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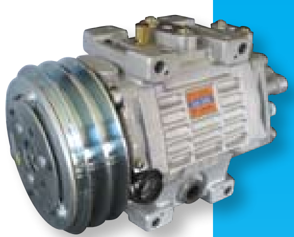
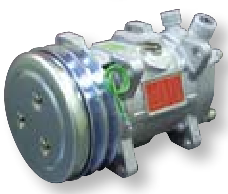


Compressor Selection Criteria



Compressor Selection Criteria

Table of contents:

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Introduction

System design, including the compressor to be used, is based on a heat load calculation conducted for the environment to be cooled and dehumidified. In professional design this is done using test vehicles in environmental chambers where heat loads can be controlled, together with, and based on, theoretical calculation.

Technicians reading this bulletin must be familiar with reading a pressure/temperature relationship chart to identify evaporating and condensing temperatures from gauge readings.

Definitions

Heat

Heat is a form of energy. In this application heat will be defined as both temperature and humidity contained within a nominated environment. *(ie - the cabin).*

Capacity

The capacity of the system is the **amount** of heat that the system is capable of removing from a nominated environment *(ie - the cabin)*. This includes the cooling and dehumidification of that environment.

Enthalpy

Enthalpy is the sum of the energy *(heat)* applied to a medium. Enthalpy is often referred to as the **total heat content** of a substance.

In this application Enthalpy will relate to the amount of heat absorbed by the refrigerant as it passes through the evaporator.

The purpose of this booklet is to enable technicians using Unicla compressors to gain an appreciation of system design and the importance of correct compressor selection.

This will ensure adequate **capacity** of the system to cater for the heat loads under which the system will operate.

Relationship between cooling capacity and heat load

When entering a vehicle that has been parked in the sun or when a vehicle is operating in areas of high temperature and/or high humidity the heat load is high.

"Heat load" is temperature and humidity.

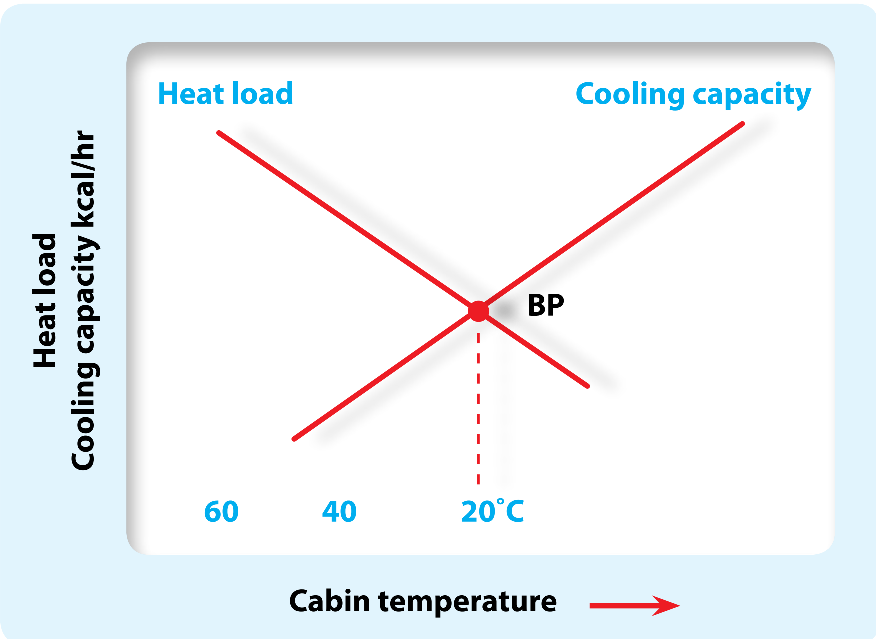
As the system continues to operate the heat load will reduce as the air is cooled and dehumidified.

With the high heat loads the cooling capacity of the system is inadequate to instantly reduce the cabin temperature down to a desirable level *(ie - 20ºC)*. With the continued operation the progressive reduction in heat load will enable the cooling capacity of the system to increase *(with respect to attaining desired cabin temperatures)*.

When we reach balancing point *(B.P.)* the capacity of the system exceeds the heat load and we are capable of cooling the cabin to
specified levels or below.

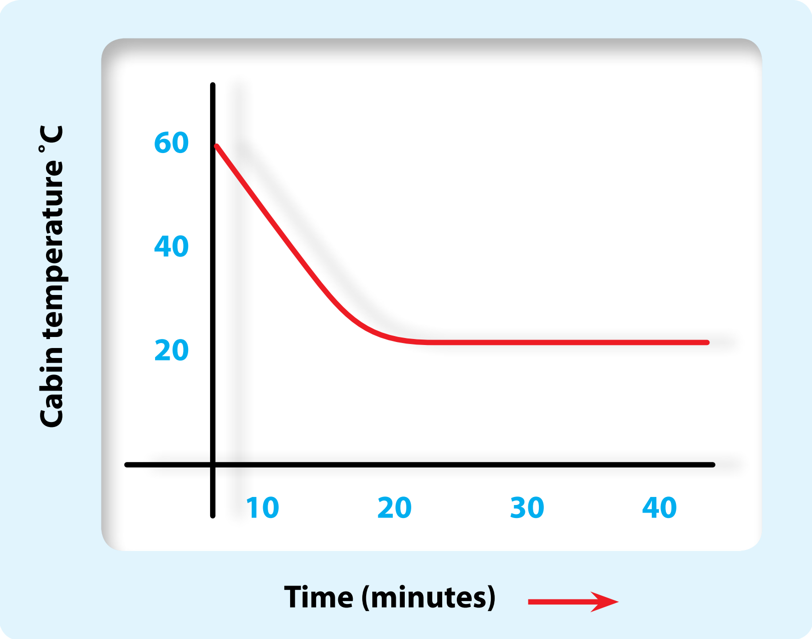
Any system must be capable of reaching B.P. to ensure adequate cabin cooling and customer satisfaction at predetermined heat loads within an acceptable time frame - this is commonly referred to as the pull down test *(see graph 2)*.

Graph 1. Cooling capacity verses heat load



Graph 2. Pull down performance

(Initial cabin temperature 60˚C at 45% R.H.)



System design

To enable B.P. to be reached within an acceptable time frame the capacity of the system must cater for:

- Initial **"heat soak"** *(vehicle parked in direct sunlight),*

and:

- Constant heat loads on the cabin environment,

namely:

- Transmission heat radiated through glass
- Entry heat via floor/firewall etc.
- Entry heat via ventilation *(flow through/natural ventilation)*
- Internal generated heat by human body
- Internal generated heat - vehicle components

Table 1 *(over)* gives an example of the heat quantities for a small/medium passenger vehicle of modern design *(actual data - used as example only)*

Table 1 – heat load quantities

Operating conditions:

- **Ambient temperature - 35ºc**
- **Relative humidity - 60%**
- **Interior temperature - 25ºc**
- **Vehicle speed - 40 km/h**
- **Sunlight conditions - bright**
- **All units in kcal/hr**

Transmitted heat through glass	Entry heat floor firewall etc	Entry heat (ventilation)	Heat generated by occupants (x4)	Internal generated heat	Total heat load
1370	490	320	400	110	2690

To cater for the above heat loads and additional heat soak,

the system must:

- Have refrigerant in the evaporator at a sufficiently low **temperature** to enable it to absorb heat from the cabin air,

and:

- Have a sufficient **quantity** of refrigerant flowing through the evaporator.

Therefore:

Let's work from the evaporator to determine system capacities.

The evaporator is the basis for capacity calculation – it is the component directly responsible for absorbing heat energy from the cabin.

Capacity ratings

Refrigeration capacity has historically been loosely referred to in **tons of refrigeration** based on a 24 hour period.

1 ton of refrigeration is the amount of heat required to melt one ton of ice.

If we are going to use tons of refrigeration as a measurement of the capacity of a system strictly it must be time referenced, normally to an hour rating.

1 Ton of Refrigeration = 288,000 BTU's or 72,528 Kilocalories

This is the amount of energy required to melt the one ton of ice *(based on a 24 hour period)*

therefore: **288,000 BTU's/24 = 12,000 BTU's/hr 72,528 kcal/24 = 3,022 kcal/hr**

We have now arrived at a capacity rating for systems:

1 Ton = 12,000 BTU/hr = 3,022 kcal/hr

Alternatively the capacity may be rated in watts or kilowatts:

1 Ton = 3,517 watts or 3.517 kilowatts

Table 2 - refrigerating capacities – cross reference chart

Tons of refrigeration	.5	1	1.5	2	2.5	3	3.5	4
kcal/hr	1511	3022	4533	6044	7555	9066	10577	12088
Btu's/hr	6000	12000	18000	24000	30000	36000	42000	48000
kW	1.758	3.517	5.275	7.034	8.792	10.551	12.309	14.068

Application chart - Refrigeration capacities

Air conditioning applications R134a at 0°C *(to be used for approximation purposes only)*

.5 Ton	Special application - low capacity *(ice box, small compartment cooling)*
1 Ton	Small/compact vehicles - low heat load - preferably tinted windows
1.5 Ton	Medium vehicles - common evaporator capacity/size
2.0 Ton	Large vehicles - higher capacity - higher heat loads
2.5 Ton	Larger vehicles - station wagons - high glass surface area applications *(ie - agricultural/plant etc)*
Over 2.5 Ton	Special applications only - bus - locomotive etc.

Special notes

Evaporator ratings may be stamped, plated onto the unit or evaporator coil itself, or available ex manufacturer, or specified in parts manuals/promotional literature in Btu's/kW/kcal/hr. Alternatively the part number may relate to the capacity of the unit ie - RT9E may be roof top unit 9 kW capacity – externally equalized TX valve.

Designing the system

Now we have a reference for system capacity we can design and manufacture an evaporator with a rating in accordance with the calculated heat load on the system.

The capacity of the evaporator ie 16,000 BTU/hr 4030 kcal/hr will be determined principally by:

- The surface area *(size, fin design etc)* of the evaporator
- The amount of refrigerant flowing through it *(tube design size, number of runs etc)*
- The temperature differential between the air and the refrigerant
- The volume of air flowing over the fins/tubes

To accurately calibrate evaporators, the heat load and refrigerant temperature are referenced ie:

- 35°C intake air
- 40% relative Humidity (R.H.)
- 0°C vaporization temperature
- 5°C superheat.

Note: The intake air and R.H. are often listed as a Dry bulb and Wet bulb temperature from which a psychometric chart can be used to determine R.H.

Of these factors the critical one is the amount of refrigerant flowing through the evaporator.

If - the refrigerant to air differential is high

and - there is the required volume of air flowing over the evaporator to "load it" with heat

then - the variable that will determine the efficiency and capacity of the unit is the volume of refrigerant flowing through the evaporator coil.

Remember - It is the refrigerant that absorbs the heat - therefore refrigerant flows must be optimised.

System capacity verses flow

With the evaporator at a nominated capacity in accordance with its calculated heat loads we must now design the rest of the system to provide adequate flow of refrigerant to achieve its capacity. We must also ensure the condenser is of adequate capacity to dissipate all heat absorbed from the cabin.

Basic component function

The heat exchangers:

Evaporator - absorbs heat from inside the cabin *(or a nominated environment)*

Condenser - radiate **all** heat absorbed into the refrigerant by the evaporator, in the suction run and in the process of compression *(superheat of compression)*

The flow controllers:

TX valve / Orifice tube - Ensures adequate flow into the evaporator *(and subsequently through the evaporator)* to cater for maximum heat loads – TX flow matches evaporator rating.

Compressor - Must be capable of circulating a sufficient volume of refrigerant to maintain maximum flow rates through the evaporator to achieve full system capacity.

All components are important from a system design/system matching perspective, but it is the compressor that must provide the flow to optimise the efficiency of all components - therefore the compressor selection becomes critical and must be rated to the system in which it will operate.

Compressor Selection

The compressor has 2 principle functions in respect to the circulation of refrigerant.

It must:

- Discharge refrigerant into the condenser against the head pressure
  (high side pressure determined in principle by the ambient temperature)

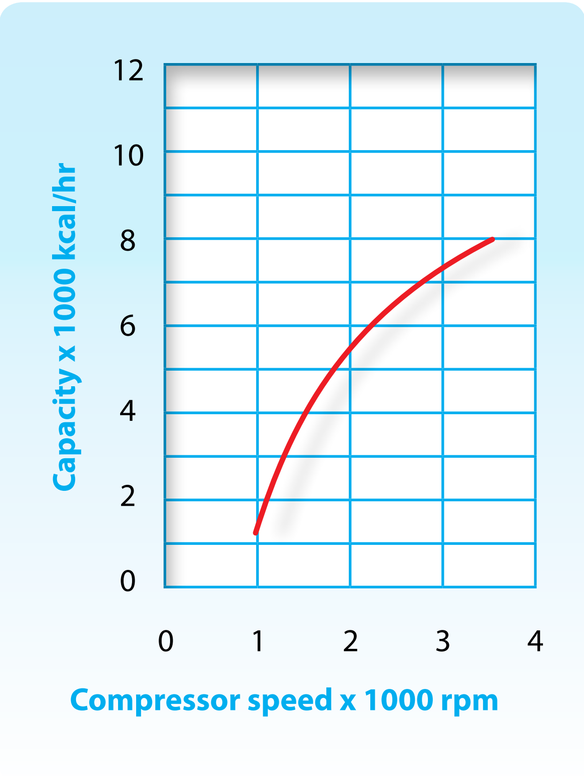
and:

- Pull refrigerant through the evaporator *(via the suction)* to provide a nominated **saturation temperature/pressure** at an adequate flow rate.
  (ie - to pull the refrigerant temperature down)

- Therefore the calibration of a compressor must be against a given standard. The flow rate will vary depending on the conditions in which the compressor is placed *(see over)*.

These conditions are often specified in the compressor performance graphs, which must be referred to in the selection process of the compressor.

Graph 3. Compressor performance example



Operating conditions:

P. Suction	1.86 kg/cm^{2}G
P. Discharge	15.5 kg/cm^{2}G
Super heat	10.0 ˚C
Sub cool	5.0 ˚C
Evap. temp.	0.0 ˚C

Specifications:

Model	sample
Displacement	200 cc/rev
Cylinders	10
Max speed	4500 rpm

1 kg/cm2 = 100 kPa (approx)
= 14.7 PSI (approx)

From graph 3 we can now relate compressor performance to system design and specifications and the conditions under which it will operate.

So looking at each element one item at a time:

Discharge Pressure

The specified discharge pressure in this example is 15.5 kg/cm^{2}G *(1520kPaG - 220psig)*. In an R134 application this corresponds to a condensing temperature of 65˚C *(R134a)*.

Therefore:

This condenser is calibrated to deliver its specified performance under a condensing pressure of 15.5 kg/cm^{2}G *(60˚C condensing temperature - refer to pressure/temperature chart)* which using industry standards relate to an ambient *(day)* temperature of approximately 30˚C.

Therefore:

This compressor will deliver performance, as specified on a 30ºC day *(ambient)* providing condensing is adequate.

A drop in condensing performance with a corresponding rise in head pressure and condensing temperature will reduce system capacity against the graph specification.

Suction Pressure

With a specified suction pressure of 1.86 kg/cm^{2}G *(182 kPaG - 27 psig)* the refrigerant vaporisation *(evaporating)* temperature is 0˚C. We can therefore deduct this compressor is matched to medium temperature application ie - R134a where an evaporating temperature of 0˚ is *"normal".*

As an alternative to specifying a suction pressure an evaporating temperature may be the published operating condition for the compressor.

> **ie - Instead of specifying *"suction Pressure 1.86 kg/cm^{2}G"***
> **The specification will be *"Evaporating temperature 0˚C"***

Special note – low temperature applications

For technicians selecting compressors for low temperature applications additional graphs are available on request that give performance data for specified sub zero applications *(usually in the form of a multiple plot graph)*. These are not openly distributed due to their complexity and special application. Additional information is available from Unicla upon special request.

Superheat and subcool specifications

Superheat and subcool specifications are detailed as a calibration parameter for the compressor since they both have an effect on the capacity of the system.

The graphed system capacity *(as shown)* is based on the system having a superheat of 10˚C and a subcool of 5˚C - an increase in either of these will have a direct effect on system capacity.

For compressor selection purposes, these readings are indicative of a *"normal system"* and are of minor significance in compressor selection unless either condition is significantly different than specified.

Reading the graphs/data

Performance/capacity graph interpretation:

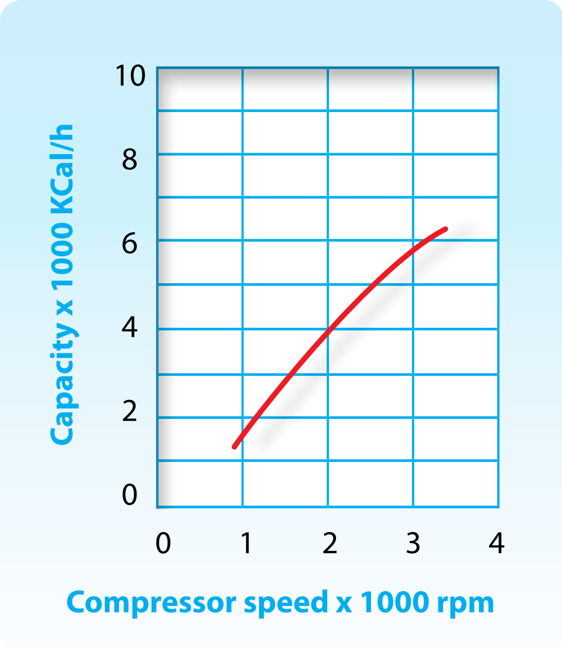
The performance/capacity graph principally indicates the level of performance from the compressor corresponding to its speed.

To ensure the reliability and validity of this data the operating conditions under which it is operating must be identified and related to the service environment into which it is being placed.

These operating conditions are the previously stated head pressure *(the pressure the compressor is working against),* the suction pressure/evaporating temperature, superheat and subcooling levels and most importantly the refrigerant in the system.

In principle, reading the graph is simple. The size or capacity of the evaporator or TX valve size can be directly related to the graph to correlate the speed at which evaporator/system capacity will be achieved. *(ie - when the compressor is providing adequate flow through the system).*

Graph 4. Compressor performance 1.



Operating conditions:

P. Suction	1.86 kg/cm^{2}G
P. Discharge	15.5 kg/cm^{2}G
Super heat	10.0 ˚C
Sub cool	5.0 ˚C
Evap. temp.	0.0 ˚C

Specifications:

Model	sample
Displacement	115 cc/rev
Cylinders	10
Max speed	7000 rpm

From this graph it can be observed that a system fitted with a 5.3 kW evaporator coil *(= 4550 kcal/hr = 1.5 ton refrigerating effect)* will not provide adequate capacity until a compressor speed of approximately 2500 rpm.

A 6.8 kW evaporator coil *(5,840 kcal/hr = 1.93 ton refrigerating effect)* will require a compressor speed of approximately 3100 rpm.

In essence the performance graph is simply indicating the speed at which the system capacity matches the compressor capacity, which in turn is compared to the operating conditions under which the system normally operates.

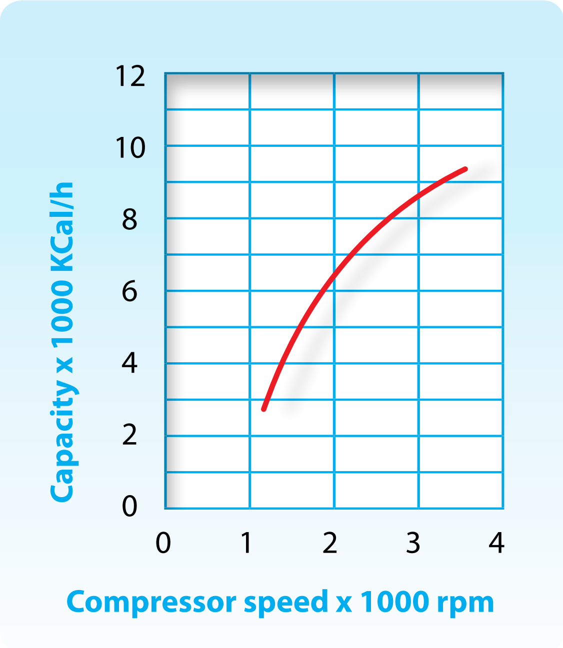
Let's use a selection example using the graph above *(Graph 4)* and compare it to Graph 5 *(over)*. A 6.8 kW (5840 kcal/hr) system operating at 2500 rpm governed speed and a drive ratio of 1:1 will produce an effective performance of 5.2 kW *(4500 kcal/hr.)*

If the system has been designed on a heat load calculation basis then performance levels will be substandard under this operating condition, ie- it does not achieve the 6.8 kW 5840 kcal/hr at its rated speed.

The next compressor *(Graph 5)* has a performance graph which indicates the 6.8 kW (5840 kcal/hr) capacity is achieved at approximately 1700 rpm.

For this operating condition this is the compressor of choice.

Graph 5. Compressor performance 2



Operating conditions:

P. Suction	1.86 kg/cm^{2}G
P. Discharge	15.5 kg/cm^{2}G
Super heat	10.0 ˚C
Sub cool	5.0 ˚C
Evap. temp.	0.0 ˚C

Specifications:

Model	sample
Displacement	200 cc/rev
Cylinders	10
Max speed	4500 rpm

Note: All or selected items may be published dependent on the compressor and its application.

Refrigerant flow characteristics

Refrigerant flow characteristics are not normally published because the principle factor in selection of the compressor is the match between capacity and speed *(as previously stated - read directly off the capacity graph).*

However:

To fully appreciate the necessity to match compressors to systems and their operating conditions we will present test data that relates to refrigerant flows against compressor speed. When an evaporator is rated there are a number of operating conditions attached *(see specifications over)* but the most critical in its relationship to the compressor is the flow rating of the evaporator.

Evaporator specifications - sample only

Coil specifications

Coil type	Sample only
Tub type	Rifled bore
Fin height	150mm
Fin length	495 mm
No. of rows	5.0
Fins/metre	510
No. of circuits	3.0
Fin material	Aluminium
Evaporating temp	0˚C
Superheat	5˚C
Airflow 1/5	187
Liquid temp	48˚C
Refrigerant	R134a

Ratings

Capacity	6.8 kW 5840 kcal/hr
Refrigerant flow	**3.2 kg/min**
Refrigerant charge	.1 kg
Pressure drop oil	67.2 kPa
Pressure drop header	14.0 kPa
Air pressure drop	190 Pa
Leaving vapor velocity	16.5 m/s

Flow Rating - the critical factor

If we now relate this to the flow ratings of the compressor (under the previously mentioned head pressure, suction pressure, superheat, subcool conditions) we can draw some conclusions.

Table 4. Compressor flow versus capacity - *sample only*

Compressor rpm	Refrigerant flow	Cooling capacity
1000	1.66 Kg/min	2.9 kW (2490 kcal/hr)
1500	2.71 Kg/min	5.8 kW (5000 kcal/hr)
2000	3.51 Kg/min	7.7 kW (6650 kcal/hr)
2500	4.13 Kg/min	9.6 kW (8300 kcal/hr)
3000	4.35 Kg/min	10.5 kW (9000 kcal/hr)

From Table 4 it can be clearly seen the flow rating at 1000 and 1500 rpm is below the specified rating of the evaporator – we need 3.2kg of refrigerant flow. We only have 1.66 kg/m at 1000 and 2.71 kg/m at 1500 rpm.

Remember:

It is the refrigerant flowing through the evaporator that absorbs heat from the cabin. At lower flow rates evaporator and system capacities are dramatically reduced – **There is simply not enough refrigerant flow at 1000 and 1500 rpm.** This is the principle reason for performance losses at idle.

At approximately 1800 rpm the refrigerant flow is at 3.2 kg/min. This matches the flow rating of 3.2 kg/min specified in the evaporator coil design to attain an efficiency of the specified 6.8 kW *(5840 kcal/hr)*. ***Above this speed evaporator flows are assured therefore system performance is optimised.***

Inadequate refrigerant flow as a result of the fitment of an undersize compressor will limit performance at all speeds where the refrigerant flow through the evaporator is below the specified value.

In reality flow ratings are normally not referred to in compressor selection, it is simply a match between the capacity of the compressor and the system – it is however important to realize **the controlling factor in capacity attainment is the flow.**

Additional methods of compressor matching

In the absence of technical data, compressor selection is often done via *"field service"* methods.

Method 1 - TX valve size

With a known TX valve or orifice size the effective maximum capacity of the evaporator can be presumed.

Example:

The previous evaporator with a 6.8 kW rating would normally have a 2 ton valve fitted to ensure adequate evaporator filling under maximum design heat load.

1 Ton = 3.517 kW 1.5 Ton = 5.275 kW 2 Ton = 7.034 kW

Therefore with reference to the performance chart *(table 4)* the 7.034 kW is obtained at just below 2000 r/min. Even though it is only a 6.8 kW coil in the absence of that specification the TX is the only form of capacity measurement we have - even though it is slightly higher than the **actual** capacity of the system it is a guide to compressor selection.

Method 2 - Low side pressure

In the absence of any data, low side pressures can be used as an **indicator** of compressor capacity and/or efficiency. The inability to *"pull down"* the low side to acceptable/normal levels may indicate a compressor with the inability to pull refrigerant through the evaporator at an acceptable rate comparative to the feed rate of the TX valve or Orifice tube. Whilst this is a common practice there are considerable dangers associated with it - as presented here.

Since the low side pressure is determined in principle by the TX valve feed rate compared to the evaporator pull rate *(suction)* it must be validated that the TX is the correct size, calibrated correctly, being *"fed"* with liquid at nominated pressures and charge rates correct.

In addition to the variable flow of the TX valve/Orifice tube, evaporator and suction line pressure drops can make this form of compressor capacity validation **virtually useless.**

A vehicle fitted with an evaporator or suction line exhibiting high pressure drop **can lead to the low side *"pulling"* extremely low in normal operation.** With an undersize compressor and this condition the low side pressures may appear *"normal"* - whereas in reality it is a system with high evaporator/suction line pressure drops coupled to an ineffective/undersize compressor.

If the compressor is the correct size the low side will pull extremely/abnormally low because of the pressure drop in this case. Pressure analysis should only be used by experienced technicians with substantial associated testing for TX flow rates, pressure/temperature drop analysis etc.

Dual evaporator systems

When fitting compressors to dual evaporator systems the total net capacity of both evaporators must be the basis for compressor selection.

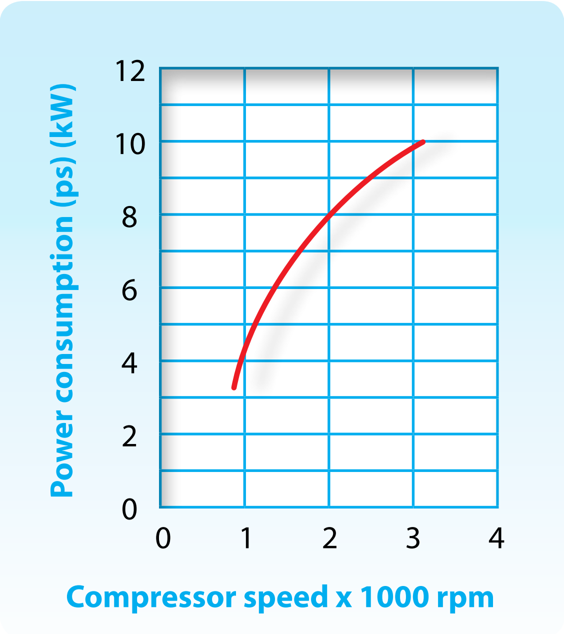
Example:

System is fitted with 2 x 4300 kcal/hr evaporators *(2 x 5kW) (2 x 1.5 ton TX valves)*

- **Total system capacity = 8600 kcal/hr 10kW)**
- **Operating conditions - city cycle vehicle**
- **Operating speed at 1500 - 2000 rpm average**

From the two graphs *(over - page 16)* compressor 2 is the compressor of choice. Fitting of compressor 1 will only provide for a total capacity of 6000 - 7600 kcal at the nominated speed of 1500 - 2000 rpm. Compressor 2 provides a capacity of 8000 - 9,600 kcal/hr - adequate for peak efficiency at the nominated compressor speed.

Graph 6. Compressor 1 performance



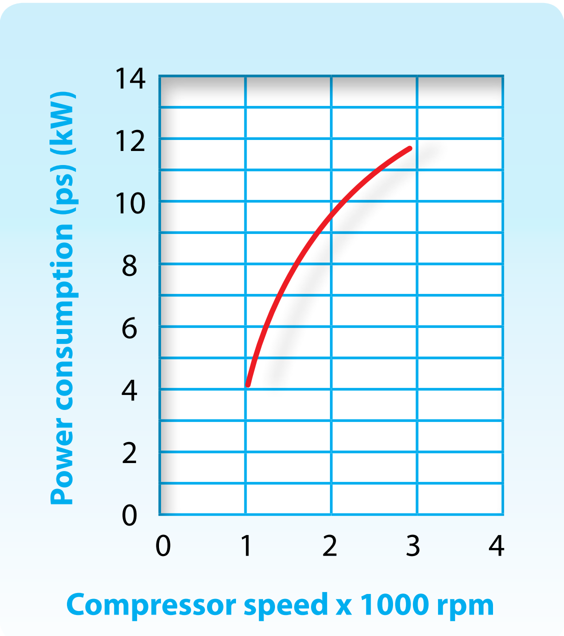
Operating conditions:

P. Suction	1.86 kg/cm²G
P. Discharge	15.5 kg/cm²G
Super heat	10.0 °C
Sub cool	5.0 °C
Evap. temp.	0.0 °C

Specifications:

Model	sample
Displacement	200 cc/rev
Cylinders	10
Max speed	4500 rpm

Graph 7. Compressor 2 performance



Operating conditions:

P. Suction	1.86 kg/cm²G
P. Discharge	15.5 kg/cm²G
Super heat	10.0 °C
Sub cool	5.0 °C
Evap. temp.	0.0 °C

Specifications:

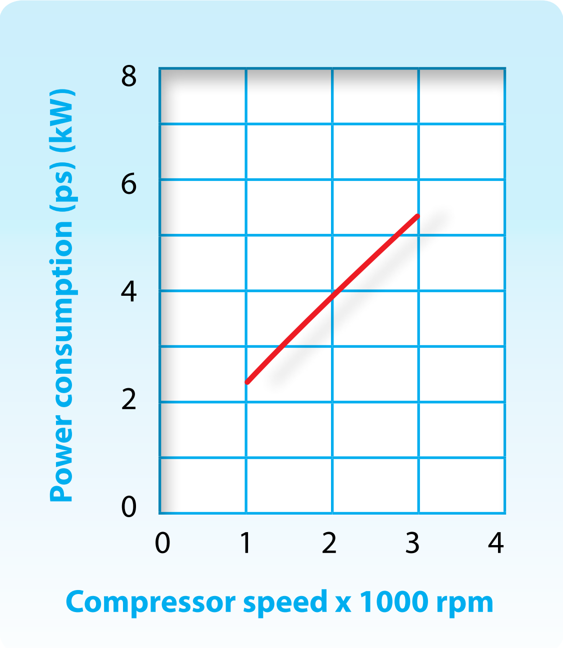
Model	sample
Displacement	200 cc/rev
Cylinders	10
Max speed	4500 rpm

Note: All or selected items may be published dependent on the compressor and its application.

Cautionary notes – dual evaporators

On dual evaporator systems with one circuit isolated *(normally done by a magnetic switching valve)*. The low side may pull abnormally low. Careful system design and compressor matching is required to ensure the low side does not pull into a vacuum in this position. A balance between efficiency and maintaining positive low side pressure is a requirement of compressor selection in many dual evaporator systems, ie - a large capacity compressor may achieve good efficiency but create a vacuum on the low side **- this is an undesirable condition.** An oil separator may be required in dual evaporator systems.

Graph 8. Compressor power consumption



Operating conditions:

P. Suction	1.86 kg/cm²G
P. Discharge	15.5 kg/cm²G
Super heat	10.0 ˚C
Sub cool	5.0 ˚C
Evap. temp.	0.0 ˚C

Specifications:

Model	sample
Displacement	200 cc/rev
Cylinders	10
Max speed	4500 rpm

Power consumption graphs

Power consumption graphs relate to the input required to drive the compressor across its speed range, measured in horsepower or kilowatts. The same operating conditions apply as for the previous capacity graph since the input requirements to drive the compressor will vary significantly depending on the amount of work it is doing. *(ie - the amount of head pressure it is working against, the volume of suction refrigerant etc).*

Operating conditions

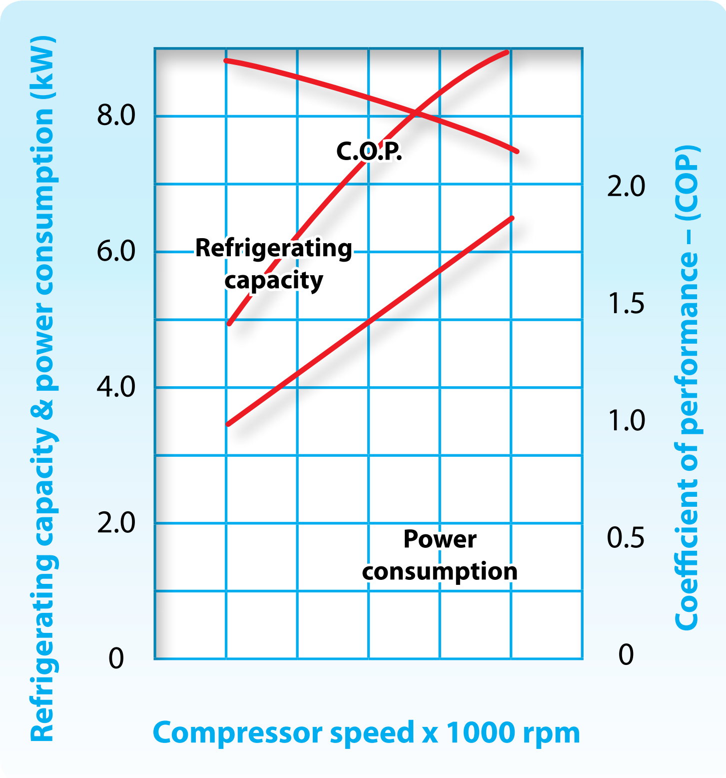
The power consumption graph is principally used in standby applications where an electric motor is being used to drive the pump at a nominated rpm. By referring to this graph a standby motor selection can be made and drive ratio's calculated.

Coefficient of performance - (COP) graphs

COP graphs are normally plotted on a power consumption and capacity graph.
COP is the ratio between the output *(measured in cooling capacity)* and the input required to drive the compressor *(power consumption)*. COP in simple terms is output compared to input.

If we input 2.1 kW and output 3.15 kW the COP is 1.5. This COP value will vary with compressor speed since the dynamics of the system will change as flow rates, internal resistances, heat absorption and rejection rates etc vary. A principle factor in COP is the refrigerant being used and the conditions under which the test is being conducted. All testing must be against a standard as previously identified.

Compressor capacity & horsepower graph



Summary - compressor selection

The principle selection criteria is **the capacity of the compressor must match the capacity of the evaporator(s)** *(system)* under the nominated operating conditions *(compressor speed)*

Insufficient flow rates at lower compressor speeds will result in dramatically reduced capacity of the system, and result is possible risk of damage to the compressor. *(See separate booklet - Unicla Service Information)*

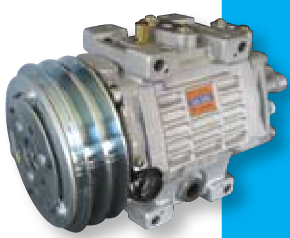
In the absence of system capacity/evaporator capacity data the TX valve size may be used as an indicator of capacity. The TX should be sized to adequately fill the coil under full heat loads.

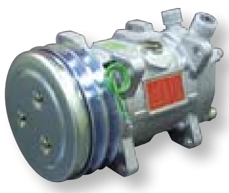
Low side pressure analysis should not be used in isolation as a method of determining compressor size/capacity due to the variables of pressure drop/TX flow rates etc.

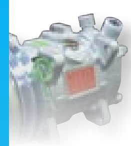
Dual evaporator systems will require a compressor with adequate capacity to cater for both evaporators at nominated compressor speeds.

Power consumption graphs are used to ascertain drive motor capacity/input power requirements.

Coefficient of performance (COP) graphs show the output compared to the input across the operating range of the compressor for a nominated refrigerant.







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