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Sensitivity of refrigeration system performance to charge levels and parameters for on-line leak detection

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Abstract

The environmental impact of refrigeration systems can be reduced by improving their efficiency and reducing refrigerant leakage. This paper presents results of experimental investigations on the effect of refrigerant charge on steady-state system performance and identifies parameters sensitive to charge level for on-line leak detection in vapour compression refrigeration systems. The investigations were performed on a small 4 kW nominal cooling capacity vapour compression water-to-water chiller equipped with plate type heat exchangers. The results indicate that the chiller could operate over a wide range of charge levels, 25% below to 25% above the design value without significant impact on its performance. Outside this range, the performance of the chiller was found to be strongly dependent upon the charge level. The degree of superheat at the evaporator outlet and sub-cooling at the condenser outlet were shown to be significantly dependent on the level of refrigerant charge. Overcharging was found to lead to excessive discharge pressures, but suction pressure was found to be relatively insensitive to charge level. It can be concluded that the parameters of superheating and sub-cooling together with discharge pressure can form the basis of a reasonably inexpensive on-line leak detection system.

Keywords: Liquid chillers; Refrigerant charge; Leak detection; Fault diagnosis

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1. Introduction

The impact of refrigeration systems on the environment can be reduced by operating at higher levels of energy efficiency and by reducing refrigerant leakage. All refrigeration systems have the potential to leak because pressures in the system are usually many times higher than atmospheric. Refrigerant loss also contributes to the reduction of the operating efficiency of the system, leading to increased power consumption and greenhouse gas emissions, higher maintenance costs and eventual system failure.

Conventional leak detection methods using refrigerant sensors require careful sensor location in order to be effective and may not be suitable for well ventilated areas. An alternative approach is to determine the refrigerant charge level from on-line system monitoring and performance analysis. The advantages of this approach are the use of existing system instrumentation, insensitivity to location and operating environment, and low implementation cost.

A number of studies have examined the performance of refrigeration and heat pump systems with various levels of system charge. Inatsu et al. [1] developed a device for the detection of low charge in an automotive air-conditioning system. The authors' work established that, for their application, measuring the liquid–gas flow in the liquid line was the best way to monitor the amount of refrigerant in the system. The device developed was capable of monitoring when the refrigerant charge dropped to 60% of normal level.

Bailey [2] investigated the operation of an air-cooled chiller with a range of refrigerant charge levels. The author concluded that the chiller power consumption in kW per ton of cooling delivered (1 ton = 3.5 kW) was directly proportional to refrigerant charge level at charges above 70% of normal. Below this charge, the chiller power consumption was inversely proportional to charge level. Sub-cooling temperature, discharge and suction pressure were found to be directly proportional to charge level whilst superheat was found to be inversely proportional.

Farzad and O'Neal [3] examined the performance of a 10 kW cooling capacity residential split-system air conditioner with various levels of refrigerant charge. The system featured a capillary tube expansion device. Performance was found to degrade more severely with undercharge conditions than overcharge. The superheat at the evaporator outlet was found to decrease with both increasing condenser temperature and charge level. Sub-cooling was shown to increase with increasing charge. A second paper by the same authors [4] examined the differences between a residential heat pump fitted with a capillary tube and a short tube orifice. The performance of the short tube orifice expansion system showed little dependence on the charge level, in contrast to the capillary tube system.

The effect of refrigerant charge level on air-conditioning systems was also examined by Goswami et al. [5]. The authors concluded that charge level has a significant effect on the performance of air-conditioning systems at levels below 80% of normal. For a charge level of 90% of normal, the effect on COP and cooling capacity was found to be negligible.

Damasceno et al. [6] used a theoretical heat pump model to predict heating and cooling capacities for various charge levels. The authors found that careful measurement of the internal volumes of the system components was necessary to quantify refrigerant mass in the system accurately and that some minor modifications to the void fraction model were required to enable good agreement between measured and predicted values.

The majority of investigations described above were carried out on air-to-air heat pump or air-conditioning systems. Very little has been reported on the effect of refrigerant charge or on-line fault diagnosis for liquid-to-liquid chiller systems that are widely used in commercial and process refrigeration applications. This paper presents results of experimental investigations on the effect of refrigerant charge on the performance of a small chiller equipped with plate type refrigerant to liquid evaporator and condenser coils. The objective was to identify parameters sensitive to refrigerant charge to be used in the development of an on-line refrigerant leak detection and diagnosis system.

2. Experimental test facility and test procedure

The experimental investigations were carried out on a small laboratory refrigeration test facility constructed using commercially available off-the-shelf components. It consists of a water-to-water chiller and a PC-based data acquisition system. The chiller incorporates a hermetic reciprocating compressor of 6.63 m^3/h volumetric displacement, plate type condenser and evaporator coils and a thermostatic expansion valve. Refrigerant R404a was used as the working fluid.

To facilitate tests at evaporating temperatures below 0 °C, a water/glycol secondary loop was incorporated on the evaporator side of the system. Water and refrigerant temperatures were measured before and after each refrigerant circuit heat exchanger. The temperatures were measured using K-type thermocouples of a range −25 °C to 200 °C. Two pressure transducers of two different ranges, 0–10 bar for the low pressure side and 0–25 bar for the high pressure side, were employed to measure refrigerant pressure at compressor suction and discharge. The thermocouples and pressure transducers were mounted on specially designed instrumentation modules inserted in the pipework, enabling the direct measurement of refrigerant temperatures and pressures. The water flow rate in the condenser and glycol flow rate in the evaporator was measured with turbine type flow meters. The power consumption of the compressor and other electrical parameters such as voltage, current and power factor were measured with a 3-phase power analyser. A schematic diagram of the test facility with the main instrumentation points is shown in Fig. 1.

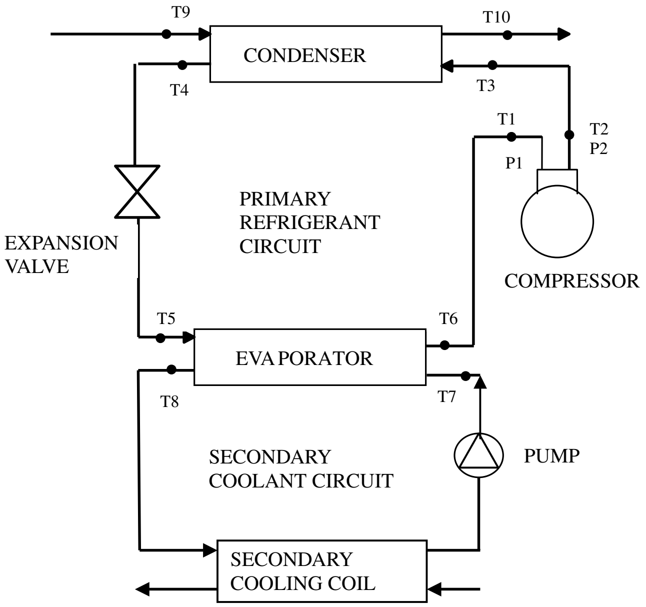
All instruments were calibrated carefully and the uncertainty in the measurements was as follows: temperature ±0.1 K (0.8–1.3%), pressure ±0.03 bar (0.18%), compressor power ±0.01 kW (0.6%) coolant flow rate ±0.02 kg/s (5%). The uncertainties in individual measurements lead to the following uncertainty in the calculated performance parameters of the refrigeration system. Cooling capacity ±0.6 kW (15.0%) and COP ±0.22 (9.0%).

The investigations were carried out for condenser coolant temperatures of 30.0 °C and 35.0 °C and a single evaporator glycol inlet temperature of 12 °C with constant glycol mass flow rate of 0.12 kg/s. The refrigerant charge was varied from 0.60 kg to 1.70 kg compared to a design system charge of approximately 1.20 kg, representing a range between 50% undercharge to 40% overcharge conditions.

3. Results and discussion

3.1. System performance

The system cooling capacity for a range of charge levels is shown in Fig. 2. It can be seen that the cooling capacity is fairly constant between charge levels of 0.9 kg and 1.5 kg, representing a range



TEMPERATURES
T1 – Compressor suction temperature
T2 – Compressor discharge temperature
T3 – Refrigerant temperature at inlet to condenser
T4 – Refrigerant temperature at condenser outlet
T5 – Refrigerant temperature at evaporator inlet
T6 – Refrigerant temperature at evaporator outlet
T7 – Glycol temperature at evaporator inlet
T8 – Glycol temperature at evaporator outlet
T9 – Coolant temperature at condenser inlet
T10 – Coolant temperature at condenser outle

PRESSURES
P1 – Compressor suction pressure
P2 – Compressor discharge pressure

Fig. 1. Schematic diagram of refrigeration test unit.

between 25% undercharge and 25% overcharge. At charge levels below 25% undercharge the cooling capacity falls very rapidly and at 50% undercharge it drops down to 50% of its maximum value of 4kW. At overcharge levels greater than 25% (1.5kg) the cooling capacity also begins to fall slowly, dropping by 7% from its maximum value at 40% overcharge conditions (1.7kg).

It can also be observed, as it would normally be expected, that the cooling capacity is slightly higher for the lower condenser water leaving temperature of 30.0°C compared to 35.0°C, particularly at lower charge levels. The lower condenser temperature leads to a higher cooling capacity due to a higher compressor volumetric efficiency and a greater refrigerating effect in the evaporator. As the charge level is increased, however, the condenser temperature and pressure are forced to rise to maintain adequate heat rejection in the condenser and ensure complete condensation of the refrigerant. As a result, the refrigerating effect and refrigerant mass flow rate are reduced causing a convergence in the cooling capacity of the system for the two condenser water outlet temperatures.

The variation of the suction (evaporator) and discharge (condenser) pressure with charge level is shown in Figs. 3 and 4 respectively. It can be seen that the higher condenser temperature results in higher suction pressures for all charge levels and that the suction pressure reduces as the charge

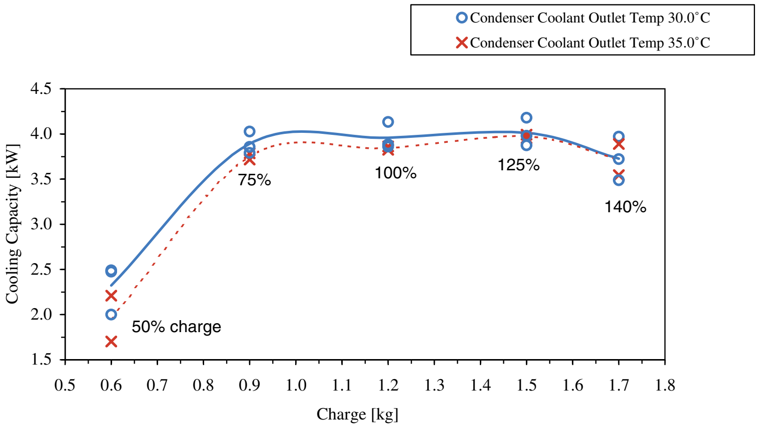


Fig. 2. Variation of cooling capacity with charge level.

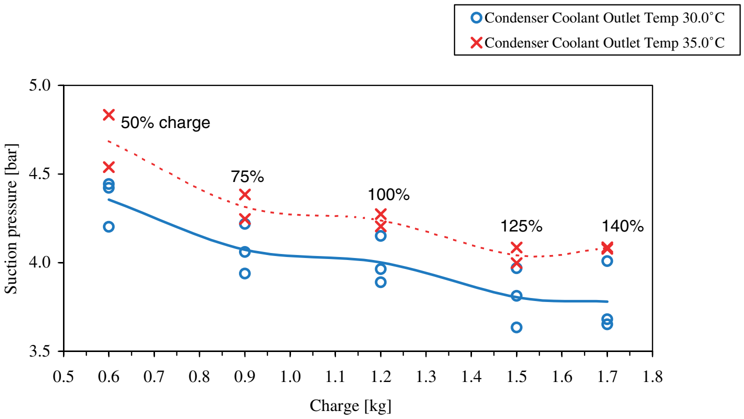


Fig. 3. Variation of suction pressure with charge level.

level is increased. In comparison to the design charge level of 1.2 kg, the suction pressure is around 10% greater at 50% undercharge (0.6 kg), 2% greater at 25% undercharge (0.9 kg), 5% lower at 25% overcharge (1.5 kg) and between 7% and 9% lower at 40% overcharge (1.7 kg). At low charge, the refrigerant mass flow rate through the evaporator is low and cooling capacity is low as a result. This leads to higher evaporating temperatures and suction pressures. As the charge increases, the mass flow rate rises and cooling capacity increases. The evaporating temperature and suction pressure fall as a result.

From Fig. 4 which shows the discharge pressure for the two condenser temperature conditions, it can be seen that the pressure decreases slightly at low charge, below 25% undercharge, it remains fairly constant between 25% undercharge and 25% overcharge but increases exponentially above 25% overcharge. At low charge levels, the discharge pressure is low due to the low mass flow rate in the condenser and the resulting low heat rejection. An increase in charge results in an increased refrigerant mass flow rate in the condenser and an increase in discharge pressure.

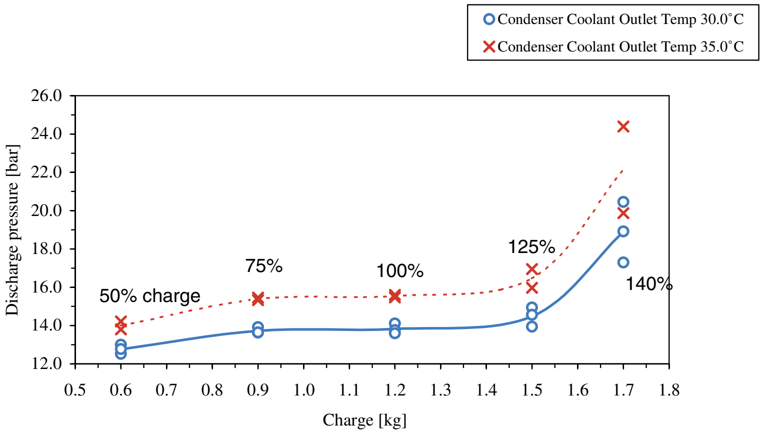


Fig. 4. Variation of discharge pressure with charge level.

At the design condition, the discharge pressure remains fairly steady. As more refrigerant charge is added, the excess refrigerant is stored in the condenser and the pressure rises significantly.

Fig. 5 shows the refrigeration cycle for various charge levels plotted on a *P–h* diagram for the 35 °C condenser water leaving temperature. It can be seen that the impact of charge level is much more pronounced on the condensing temperature and pressure than on the evaporating temperature and pressure. It can also be seen that for the conditions tested, the system operates with incomplete condensation at charge levels below 25% of design value. Also, moving from 25% undercharge to 25% overcharge provides only a small change in the cycle on the *P–h* diagram.

Fig. 6 shows the variation of the compressor power consumption with charge level for the two condenser water outlet temperatures. Staring from low charge conditions, the power consumption increases slowly with charge up to about 25% undercharge conditions. It then remains fairly constant but begins to increase more steeply above 25% overcharge conditions. At low charge, the refrigerant mass flow in the evaporator is low and the evaporator temperature and pressure is higher than at the design charge level (see Fig. 3). The refrigerant mass flow in the condenser is also low and the condenser temperature and pressure is relatively low with reduced heat rejection

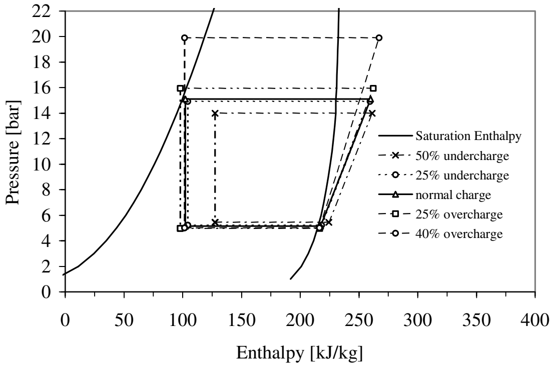


Fig. 5. Pressure–enthalpy diagram for various charge levels.

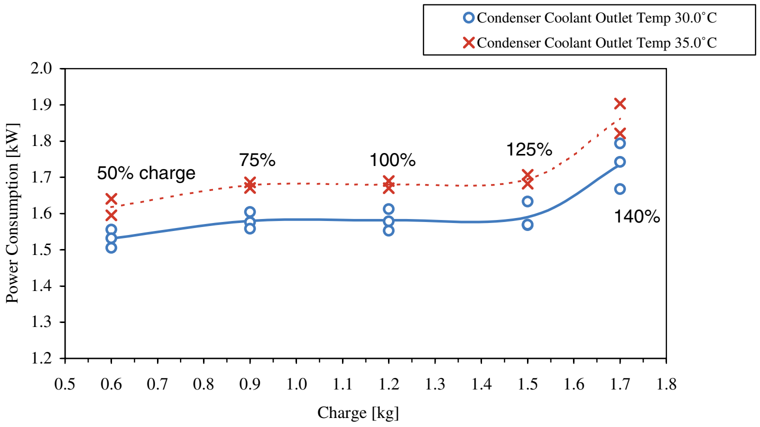


Fig. 6. Variation of compressor power consumption with charge level.

(Fig. 4). This leads to a low pressure ratio and together with the low mass flow rate result in a low compressor power consumption. The mass flow rate increases with refrigerant charge and the evaporator temperature and pressure decrease slightly whilst the condenser temperature and pressure rise more substantially. The power consumption increases as a result of the increased mass flow rate and pressure ratio. As the charge approaches the normal level, the mass flow rate and pressure ratio remain relatively constant, resulting in fairly steady power consumption. Further increases in charge above the normal level lead to an increased mass of refrigerant stored in the condenser and a rise in condenser pressure. This results in higher power consumption due to the increased pressure ratio.

Fig. 7 shows the variation of the coefficient of performance (COP) with charge level. The COP remains fairly constant between 25% undercharge and 25% overcharge but drops by 45% at 50% undercharge and by 13% at 40% overcharge conditions. The COP for the condenser water leaving temperature of 30 °C is consistently higher than that at 35 °C, as expected, due to the impact of the increased pressure ratio on power consumption.

Figs. 8 and 9 show the system superheat and sub-cooling for the two condenser coolant temperatures. The superheat is shown to remain relatively constant between 25% undercharge and 25% overcharge, and to increase significantly at 50% undercharge. There is no significant difference between the results for the two condenser temperatures in the range between 25% undercharge and 25% overcharge indicating good control by the thermostatic expansion valve in this range. The high superheat at low charge levels is a result of insufficient refrigerant in the evaporator which leads to earlier evaporation and a larger evaporator heat transfer area devoted to superheating refrigerant vapour.

From Fig. 9 it can be seen that sub-cooling at the condenser outlet is zero up to normal charge conditions and increases with overcharging. At 25% overcharge, sub-cooling is around 2 K and increases to above 9 K at 40% overcharge. Sub-cooling is low at low charge levels due to the low refrigerant mass flow rate in the condenser resulting in a condenser outlet condition which is saturated or in the two-phase region. As the charge level is increased, the refrigerant mass flow rate increases and the condenser pressure rises in response to the increased mass of refrigerant stored in the condenser. This results in higher refrigerant liquid sub-cooling (see Fig. 5).

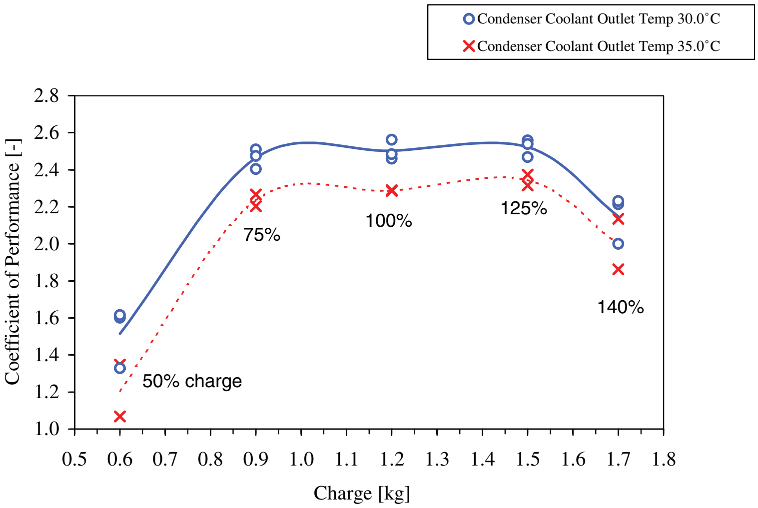


Fig. 7. Variation of coefficient of performance with charge level.

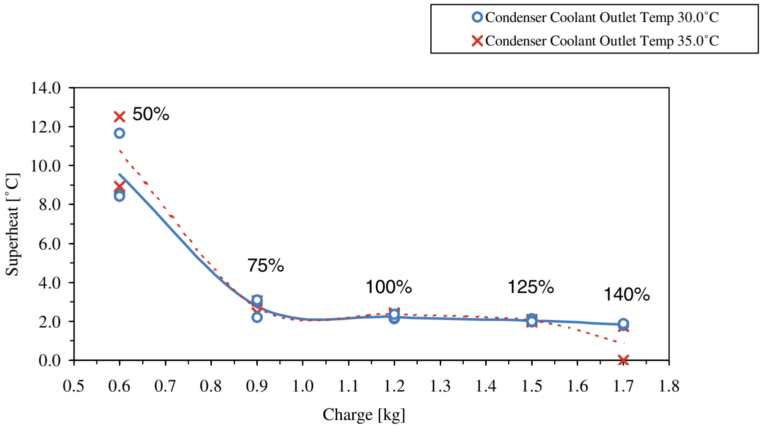


Fig. 8. Variation of degree of superheat with charge level.

3.2. Charge sensitive parameters for refrigerant leak detection

The results presented in the previous section illustrate that for the system tested, significant deterioration in performance occurs at charge levels greater than 25% overcharge and less than 25% undercharge. Maintaining refrigerant charge in the range between 25% undercharge and 25% overcharge, performance deterioration from normal is limited to 2% for COP, 2% for cooling capacity and 1% for power consumption. By the identification and on-line monitoring of parameters sensitive to charge levels lower than 25% undercharge and greater than 25% overcharge, system performance can be guaranteed within these performance limits.

The results also show that superheat is sensitive to low system charge. For charge levels between 25% undercharge and 25% overcharge, superheat is between 2 K and 3 K. At 50% undercharge the

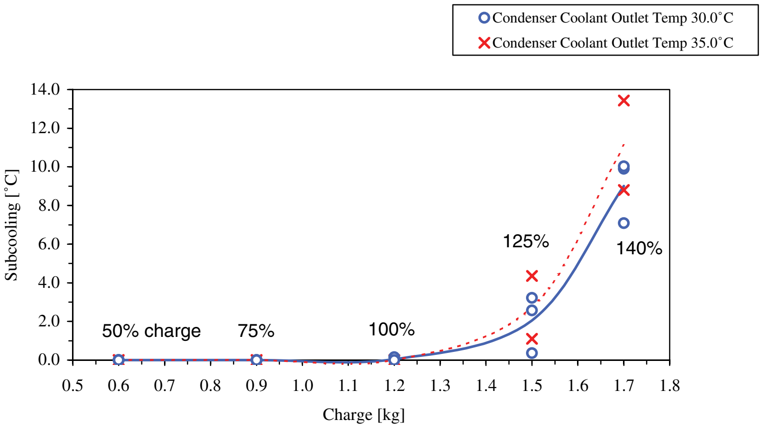


Fig. 9. Variation of degree of sub-cooling with charge level.

superheat rises to approximately 10 K. On-line monitoring of superheat therefore may enable charge levels below 25% undercharge to be detected.

Sub-cooling has been shown to be sensitive to high system charge levels. For normal and lower charge levels the degree of sub-cooling is approximately 0 K. At 25% overcharge, sub-cooling rises to between 2 K and 3 K and for 40% overcharge to over 9 K. The sensitivity of sub-cooling to charge level enables overcharging above 25% to be detected by on-line monitoring of this parameter.

It should be noted, however, that both superheat and sub-cooling are derived parameters and must be calculated from more than one instrumentation measurement. In both cases, the simplest approach is to use the suction and discharge pressure measurements to identify the corresponding saturation temperatures. The sub-cooling and superheat can then be determined from the difference between the condenser or evaporator outlet temperature and the saturation temperature.

Discharge pressure has also been shown to be a function of system charge level. At 50% undercharge, discharge pressure is approximately 10% lower than normal, compared to 1% lower at 25% undercharge. By monitoring discharge pressure, charge levels below 25% undercharge may be detected. For excess charge, discharge pressure is highly sensitive, with values approximately 5% greater than normal for 25% overcharge and 40% greater for 40% overcharge. Discharge pressure monitoring would, therefore, enable overcharging in excess of 25% to be detected.

Even though the results have shown that the suction pressure decreases as the charge level is increased, the variation between 50% undercharge and 40% overcharge is only around 17%. Suction pressure is therefore not sufficiently sensitive to be used on its own to predict the charge level. However it may be a useful indicator of general overcharging and may be used to reinforce a diagnosis developed from one or more other charge sensitive parameters.

In contrast to superheat and sub-cooling monitoring, suction and discharge pressures are measured directly and do not require any processing or manipulation. However, suction and discharge pressures are also a function of the heat source and heat sink temperatures which would require compensation in order to calculate divergence from the normal level, whereas superheat and sub-cooling are relatively independent of heat source and sink temperatures.

4. Conclusions

The following conclusions can be drawn from this investigation:

1. The level of refrigerant charge can have a significant impact on the performance of refrigeration systems. How sensitive the system performance is to charge level will depend on the design of the system and the type of heat exchangers used and their refrigerant storage capacity. For the system tested, which employed plate type condenser and evaporator coils, it was found that its performance remained fairly constant within a charge range of ±25% from the design value. Outside this range the performance dropped significantly.
2. The use of an on-line leak detection system can ensure that overcharging or refrigerant leakage is detected and an alarm raised at an early stage to maintain system performance and prevent system failure. In a system that employs on-line monitoring, a relatively inexpensive leak detection system can be implemented using the degree of superheat and degree of sub-cooling as the detection parameters. These can be easily determined from pressure and temperature measurements using simple algorithms to calculate the refrigerant saturation temperature corresponding to the measured pressure. In its simplest form such a system will require only two pressure and two temperature sensors at the condenser and evaporator outlet. Superheat may be monitored to detect charge levels below 25% undercharge and sub-cooling may be used to indicate charge levels greater than 25% overcharge.
3. Other faults in the system such as heat exchanger fouling may have similar effects on superheating and sub-cooling as refrigerant charge. To discriminate between faults, systems are now being developed by a number of researchers based on expert knowledge and artificial intelligence techniques.

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