WHAT IS THE OPTIMUM COMPRESSOR DISCHARGE PRESSURE SET POINT FOR CONDENSERS?

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ABSTRACT

Optimisation of condenser set points to minimise energy use requires a tradeoff between high compressor energy use at high head pressures and high condenser fan and pump energy use to achieve low head pressures. It is shown that for most condenser selections used in New Zealand industrial applications, floating the head pressure is the best strategy. Multi-speed fans and variable speed drive (VSD) fan controls only give significant energy use reductions compared with on/off control if compressors operate highly unloaded and/or the condenser is grossly oversized. Oil separators, discharge and high pressure liquid lines, and expansion and other refrigerant control valves should be designed to operate satisfactorily across the full range of discharge pressures likely to be encountered if discharge pressure is floated.

Keywords: condenser, energy use, pressure, compressor

1 - Introduction

Most industrial refrigeration systems employ compressor discharge (head) pressure controls. Generally these controls modulate the condenser fans (for air-cooled or evaporative condensers) or water flow rates and cooling tower fans (for water-cooled condensers) to keep the head pressure within a specified range. Reducing fan speed or cooling water flow reduces the effective capacity of the condensers so that it equals the required heat rejection by maintaining a larger temperature difference between the refrigerant saturated condensation temperature (SCT) and the ambient wet bulb (WB) temperature. The compressor discharge pressure adjusts to equal the pressure corresponding to the SCT plus the discharge line pressure drop and the partial pressure of any non-condensable gases present in the refrigerant. For simplicity, it is common practice to express compressor discharge conditions in terms of the equivalent SCT rather than pressure terms based on the unique pressure-temperature relationship for each refrigerant.

It is well known that lowering the compressor head pressure has only a small beneficial effect on compressor and hence overall system cooling capacity, but that it significantly reduces compressor energy use for the same amount of cooling. Figure 1 shows these effects for a typical screw compressor relative to operation at a SCT of +35ºC (for -2ºC SET). Generally, the energy use reduces by 2 to 3% for every 1ºC reduction in SCT. However, to achieve lower SCT requires increased condenser fan and/or pump power. Often head pressure set points are kept high because it is believed that the extra condenser energy required to reduce head pressure is greater than the resulting reduction in compressor energy.

This paper investigates the optimal head pressure for compressor-condenser designs and operating conditions commonly encountered in New Zealand. It also considers constraints to lowering the head pressure and compares alternative condenser fan controls. The paper focuses on screw compressors and evaporative condensers used in industrial refrigeration applications but the same principles also apply to other types, sizes and combinations of equipment (eg. air cooled condensers with reciprocating compressors in commercial refrigeration).

Design and operating conditions

2.1 - Design conditions

In New Zealand a common evaporative condenser design condition used is the ambient wet-bulb (WB) temperature exceeded only 1% or 5% of the time (IRHACE 05). The coincident dry-bulb (DB) temperature is often of interest if the cooling load is highly temperature dependent. An alternative is the WB temperature coincident with the 1% or 5% summer extreme dry bulb temperature which often corresponds to the maximum cooling load (ASHRAE 01). Table 1 gives both types of 1% design wet-bulb temperatures for some NZ locations. Note the significant difference between the two sources of data and the alternative design WB definitions.

Most often the condensers are selected to handle the total refrigeration system heat of rejection when all compressors are operating with a SCT between 35ºC and 40ºC (occassionally the design SCT can be as low as 30ºC or as high as 45ºC but it is seldom more extreme) and at suction conditions equivalent to the design saturated evaporation temperature (SET). Hence, if the design ambient wet bulb temperature is 20ºC, a compressor with a refrigeration capacity of 1000 kWr at design conditions of -2ºC/+35ºC (SET/SCT) and motor input power of 260 kWe might be matched with condenser that can reject slightly more than 1260 kW of heat with 15ºC temperature difference between SCT and WB.

From a design perspective, it should be noted that in many cases axial fan condensers provide the same heat rejection capacity with lower fan power than similar condensers using centrifugal fans. The main advantage of centrifugal fans is lower noise.

2.2 - Operating conditions

The design condition is often an extreme both in terms of cooling load and ambient WB conditions. For most of the time a refrigeration system will operate with a lower refrigeration load and less extreme ambient conditions. Hence the condenser often has excess capacity at the design SCT and can operate either part-loaded or with a SCT lower than the design condition. Table 2 gives estimated average monthly WB temperatures for a number of locations in New Zealand.

2.3 - Condenser fan modulation

Most evaporative condensers used in New Zealand have a modular design with a separate fan for each module. Often a number of fans are connected to a common motor drive (eg. two motors each driving two fans in a four module condenser). There are three main ways that condenser fans are modulated:

- On/Off – fans are turned off sequentially.
- Two-speed fans – if multiple two-speed fans are installed then the controller can ramp through a list of combinations in order of increased air flow (e.g. for two fans the order might be - both off; one on low speed; both on low speed; one on high speed; one on low plus one on high speed; both on high speed).
- Variable speed drives (VSDs) – fan speed is continuously ramped up or down using VSDs in proportion to the required change in condenser capacity and discharge pressure.

Figures 2 and 3 show the effect of fan loading/speed and the different fan controls on condenser heat rejection capacity and fan power. With all fans off (but water circulation pumps operating) a condenser still has about 10% of full load capacity due to natural convection of air. For on/off control capacity and energy are both proportional to fan loading. For multi-speed and VSD fans, the power use is proportional to fan speed cubed and heat rejection capacity is proportional to fan speed to the power of 0.76 [Manske 99; Manske 01]:

$$\text{fan power} \propto (\text{fan speed})^3 \quad (1)$$

$$\text{condenser capacity} \propto (\text{fan speed})^{0.76} \quad (2)$$

For multi-speed fans, there are a limited number of steps. To achieve a capacity intermediate to available steps requires switching between the adjacent fan speed steps. Therefore the effective relationship between condenser capacity and fan power is linear between the steps as shown in Figures 2 and 3 for two two-speed fans. Note that for the two two-speed fans, the option of one fan off and the other on high speed (off, high) is less energy efficient and cost-effective than the other options (it is better to have all of the condenser operating part-loaded than part of the condenser with no air flow and part with high air flow). In general, for multiple multi-speed fans, options that have different fans more than one speed step apart should be avoided.

3 – Case studies

3.1 - Normal condenser selection

Consider a design with a twin screw ammonia compressor with 1000 kWr capacity at -2ºC/+35ºC (SET/SCT) and an evaporative condenser that can reject 1300 kW at +35ºC/20ºC (SCT/WB). The condenser has two 10 kW fans and a 3 kW water pump. Figure 4 shows the total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for the fully loaded compressor at different ambient WB temperatures with on/off condenser fan control. Figure 5 shows the total power as a function of discharge pressure and SCT for an ambient WB of 12ºC at different compressor percentage loadings with on/off condenser fan control. The left hand end of each line represents the condenser fans operating fully loaded (floating head pressure). Figure 6 is the same as Figure 4 but with VSD condenser fan control. Figure 7 compares on/off, two-speed and VSD fan controls for the fully loaded compressor with an ambient WB of 8ºC or 16ºC.

Generally, the most energy-efficient mode of operation for such a design, if the compressor operates fully loaded, is to keep the fans fully loaded and let the discharge pressure float. VSD controlled and multi-speed fans offer little energy use advantage over on/off controlled fans unless:

- the compressor is operated partly loaded (see Section 3.2),

or

- the discharge pressure set point is greater than about 1000 kPa.g (28ºC SCT) or WB is greater than about 12ºC.

However, with VSD compared with on/off or two-speed controls, the minimum total power can be achieved at higher discharge pressures (by about 50 kPa).

Location	WB or WB/MDB (ºC)		DB or DB/MWB (ºC)	
	1% IRHACE	1% ASHRAE	1% IRHACE	1% ASHRAE
Auckland	22.5	20.4/22.9	27.9	24.2/19.1
Hamilton	22.8		28.0	
Tauranga	22.4		27.7	
Napier	22.7		30.2	
Palmerston North	21.2		27.6	
Wellington	20.4	18.3/20.7	25.5	21.9/17.4
Nelson	20.7		25.9	
Christchurch	21.3	17.6/23.6	29.8	26.1/16.2
Dunedin	18.9	15.4/17.4	24.2	18.9/13.8
Invercargill	19.6		25.0	

Table 1 – 1% design weather conditions for some New Zealand locations (IRHACE 05; ASHRAE 01). WB = wet bulb, DB = dry bulb, MDB = mean coincident DB, MWB = mean coincident WB.

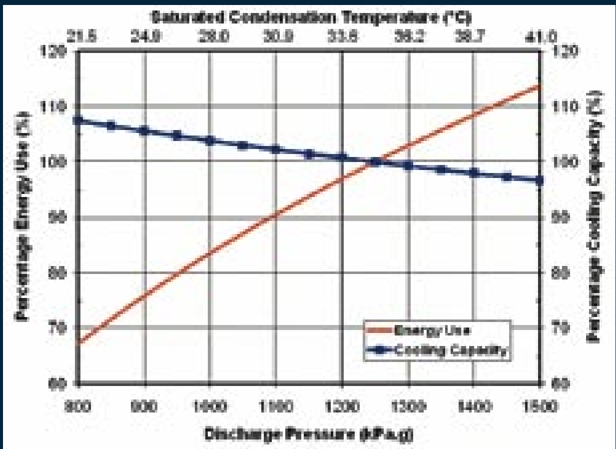

Figure 1 – change in cooling capacity and energy use as a function of discharge pressure (and SCT) for a typical fully-loaded ammonia screw compressor relative to that at 1250 kPa.g (+35ºC) with a suction pressure of 300 kPa.g (-2ºC SET equivalent).

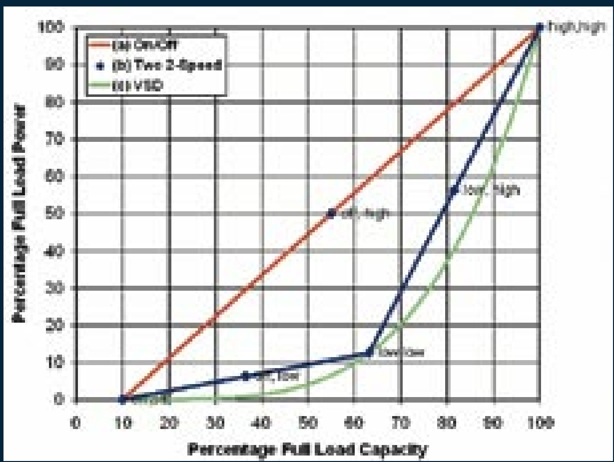

Figure 2 – the relationship between percentage of full load heat rejection capacity and percentage of full load fan power for different condenser fan controls: (a) on/off fan control, (b) two 2-speed fans, and (c) VSD fans.

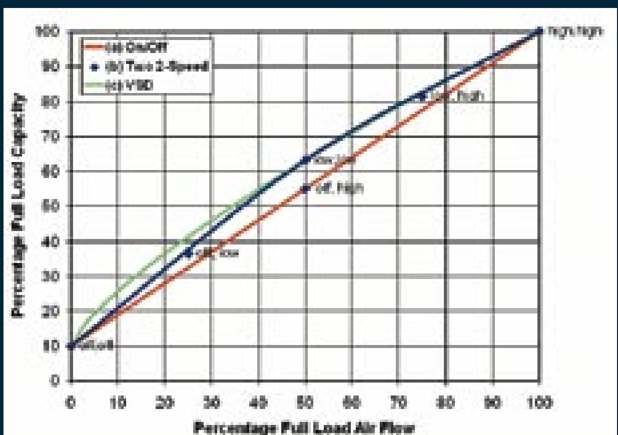

Figure 3 – the relationship between percentage of full load air flow and percentage of full load heat rejection capacity for different condenser fan controls: (a) on/off fan control, (b) two 2-speed fans, and (c) VSD fans.

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Auckland	16.7	17.3	16.2	14.3	12.3	10.6	9.8	10.0	10.8	11.8	13.3	15.2
Hamilton	16.9	16.9	15.5	13.3	11.0	9.2	8.4	9.1	10.1	11.8	13.7	15.5
Tauranga	16.2	16.4	15.5	13.5	10.9	9.2	8.3	8.7	9.9	11.3	12.7	14.4
Napier	14.6	15.3	14.2	12.4	10.2	8.6	7.7	8.0	8.9	10.1	11.4	13.0
Palmerston N.	15.3	15.8	14.3	12.2	10.0	8.2	7.6	8.0	9.1	10.6	12.0	13.8
Wellington	15.7	15.8	14.2	11.7	9.1	7.2	6.4	7.2	8.7	10.4	12.1	14.1
Nelson	13.7	13.8	12.9	11.2	9.2	7.3	6.7	7.0	8.3	9.5	10.7	12.5
Christchurch	12.4	13.1	10.7	8.1	4.9	1.7	0.9	2.4	5.0	6.9	8.6	10.5
Dunedin	13.8	14.2	12.0	9.1	5.9	3.1	2.8	4.2	6.5	8.4	10.3	12.3
Invercargill	12.1	12.3	11.1	9.2	7.0	4.8	4.4	5.4	6.9	8.2	9.4	11.0

Table 2 – estimated average monthly ambient wet bulb (WB) temperatures (ºC) for some New Zealand locations (based on NIWA average relative humidity at 9 am and average monthly DB temperature).

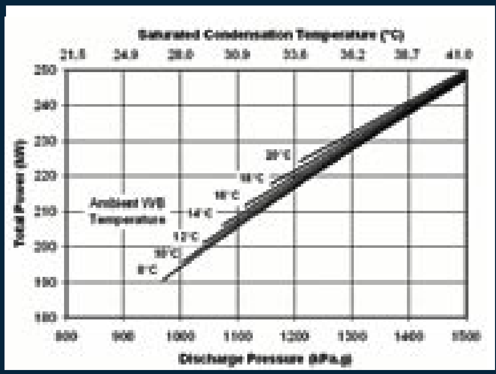

Figure 4 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at different ambient WB temperatures with on/off condenser fan control (normal condenser selection).

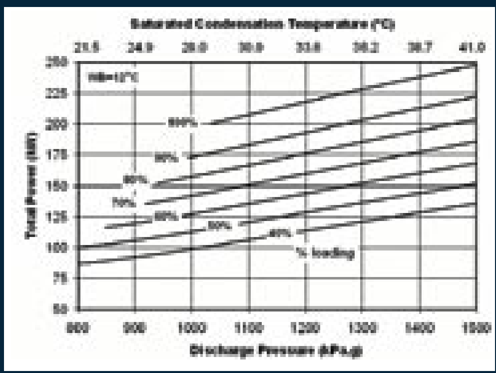

Figure 5 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for an ammonia screw compressor at different percentage loading, with an ambient WB temperature of 12ºC and on/off condenser fan control (normal condenser selection).

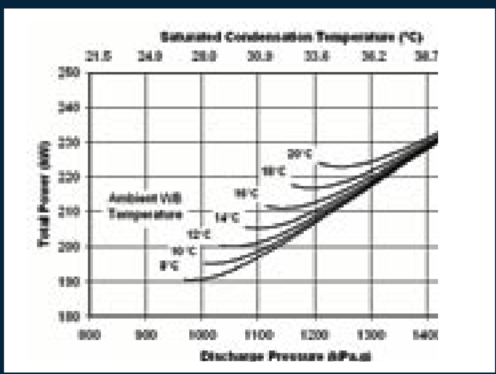

Figure 6 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at different ambient WB temperatures with VSD condenser fan control (normal condenser selection).

3.2 - Conservative condenser selection

Now consider a design with a twin screw ammonia compressor with 1000 kWr capacity at -2ºC/+35ºC (SET/SCT) and an evaporative condenser that can reject 1300 kW at +28ºC/20ºC (SCT/WB). The condenser has four 10 kW fans a 6 kW water pump. Compared with the normal design, this condenser has about twice the heat rejection capacity (ie. the heat rejection at +35ºC/20ºC SCT/WB would be about 2440 kW). Note that this design behaves very similarly to the normal design selection operating with the compressor at 50% of full load.

Figures 8 to 11 are the equivalent to Figures 4 to 7 for the conservative condenser selection (or the normal selection but with about half the cooling load and hence about 50% loading for the compressor). With on/off control, floating discharge pressure is still the most energy-efficient option if the compressor is fully loaded. However, for multi-speed and VSD control, there is now a distinct optimum discharge pressure at all conditions. For this example, the optimum corresponds to the VSD fans being operated at about 70% to

75% of full speed. Comparison of Figures 8, 10 and 11 shows that for the same discharge pressure VSD control reduces total energy use by up to 20 kW. Also, the lowest total energy use is achieved at a higher discharge pressure than the optimum for on/off control so any constraints to lower discharge pressures discussed below are less likely to occur. For a part-loaded compressor there is an optimum even with on/off control but the difference from floating head pressure is quite small (Figure 9).

Figures 10 and 11 show that the optimum discharge pressure for VSD controlled fans depends on the ambient conditions and is not a constant value suggesting the set point discharge pressure should be set relative to ambient WB conditions (for the example, the optimum discharge pressure set point corresponds to a SCT about 10ºC to 15ºC above the ambient WB temperature). A set point that changes relative to ambient WB conditions is one possible way to optimise condenser controls.

Constraints and advantages

Possible operating constraints that may arise as discharge pressures are reduced include:

(a) Reduced oil separator efficiency – At lower discharge pressure volumetric vapour flow rates will increase and it is possible that oil separators may become less effective leading to oil management problems.

(b) Excessive discharge line pressure drops - At lower discharge pressure volumetric vapour flow rates will increase and pressure drops in discharge lines may increase sharply. Discharge lines should be sized for minimum, not maximum, discharge pressures.

(c) Flash gas formation – If condensation temperatures become lower than air dry-bulb (DB) temperatures then there could be heat gains through uninsulated liquid refrigerant lines resulting in formation of flash gas. This flash gas might affect the performance of expansion and high-side float valves leading to poor control of refrigerant levels and/or flows and hence less stable and poorer evaporator performance.

(d) Inadequate liquid refrigerant supply – At lower head pressures any expansion and high-side float valves may not have sufficient capacity to maintain stable liquid levels or refrigerant flows to the low-pressure parts of the system. Thermostatic expansion valves are particularly sensitive to changes in discharge pressure especially if they are selected for summer design conditions. Problems can include instability, poor evaporator performance or increased risk of liquid carry-over to compressors.

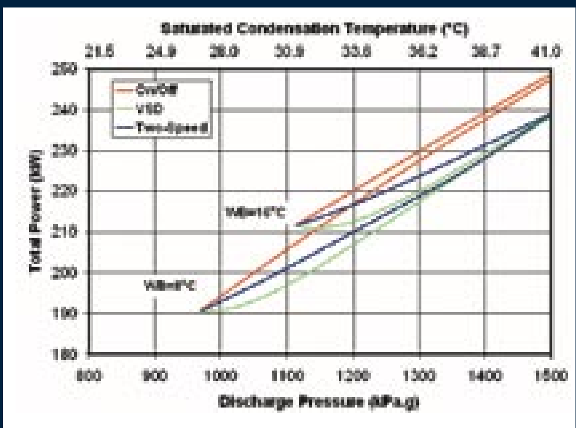

Figure 7 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at 8ºC and 16ºC ambient WB temperature with on/off, two-speed or VSD condenser fan controls (normal condenser selection).

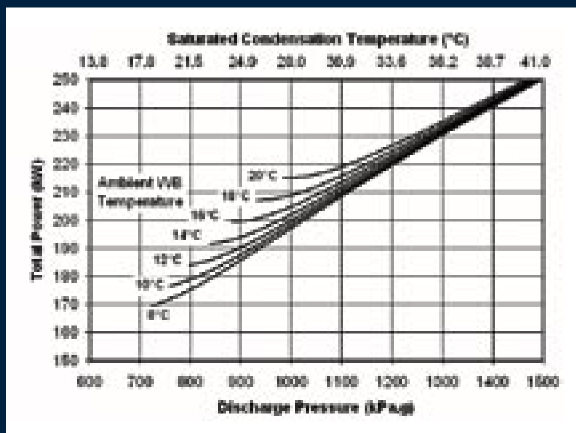

Figure 8 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at different ambient WB temperatures with on/off condenser fan control (conservative condenser selection).

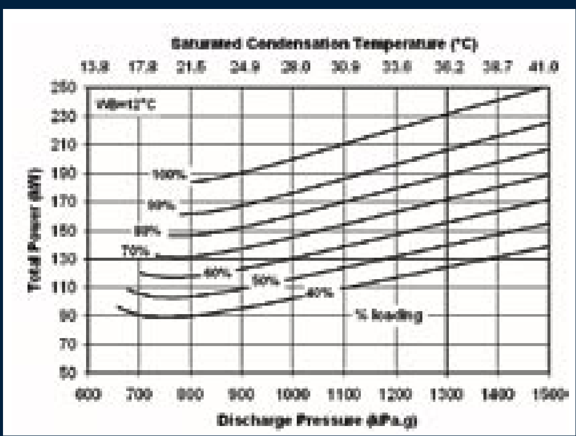

Figure 9 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for an ammonia screw compressor at different percentage loading, with an ambient WB temperature of 12ºC and on/off condenser fan control (conservative condenser selection).

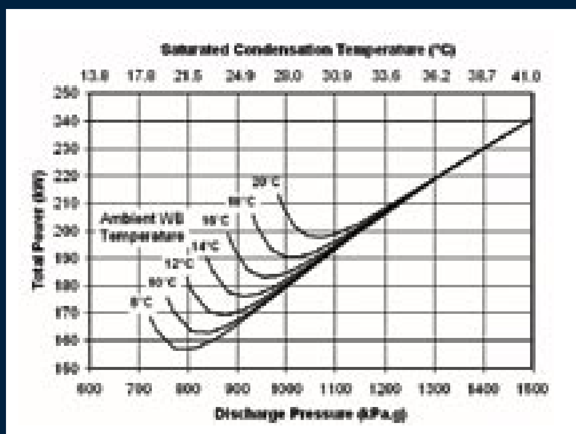

Figure 10 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at different ambient WB temperatures with VSD condenser fan control (conservative condenser selection).

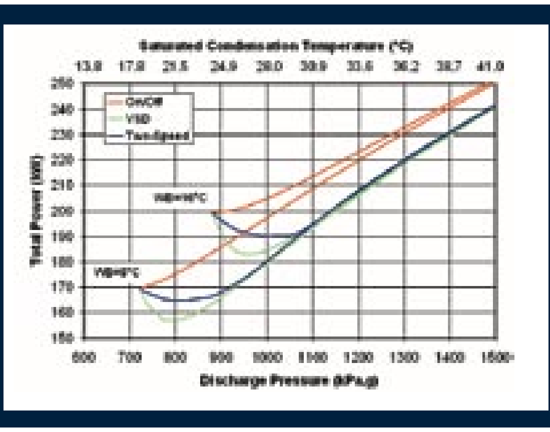

Figure 11 – total power (compressor plus condenser fans and pumps) as a function of discharge pressure and SCT for a fully loaded ammonia screw compressor at 8ºC and 16ºC ambient WB temperature with on/off, two-speed or VSD condenser fan controls (conservative condenser selection).

(e) Compressor Performance - The energy efficiency of some compressors will become very poor if discharge pressure becomes too low relative to suction pressures (e.g. screw compressor with too high a volume ratio). Generally, this is unlikely to occur in practice.

(f) Condenser Performance – Many condensers do not perform close to their rated capacity. Common reasons include: presence of non-condensable gases, liquid logging into parallel condensers, air/water side fouling and air recirculation (short-circuiting). Solutions to these problems include purging, better design of liquid legs for drainage, water treatment, cleaning and improved condenser location.

Rather than accepting these constraints as inevitable, designers and service staff should work to eliminate them. In most cases, low or no cost modifications can allow discharge pressures to be further reduced. Furthermore, additional advantages of operation at lower discharge pressure include:

(g) Compressor Cooling Capacity – As discharge pressure decrease most compressors will have slight increases in cooling capacity (Figure 1).

(h) Discharge Conditions – Lower discharge pressure can result in lower discharge temperatures and hence reduced compressor cooling and maintenance requirements.

(i) Condenser Fan Maintenance – Use of multi-speed or VSD controlled fans can reduce maintenance due to the lower number of fan starts (e.g. problems with belt drives). Also such controls can provide significant noise reduction at lower than full speed.

While the examples used here are large scale systems using evaporative condensers, the same principles apply to smaller scale refrigeration systems using air-cooled or water cooled condensers except that the approach is to the ambient DB or water temperature respectively rather than the ambient WB temperature. Water cooled condensers supplied with water from an evaporative cooling tower behave in a similar way to an evaporative condenser, except there is an extra temperature difference so the optimum temperature difference between ambient WB and SCT will tend to be higher.

Conclusions

Most condenser selections provide lowest total energy use when the discharge pressure is floated (ie. an unattainably low set point is used) unless the compressors operate highly unloaded most of the time. Therefore refrigeration systems should be designed to operate at floating discharge pressure levels and not just the design summer conditions. In particular, oil separators, discharge and high pressure liquid lines, and expansion and other refrigerant control valves should be designed to operate satisfactorily across the full range of discharge pressures likely to be encountered if discharge pressure is floated.

If discharge pressure is floated then multi-speed fans and VSD fan controls offer low energy use reduction compared with on/off control for a system with fully loaded compressors and a condenser sized in accordance with "normal" design practices. However, if compressors are operated highly unloaded, the system has grossly oversized condensers, ambient conditions are very cold, or if a high discharge pressure set point must be used for other reasons, then multi-speed and VSD fan control can provide significant energy use advantages. Optimal set points for such systems depend on the ambient WB conditions. A 10ºC to 15ºC approach to ambient WB temperature appears to be a reasonable starting point for discharge pressure (SCT) set points. ■

References

ASHRAE 01 ASHRAE. 2001. 2001 ASHRAE Handbook—Fundamentals, Chapter 27. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

IRHACE 05 IRHACE. 2005. IRHACE Industry Directory and Handbook 2005, IRHACE Journal, Vol. 16, No. 6. Institute of Refrigeration, Heating and Air-Conditioning Engineers of New Zealand.

Manske 99 Manske, K. A., 1999. Performance Optimization of Industrial Refrigeration Systems, University of Wisconsin-Madison, Masterate Thesis.

Manske 01 Manske, K. A., Reindl, D. T., Klein, S. A., 2001. Evaporative Condenser Control in Industrial Refrigeration Systems, International Journal of Refrigeration, Vol. 24, No. 7, pp. 676-691.

Nomenclature

DB	ambient dry bulb temperature (ºC)
MDB	mean coincident DB (ºC)
MWB	mean coincident WB (ºC)
SCT	saturated condensation temperature (ºC)
SET	saturated evaporation temperature (ºC)
WB	ambient wet bulb temperature (ºC)

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