The control panel live temperature display was monitored until the set temperature corresponded to a stable output temperature. This was realistically performed for 30 minutes, until the temperature was approximately stable to ± 1°C.
5. Readings for temperature, pressure, specific enthalpy and humidity were then taken for each component of the unit. These were taken directly from the control panel display.
6. The p-h and h-x diagrams were plotted with the software package.
7. This process was repeated for the set temperatures (T8) of 16.5°C, 14.5°C and 12.5°C.

4 Results

The data shown overleaf, in Table 1, represent the findings of the above procedure for each set temperature. The values of Tev and Tcon represent the evaporating and condensing temperatures of the unit, respectively. These were found through interpolation of the saturation pressures P1 and P2. The values for h2 isentropic were found through interpolation of the entropy values corresponding to the difference in enthalpies between the theoretical and real work of the compressor. The subsequent values of heat flow (Q), work (W), coefficients of performance (COP) and the isentropic enthalpies/entropies/efficiencies ($h_{\prime}$ $S_{\prime}$ and $\eta_{\prime}$) were calculated using the formulae shown in Appendix 1.

Variable	**Test 1**	**Test 2**	**Test 3**	**Test 4**
T8 set (°C)	12.5	14.5	16.5	18.5
T1 (°C)	3.0	5.0	6.5	8.7
T2 (°C)	39.5	50.4	40.6	40.7
T3 (°C)	33.6	30.5	34.7	36.0
T4 (°C)	33.6	25.0	36.1	35.6
T5 (°C)	3.9	0.3	6.7	9.4
T6 (°C)	4.0	1.8	7.0	9.8
P1 (bar)	3.11	2.82	3.48	3.82
P2 (bar)	9.60	8.50	9.95	10.27
Tev (°C)	1.60	-1.02	4.85	7.47
Tcon (°C)	37.83	33.40	39.10	40.37
T7 (°C)	28.9	26.3	28.9	29.4
ϕ1 (%)	34.4	33.0	30.9	31.0
T8 (°C)	12.5	14.6	16.3	18.4
ϕ2 (%)	71.3	51.6	55.6	51.2
ΔP1 (mbar)	2.30	2.33	2.14	2.24
h1 (kJ/kg)	399.9	403.6	402.7	404.4
h2 (kJ/kg)	420.1	434.5	420.4	420.3
h2 isentropic, $h_{2'}$ (kJ/kg)	412.2	417.9	411.9	412.0
h3 (kJ/kg)	242.0	242.5	248.2	250.8
h4 (kJ/kg)	240.6	242.5	248.2	250.8
h5 (kJ/kg)	240.6	242.5	248.2	250.8
h6 (kJ/kg)	400.2	400.5	403.4	404.9
h7 (kJ/kg)	45.5	44.3	48.3	49.9
h8 (kJ/kg)	25.0	27.8	32.5	35.8
Qair,out (kJ/kg)	20.5	16.5	15.8	14.1
Qin (kJ/kg)	159.6	158.0	155.2	154.1
W (kJ/kg)	20.2	30.9	17.7	15.9
COP_{carnot}	7.58	7.90	8.11	8.52
COP_{real}	7.90	5.11	8.77	9.69
Isentropic Efficiency, $\eta_{'}$ (%)	60.9	46.3	52.0	47.2

Table 1: Experimental Results.



(a) COP(real), COP(carnot) vs T8

(b) Qin, Qout, COP(real) vs T8

Figure 1: (a),(b) Performance measures vs set temperature

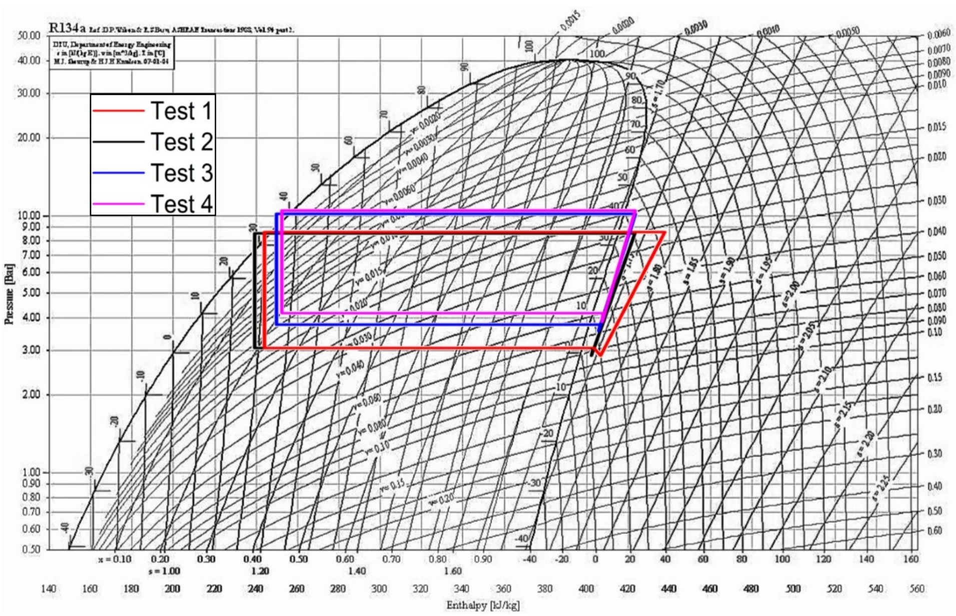


Figure 2: PH diagrams for all tests

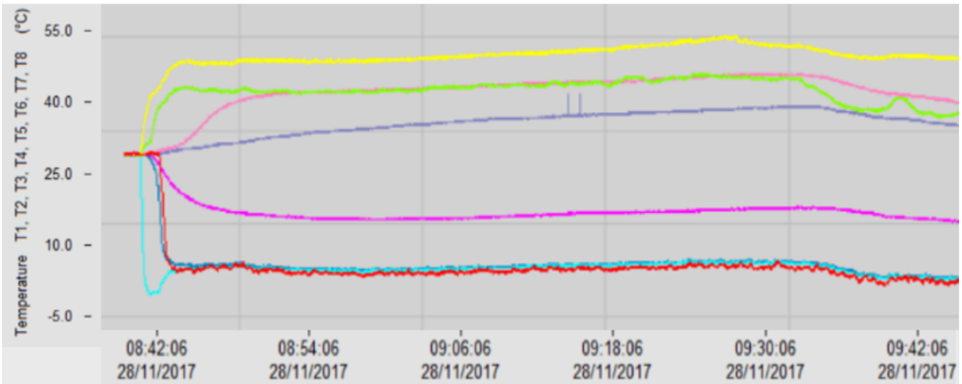


Figure 3: Instability and temperature fluctuations throughout the tests

5 Analysis

The results show that numerous errors are present. There is great discrepancy in that tests 1, 3 and 4 present COPs higher than their theoretical Carnot limits. However, the Carnot COPs still decrease with lower set temperature. Reassuringly, this corresponds to the fundamental principle that the COP must decrease with increased difference between the evaporating and condensing temperatures. The higher set temperatures were closer to room temperature (approximately 20°C) and so less work was required for refrigeration.

The unlikely real COPs may have resulted from any combination of errors arising from: heat lost from air into the evaporator, thermal inertia of the system, hysteresis of the control mechanism or latency of the solenoid valves. Given that the enthalpy values used for COP calculation were provided by the test unit, it may be possible that error from sensors in the unit provided anomalous data, resulting in physically implausible results. It is also likely that the suboptimal conditions in which the tests were conducted, led to large fluctuations and hence errors in measurement. This is represented by **Figure 3** above. Readings were taken in a room with variable heat flow due to its occupants and so great instability in measurement can be seen.

Analysis of the work done by the compressor was done thermodynamically, providing the work output. This is open to distortion by temperature fluctuations derived from the laboratory conditions. The input work was not considered and should have been calculated given the electrical power drawn by the compressor. This would have given a more accurate real COP, being based on the work input. The Isentropic efficiencies of the compressor were reasonable (45 to 60%) but fall short of typical values (75 to 90%). This could well be attributed to age and wear on its mechanical components.

6 Conclusion

The results of the experiment have shown inconsistency with regards to some of the fundamental principles of thermodynamics. The correlation found between Carnot COP and the evaporating/condensing temperature difference is plausible, whereas the disparity between the Carnot and real COPs is not. The laboratory conditions and instability in attempting consistent output temperatures along with all the aforementioned errors have therefore produced variable and inconsistent results. Consequently, it is recommended that the experiment be repeated with better controlled temperature conditions and with more time allowed for thermal equilibrium to be achieved at each set temperature.

7 Appendices

7.1 Employed Formulae

Cooling Energy

$Q_{air,out} = h_7 - h_8$ kJ/kg (air)

$Q_{in} = h_6 - h_5$ kJ/kg (refrigerant)

Compressor Work

$W = h_2 - h_1$ kJ/kg

Coefficients of Performance

$COP_{carnot} = \frac{T_{ev}}{T_{con} - T_{ev}}$

$COP_{real} = \frac{Q_{in}}{W}$

Isentropic Enthalpy at Compression

$h_{2'} = h_1 + \frac{(S_{2'} - S_g)}{(S_{10k} - S_g)} \cdot (h_2 - h_1)$ kJ/kg

Isentropic Efficiency of the Compressor

$\eta_{'} = \frac{(h_{2'} - h_1)}{(h_2 - h_1)}$