CHAPTER 6

DISCUSSION AND CONCLUSION

6.1 PERFORMANCE OF REFRIGERATING PLANTS

In Chapter 3 and Chapter 4, the test of refrigerating plant with "Freon-12" and "Freon-22" refrigerants charged was operated on two conditions.

The first one, to find the refrigeration performance of refrigerating cycle for various pressures at the same delivery pressure and suction superheated vapor temperature. The tests were carried on Test 3.1 and Test 4.1, they gave the satisfactory results as shown on the curves of Graph 1,2,5, and 6 in Appendix III, the performance of refrigerating cycle which the evaporation pressure was varied, are concluded as follow:

a. The refrigerating effect per pound of refrigerant circulated decreases with the increasing of suction pressure on both Graph 1 and Graph 5.

It can be observed from the p-h diagram of R-12 and R-22, that the slope of the constant temperature lines in the superheated region is negative, enthalpy of the constant temperature suction vapor as the evaporating pressure is varied. This results in the decreasing of refrigerating effect. The graphs plotted go together with the theoretical argument.

b. The mass flow rate of refrigerant and the capacity increase when the suction pressure increases.

As the suction pressure is increased, the vapor density at the compressor inlet increases. Therefore for the same capacity of compressor (volume intake), the compressor can take more refrigerant vapor.

Since the mass flow rate increase as discussed, the capacity of the plant also increases.

c. Graph 2 and Graph 6 show the power required to drive the compressor, the bhp increases with the suction pressure. This is to compensate the liquid mass flow rate or refrigeration capacity.

Since the refrigerating capacity goes up higher than the power required at higher suction pressure, the result is that the plant operates more efficiently which can be seen from the coefficient of performance, this means the efficiency of the cycle improves considerably as the suction pressure increases.

d. The volumetric efficiency is affected by the re-expansion of the clearance vapor.

In the compression cycle, as the suction pressure is raised, the re-expanded vapor would occupy less volume, hence allows more vapor intake during suction stroke. The result is the increase in the volumetric efficiency.

Theoretically, as the suction pressure equals to the discharge pressure, the volumetric efficiency is 100 %. That is the volumetric efficiency goes up as the suction pressure approaches the discharge pressure. It is confirmed by the experimental result.

The second setting is, to find the refrigerating performance of refrigerating cycle for different suction superheat temperatures at constant suction pressure and delivery pressure. The results are discussed as follow:

a. The refrigerating effect, Btu/lb of refrigerant circulated increases by increasing the degree of superheat as shown in Graph 3 and Graph 7. The effect can be observed from the p-h diagram, that is a pound of refrigerant absorbed more heat in passing through the evaporator.

b. The temperature of discharge gas leaving the compressor increases when the superheat temperature of suction vapor increases.

c. The power required by the compressor decreases slightly as the superheat temperature increases. This is shown in Graph 4 and 8.

d. Three principal factors which affect compressor volumetric efficiency are

i. re-expansion of clearance vapor,

ii. pressure drop in suction and discharge valves,

iii. heating of the vapor on the intake stroke.

In our case by increasing the degree of superheat, the heat transfer to the gas was reduced during the intake stroke. Hence the volumetric efficiency increases.

e. Since the refrigerating capacity increases and the power decreases slightly as the degree of superheat of vapor leaving the evaporator increases. It results in better coefficient of performance of the system.

During the experiment, it was found difficult to bring the plant to steady state conditions. One of the contributing factors was fluctuating electric voltage which affected the load on the evaporator. Another factor is the temperature of the cooling water which is rather high.

Nevertheless the results obtained from the R-12 plant and R-22 plant go together and are satisfied.

6.2 BEST COMPOSITION OF MIXED R-12 AND R-22 REFRIGERANT

In investigation for the best composition of R-12 and R-22 which gives better properties than their pure components are carried out at the same chosen conditions, i.e, evaporating temperature, suction gas temperature, for various mass compositions.

Graph 10 shows the performance of the plant operating with various compositions of mixed refrigerant charged. The power required by the plant, the coefficient of performance, the delivery gas temperature, the refrigerating effect, and the refrigerating capacity are plotted. All are convex curves joined between their pure component values, except the refrigerating effect line which is nearly a straight line.

Graph 11 shows the deviation of the refrigerating capacity and the power required from their proportional values to mass fraction. Observation from the capacity curve, the composition of R-12 and R-22 at 69 : 31 percent by weight has the peak deviation, and the coefficient of performance is also maximum at that point. This means that the capacity deviation and the coefficient of performance back up each other. We could draw a conclusion from our experiment that our mixed refrigerant of R-12 and R-22 at a proportion of 69 : 31 is a proposed refrigerant for industry, because it has many advantages over their original components.

Table 5.3 shows the discovered mixed refrigerant comparing with R-12 and R-22, at the same operating conditions of 85 F suction gas temperature, 15F expansion temperature and 95.4 F liquid temperature at condenser outlet.

6.3 ADVANTAGES COMPARED WITH R-12

a. The refrigerating capacity increases 33 % per compressor displacement. Since the power required increases only 25 %, hence the coefficient of performance is better. Fortunately, it also gives the best coefficient of performance.

b. Smaller compressor required to get the same refrigerating capacity, it means compressor cost reduced.

c. At evaporating temperature below -21 F, it remains positive suction pressure when R-12 must be operated below atmospheric pressure which is undesired operating pressure.

d. By adding about 30 % by weight of R-22, ability to absorb moiture in the system is better, therefore less trouble with freeze-up in the system can be achieved.

6.4 ADVANTAGES COMPARED WITH R-22

a. Compressor discharge temperature and discharge pressure are lower, this avoids overheating of the compressor, and light weight equipment can be employed.

b. Increasing oil miscibility of R-12 under all operating conditions, less trouble with lubrication is obtained.

c. The proportion of R-12 and R-22 is 69 : 31 percent by weight, and R-22 cost is about 70 % higher than R-12. Hence the refrigerant cost lower.

6.5 SUGGESTION FOR FURTHER WORK

In the experiment with the refrigeration calorimeter, most components, meters and gages give satisfied results for the purpose of this work. One thing should be provided is the supply of lower temperature cooling water for the plant.

The carried out mixed refrigerant of R-12 and R-22 is interesting. The study could be extended to cover its performance at other conditions. Due to its dominant properties, it may be popular and in common use in future.

Since there are about fifty compounds in the halocarbon family, a lot of work is left to be studied for any combination of those.