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# System Balancing For Zeotropic Refrigerants

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## ABSTRACT

Current methods for refrigeration component rating, data presentation, selection and balancing were originally developed for systems using pure refrigerants. Due to recent changes in refrigerant usage, a large number of commercialised alternative refrigerants are mixtures of various fluids, and many of these mixtures produce temperature glide during phase-change. This fundamental difference impacts on the operating behaviour of refrigeration components and as such annuls many of the current methods. A new balancing method has been developed previously, that enables components using a refrigerant with temperature glide to be matched correctly. In addition, the influence of superheat, desuperheat and pressure drop is incorporated into the new method. The object of this paper is to present a worked example using this new balancing method.

## NOMENCLATURE

**Notation**

| | |
|---|---|
| $c$ | a constant |
| $Q$ | rate of heat flow, capacity (W) |
| $T$ | temperature (°C) |
| $\phi$ | denotes 'function of' |

**Subscripts**

| | |
|---|---|
| $c$ | condenser |
| $c/u$ | condensing unit |
| $dp$ | dew-point condition |
| $e$ | evaporator |
| $e/u$ | evaporating unit |
| $HTF$ | heat transfer fluid |
| $(in)$ | inlet condition |
| $(out)$ | outlet condition |

## INTRODUCTION

For successful operation of a refrigeration system, a series of technical evaluations related to individual components and the system as a whole are necessary. The final stage of these evaluations is system balancing. The purpose of system balancing is to enable the designer of a refrigeration system to accurately identify the system operating conditions. It is necessary to select those components that closest meet the system needs and to then determine the equilibrium points based on the performance characteristics of the selected components. Colbourne and Suen (2001) provides the logic and methodology for a new component rating and system balancing technique. This new technique allows the use of both pure and mixture refrigerants to be used, and it accounts for pressure loss and varying compressor discharge superheat, referred to as the "advanced cycle" in this paper. In order for the balancing process to be adopted for the advanced cycle with zeotropic refrigerant, a revised procedure is employed to account for the following factors which are required as part of the new rating method:

- All reference temperatures must be based on dew-point temperatures of the refrigerant relative to condenser inlet or evaporator outlet pressure (i.e. $T_{c,dp(in)}$, $T_{e,dp(out)}$).
- Compressor ( $\Delta T_{dsh,comp}$ ) and condenser desuperheat ( $\Delta T_{dsh}$ ) is accounted for in condenser characteristics.
- Condenser pressure loss is considered i.e. $T_{c,dp(in)} \neq T_{c,dp(out)}$ .

- Condenser outlet/expansion valve inlet temperature ($T_{TEV(in)}$) is included in evaporator characterisation (assumed constant subcooling).
- Circulating composition of the mixture is assumed constant throughout the entire system where a zeotrope is used.[1]

A practical example of the new balancing process is given in the following sections.[2] This will be based on the data provided in the Appendix, which was generated using a set of component performance simulation models. The authors hope that with the new rating method, all the necessary data will be eventually made available by the manufacturers in the new product catalogues.

## BALANCING CONDENSER AND COMPRESSOR

The first stage, as with the conventional method, is to balance the compressor and condenser to form the condensing unit. For the advanced cycle, however, it is not possible for the rated compressor data to be matched directly to the condenser data because some uncommon variables are used in establishing their characteristics as shown in the catalogue data for the compressor (Table A2) and condenser (Table A5) capacities. It is essential for both sets of data to use identical variables if matching is to take place. The solution is to utilise the compressor desuperheat data (Table A4) to convert condenser capacity data rated for a specific $\Delta T_{dsh}$ to condenser capacities that correspond to an equivalent $T_{e,dp(out)}$, via the compressor superheat data in Table A4. In other words, for a fixed set of compressor $T_{e,dp(out)}$ and $T_{c,dp(in)}$, desuperheat can be measured, and the corresponding $Q_c$ is obtained for each of the values of $T_{e,dp(out)}$ and $T_{c,dp(in)}$. Subsequently a new condenser characteristic can be produced which is directly related to the operating temperatures for the selected compressor. By examining the data in Table A5 and the condenser characteristic in Fig. 1, it is seen that the condenser capacity becomes a function of $T_{e,dp(out)}$ (see function 1).

$$Q_c = \phi\left(T_{c,dp(in)}, T_{e,dp(out)}\right) \; ; \; T_{HTF(in)} = \mathrm{c} \tag{1}$$

This new characteristic is obtained, as demonstrated, from: (i) select an evaporating temperature, e.g. $T_{e,dp(out)}$ = -30°C from Table A4; (ii) select a condensing temperature, e.g. $T_{c,dp(in)}$ = +30°C; (iii) from Table A4 obtain the corresponding desuperheat, i.e. $\Delta T_{dsh,comp}$ = 69 K; (iv) in Table A5, interpolate between the $\Delta T_{dsh}$ values at the selected $T_{c,dp(in)}$ to find the corresponding capacity, i.e. $Q_c$ = 1.471 kW; (v) proceed for the series of $T_{e,dp(out)}$ for the range of $T_{c,dp(in)}$ so that the interpolated $Q_c$ data can be plotted as seen in Fig. 1. After this conversion, both the condenser characteristic (function 1), and the compressor characteristic (Table A2) use identical variables (i.e. $Q_c$ or $Q_{c,comp}$, $T_{c,dp(in)}$ and $T_{e,dp(out)}$). Now the new condenser data can be combined with the compressor data, where the balance points converge on

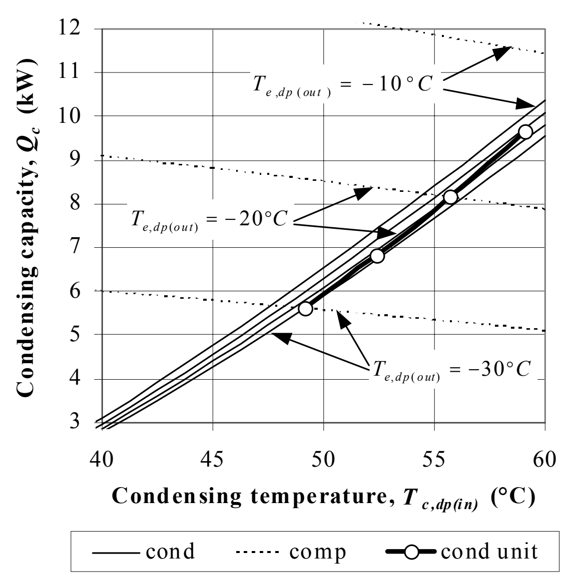


**Fig. 1: Balancing of condenser and compressor**

[1] Changes in circulating composition is a result of many factors and cannot be determined for a single component.

[2] Note that all descriptions are based on fixed HTF inlet temperatures for the evaporator and condenser. Often the engineer is interested in the effect of different HTF inlet temperatures, so the exercises should be repeated for different $T_{HTF(in)}$.

the same evaporating temperature of each component. This exercise with the resulting condensing unit characteristic is shown in Fig. 1. Function (2) is an expression for this condensing unit characteristic where $T_{e,dp(out)}$ is implicit.

$$Q_{c,c/u} = \phi\left(T_{c,dp(in)}\right)_{T_{e,dp(out)}} \; ; \; T_{HTF(in)} = \text{c} \tag{2}$$

## BALANCING EVAPORATOR AND COMPRESSOR

In the conventional balancing method, the evaporator would be balanced with the condensing unit at this point. However, from the evaporator rating (Table A7) it is known that $Q_e$ is now a function of an additional variable, $T_{c,dp(out)}$. Since $T_{c,dp(out)}$ is not a variable in the condensing unit characteristic (function 2), $Q_e$ cannot be matched directly to it. To solve this, $T_{c,dp(in)}$ must be integrated into the evaporator characteristic. This is achieved with the development of an additional sub-system comprising the compressor and evaporator, termed the "evaporating unit".[3] Since the compressor evaporating capacity characteristic (Table A1) and the evaporator capacity characteristic (Table A7) only share one variable ($T_{e,dp(out)}$), the evaporator characteristic must be adjusted so that $T_{c,dp(out)}$ matches the other variable of the compressor characteristic ($T_{c,dp(in)}$).

From Table A7 it is known that $Q_e$ is a function of both $T_{c,dp(out)}$ and $T_{e,dp(out)}$. Note that the evaporator rating refers to the outlet condition of the condenser (i.e. corresponding $T_{TEV(in)}$) and not $T_{c,dp(in)}$ since the condenser pressure drop is unknown at the time of evaporator rating (i.e. $T_{c,dp(out)} \neq T_{c,dp(in)}$). Thus, Table A7 cannot be used directly for combining evaporator with the compressor. In other words, the compressor rating uses $T_{c,dp(in)}$, whereas the evaporator rating does not recognise the pressure loss present in the condenser. In order to account for this disagreement, the evaporator must be re-rated with respect to the condenser pressure drop. (A greater pressure drop in the condenser changes $T_{TEV(in)}$, thus the inlet quality, and the evaporator inlet temperature since the temperature glide is intersected and therefore changes $Q_e$.)

In its present format, the condenser pressure loss data (Table A6) cannot be used to re-rate the evaporator data, since the condenser data is based on $\Delta T_{dsh}$ (which is not a variable in the evaporator characteristic). Also, by comparing Tables A5 and A7, it is seen that $T_{e,dp(out)}$ is absent from Table A5, indicating that condenser $\Delta P$ characteristic (Table A6) needs to be converted, so that $T_{c,dp(out)}$ becomes a function of $T_{e,dp(out)}$ rather than $\Delta T_{dsh}$. As with the earlier discussion on conversion of condenser capacity with compressor data, the same logic is applied here. It is known that for a specific set of operating temperatures, the compressor will provide a specific $\Delta T_{dsh}$ (Table A4). Therefore, the corresponding evaporating temperature can be determined from the compressor $\Delta T_{dsh}$ data at a specified $T_{c,dp(in)}$.

**Table 1: Converted condenser outlet temperature ($T_{c,dp(out)}$ /°C) characteristic**

| $T_{c,dp(in)}$ | Evaporating temp, $T_{e,dp(out)}$ | | | |
|---|---|---|---|---|
| | -30°C | -20°C | -10°C | 0°C |
| 30°C | 30.0 | 30.0 | 30.0 | 30.0 |
| 40°C | 39.1 | 39.0 | 39.0 | 38.9 |
| 50°C | 47.4 | 47.0 | 46.8 | 46.5 |
| 60°C | 55.0 | 54.2 | 53.6 | 53.0 |

Table A6 illustrates original condenser pressure loss characteristic, and Table 1, the converted characteristic. This converted characteristic is obtained from: (i) select an evaporating temperature, e.g. $T_{e,dp(out)}$ = -30°C from Table A4; (ii) select a condensing temperature, e.g. $T_{c,dp(in)}$ = +60°C; (iii) from Table A4 obtain the corresponding

[3] The compressor data has been used twice, once to produce condensing unit data, and now for evaporating unit data. It should be borne in mind that the use of the compressor characteristic is simply employed to fix a relationship between capacities of the various system components.

desuperheat, i.e. $\Delta T_{dsh,comp}$ = 68 K; (iv) in Table A6, interpolate between the $\Delta T_{dsh}$ values at the selected $T_{c,dp(in)}$ to find the corresponding $T_{c,dp(out)}$, i.e. 54.9°C; (v) proceed for the series of $T_{e,dp(out)}$ for the range of $T_{c,dp(in)}$ so that the interpolated $T_{c,dp(out)}$ data can be tabulated in Table 1. The new condenser pressure drop characteristic is represented in function (3).

$$T_{c,dp(out)} = \phi\left(T_{e,dp(out)}, T_{c,dp(in)}\right); T_{HTF(in)} = \mathrm{c} \tag{3}$$

With $T_{e,dp(out)}$ as part of the condenser pressure drop characteristic (function 3), the existing evaporator characteristic (Table A7) can be modified so that the capacity is represented as a function of $T_{c,dp(in)}$. The conversion of the evaporator data to account for condenser pressure drop is achieved as follows: (i) select an evaporating temperature, e.g. $T_{e,dp(out)}$ = -30°C; (ii) select an inlet condensing temperature from Table 1, e.g. $T_{c,dp(in)}$ = +60°C; (iii) from Table 1 obtain the corresponding outlet condensing temperature, i.e. $T_{c,dp(out)}$ = 55°C; (iv) in Table A7, interpolate between the $T_{c,dp(out)}$ values at the selected $T_{e,dp(out)}$ to find the corresponding capacity, i.e. $Q_e$ = 5.12 kW; (v) proceed for the series of $T_{e,dp(out)}$ for the range of $T_{c,dp(in)}$ so that the interpolated $Q_e$ data can be plotted as seen in Fig. 2. Thus, the evaporator characteristic becomes function (4). The evaporator characteristic is now specific to both the selected condenser and compressor. Note that if condenser pressure drop is negligible the transformation of evaporator data in Table A7 to function (4) is not necessary, since $T_{c,dp(in)} = T_{c,dp(out)}$.

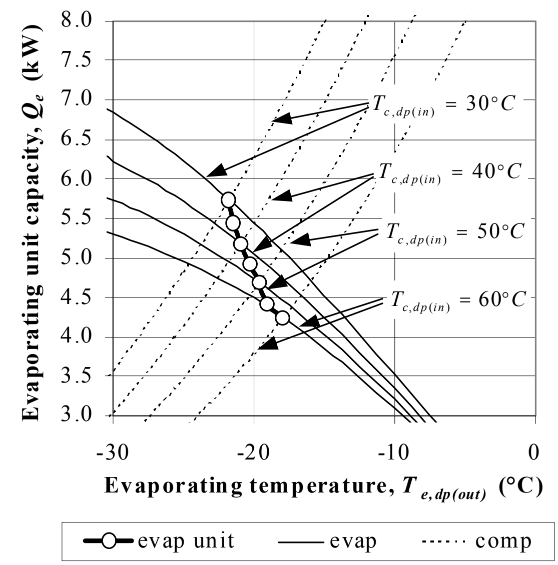

**Fig 2: Balancing of evaporator and compressor**

$$Q_e = \phi\left(T_{c,dp(in)}, T_{e,dp(out)}\right)\ ;\ T_{HTF(in)} = \mathrm{c} \tag{4}$$

Now that the evaporator characteristic is a function of the same reference temperatures as the compressor characteristic, it is simple to combine the two to produce an evaporating unit as shown in Fig. 2. This evaporating unit characteristic is provided in function (5), where the implicit condensing temperature is denoted by the subscript $T_{c,dp(in)}$.

$$Q_{e,e/u} = \phi\left(T_{e,dp(out)}\right)_{T_{c,dp(in)}}\ ;\ T_{HTF(in)} = \mathrm{c} \tag{5}$$

## BALANCING CONDENSING UNIT AND EVAPORATING UNIT

The final stage is to combine the condensing unit and evaporating unit to obtain the overall system balance points. Condensing unit capacity data as a function of $T_{c,dp(in)}$ (function 2) can be transposed in the normal way - i.e. using the compressor characteristics in Tables A1 and A2 - to provide the equivalent evaporating capacity of the condensing unit as a function of $T_{e,dp(out)}$ (function 6). In this example $Q_e$ = 4.33 kW at $T_{e,dp(out)}$ = -18.9°C.

$$Q_{e,c/u} = \phi\left(T_{e,dp(out)}\right)_{T_{c,dp(in)}}\ ;\ T_{HTF(in)} = \mathrm{c} \tag{6}$$

The evaporating unit (function 5) and the condensing unit (function 6) are now balanced to provide the overall system equilibrium $T_{e,dp(out)}$ and $Q_e$. A graphical interpretation of this is illustrated in Fig. 3. Similarly, the evaporating unit capacity data in function (5) can be transposed in the normal way - i.e. using the compressor characteristics in Tables A1 and A2 - so that the condensing capacity of the evaporating unit is obtained (function 7). This is then balanced with the condensing unit condensing capacity (function 2) to provide the overall system equilibrium $T_{c,dp(in)}$ and $Q_c$ (Fig. 4). In this example $Q_c$ = 8.47 kW at $T_{c,dp(in)}$ = 56.6°C.

$$Q_{c,e/u} = \phi\left(T_{c,dp(in)}\right)_{T_{e,dp(out)}} \;;\; T_{HTF(in)} = \mathrm{c} \tag{7}$$

With the balance points known, other corresponding system parameters such as $\Delta T_{dsh}$ (and therefore discharge temperature), $W_{comp}$ and $COP$ can be obtained. Finally, the evaporator pressure drop data (Table A8) is employed to identify the inlet conditions of the evaporator for expansion valve selection purposes.

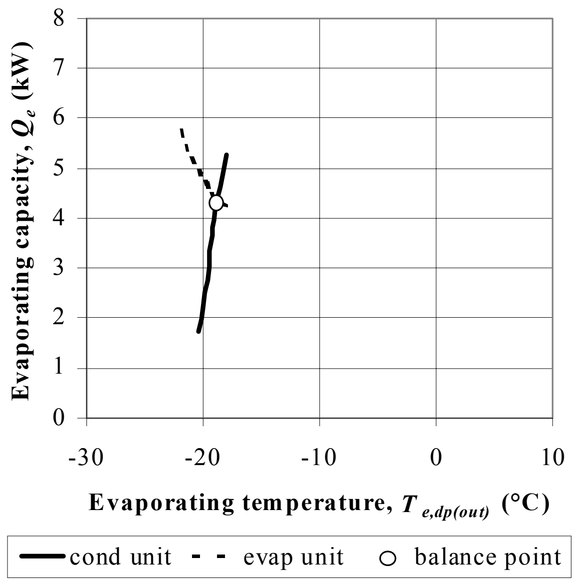


**Fig. 3: Evaporating balance points**

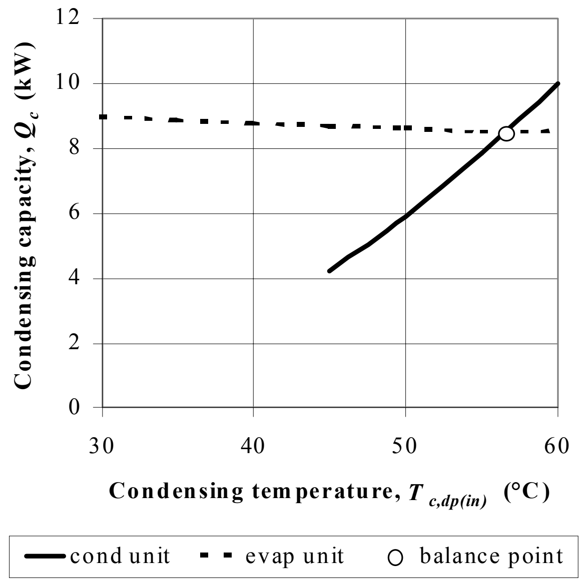


**Fig 4: Condensing balance points**

## DISCUSSION

The example detailed here is based on a new standardised procedure for rating and balancing system components for an advanced cycle, as described in Colbourne and Suen (2001). Although it is primarily developed for zeotropic refrigerants, it can be used for pure refrigerants since the reference temperatures are applicable regardless of whether the refrigerant possesses a temperature glide.

A number of additional examples were conducted primarily to validate the methodology, but also to compare its accuracy against the conventional methods that substitute reference temperatures for dew- or mid-point temperatures. A convenient approach to this is with component simulation, as it enables fictitious component data to be produced, and then ultimately checked against the balance points. Capacity data obtained from the new balancing technique should be equal to that obtained from the direct simulation at balance temperatures if the described balancing method is valid. As a comparison, both dew-point conditions and mid-point conditions as a basis for component rating with zeotropes were used in the checking method. These reference points are mentioned in other literature (e.g. Hundy and Vittal, 2000 and Murphy *et al*, 1998).

All the examples were based on fictitious component performance data generated for standard catalogue conditions using a refrigerant mixture of 50% R290 and 50% R600a by mass. The generated component data was according to a range of parameters. Various combinations of evaporator and condenser pressure loss and fictitious temperature glide were examined, where Cond 1/Evap 1 has $\Delta T_G$ = 10K, high $\Delta P$; Cond 2/Evap 2 has $\Delta T_G$ = 10K, low $\Delta P$; Cond 3/Evap 3 has $\Delta T_G$ = 5K, high $\Delta P$; Cond 4/Evap 4 has $\Delta T_G$ = 5K, low $\Delta P$.

The results of the exercise (Table 2) indicate that the new balancing method will generally return the correct capacities to within 1.5% of the simulated values on the evaporating side, and within 1% on the condensing side. Although the technique is deemed numerically precise, small errors are to be expected because of the successive linear interpolation between data points instead of curve-fitting the data. By comparison with the conventional method using either the mid-point or the dew-point as the reference evaporating and condensing temperatures, a number of general observations can be made:

- In most cases errors were larger when the refrigerant possessed a larger temperature glide.
- The conventional method with mid-point and dew-point over-predicts capacity in both the evaporator and condenser, with the larger error generally seen with the evaporator and also giving a higher evaporating temperature.
- High evaporator and condenser $\Delta P$ scenarios result in significantly greater error than with the low $\Delta P$ scenarios.
- Capacity and temperature errors were greater when the approach temperature differences were relatively large.
- In none of the examples did using the mid-point or dew-point temperature provide a better degree of accuracy than that of the new method. Neither can a preference be made between these temperatures as the greatest error of the two varies from case to case.

Errors in capacity generated in this example range from about 3% to 20% as compared to the simulation. It should be noted that errors of any other systems would vary depending upon the equipment used and the operating conditions. Over-prediction of capacity that was observed in all cases is consistent with the fact that the conventional balancing method does not account for the additional losses associated with the components operating under non-standard conditions (e.g. change in $\Delta T_G$, reduction in capacity with $\Delta P$ and variable $\Delta T_{dsh}$). It is clear that an improvement in accuracy will always be achieved when using this new method.

**Table 2: Comparison of balance-points using different methods**

| Combination | Method | Evaporating unit | | | Condensing unit | | | |
|---|---|---|---|---|---|---|---|---|
| | | $T_{e,dp}$ (°C) | $Q_e$ (kW) | Error (%) | $T_{c,dp}$ (°C) | $\Delta T_{dsh}$ (K) | $Q_c$ (kW) | Error (%) |
| **Evap 1 Cond 1** | model | -18.93 | 4.15 | - | 56.58 | 46.1K | 8.49 | - |
| | new | -18.93 | 4.33 | 4.3 | 56.58 | - | 8.47 | -0.2 |
| | mid-pt | -16.05 | 4.92 | 18.6 | 57.87 | - | 9.38 | 10.5 |
| | dew-pt | -16.63 | 4.81 | 15.9 | 57.43 | - | 9.21 | 8.5 |
| **Evap 2 Cond 2** | model | -21.62 | 3.94 | - | 53.49 | 50.3K | 7.76 | - |
| | new | -21.62 | 3.97 | 0.8 | 53.49 | - | 7.80 | 0.5 |
| | mid-pt | -19.66 | 4.41 | 11.9 | 53.84 | - | 8.44 | 8.8 |
| | dew-pt | -20.18 | 4.31 | 9.4 | 53.48 | - | 8.29 | 6.8 |
| **Evap 3 Cond 3** | model | -18.91 | 4.34 | - | 55.73 | 46.0K | 8.53 | - |
| | new | -18.91 | 4.40 | 1.4 | 55.73 | - | 8.53 | 0.0 |
| | mid-pt | -15.03 | 5.19 | 19.6 | 57.86 | - | 9.75 | 14.3 |
| | dew-pt | -16.13 | 4.99 | 15.0 | 57.03 | - | 9.42 | 10.4 |
| **Evap 4 Cond 4** | model | -21.18 | 4.07 | | 52.90 | 49.5K | 7.93 | |
| | new | -21.18 | 4.11 | 1.0 | 52.90 | - | 7.97 | 0.5 |
| | mid-pt | -20.29 | 4.37 | 7.4 | 52.53 | - | 8.32 | 4.9 |
| | dew-pt | -19.61 | 4.49 | 10.3 | 53.02 | - | 8.51 | 7.3 |

## FINAL REMARKS

The new method detailed in this paper is more complex than the conventional method described in textbooks and requires a number of additional stages. This is a consequence of improving accuracy of system design. However, the implementation of this new method can be simplified in certain situations:

- Where there is a lack of appropriate data, for example only a single evaporator characteristic instead of a series of characteristics for different increments of $T_{c,dp(out)}$, the relevant stage in the balancing process may be neglected. For example, where a refrigerant with $d\Delta T_G / dx_{e(in)} \approx 2.5$ is used, one single evaporator characteristic line may be present and there is no need to create an evaporating unit characteristic.
- If the pressure drop in the condenser is small, certain steps can be eliminated as detailed earlier in the paper.
- The method can be automated through the use of personal computers, for example by adapting the approaches presented by Page (1989), Stoecker and Jones (1982) and Trott (1981).
- Notation can be simplified by denoting $T_{c,dp(in)}$ as $T_c$, $T_{c,dp(out)}$ as $T_c'$, $T_{e,dp(out)}$ as $T_e$ and $T_{e,dp(in)}$ as $T_e'$ throughout the entire exercise with an understanding that they actually represent the dew-point temperatures.

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# APPENDIX: EXAMPLE CATALOGUE DATA

## Compressor data

Table A1: Evap. capacity, $Q_{e,comp}$ /kW

| $Q_{e,comp} = \phi\left(T_{e,dp(out)}, T_{c,dp(in)}\right)$ | | | | |
|---|---|---|---|---|
| $T_{c,dp(in)}$ | Evaporating temp, $T_{e,dp(out)}$ | | | |
| | -30°C | -20°C | -10°C | 0°C |
| 30°C | 3.58 | 6.28 | 10.00 | 15.12 |
| 40°C | 3.05 | 5.44 | 8.78 | 13.44 |
| 50°C | 2.53 | 4.60 | 7.56 | 11.71 |
| 60°C | 2.03 | 3.78 | 6.33 | 9.97 |

Table A2: Cond. capacity, $Q_{c,comp}$ /kW

| $Q_{c,comp} = \phi\left(T_{e,dp(out)}, T_{c,dp(in)}\right)$ | | | | |
|---|---|---|---|---|
| $T_{c,dp(in)}$ | Evaporating temp, $T_{e,dp(out)}$ | | | |
| | -30°C | -20°C | -10°C | 0°C |
| 30°C | 6.39 | 9.62 | 13.71 | 18.91 |
| 40°C | 6.01 | 9.10 | 13.05 | 18.07 |
| 50°C | 5.58 | 8.52 | 12.28 | 17.09 |
| 60°C | 5.11 | 7.88 | 11.43 | 15.98 |

Table A3: Power, $W_{comp}$ /kW

| $W_{comp} = \phi\left(T_{e,dp(out)}, T_{c,dp(in)}\right)$ | | | | |
|---|---|---|---|---|
| $T_{c,dp(in)}$ | Evaporating temp, $T_{e,dp(out)}$ | | | |
| | -30°C | -20°C | -10°C | 0°C |
| 30°C | 2.81 | 3.35 | 3.71 | 3.79 |
| 40°C | 2.96 | 3.67 | 4.26 | 4.63 |
| 50°C | 3.05 | 3.92 | 4.73 | 5.38 |
| 60°C | 3.08 | 4.09 | 5.11 | 6.02 |

Table A4: Discharge superheat, $\Delta T_{dsh,comp}$ /K

| $\Delta T_{dsh,comp} = \phi\left(T_{e,dp(out)}, T_{c,dp(in)}\right)$ | | | | |
|---|---|---|---|---|
| $T_{c,dp(in)}$ | Evaporating temp, $T_{e,dp(out)}$ | | | |
| | -30°C | -20°C | -10°C | 0°C |
| 30°C | 69 | 46 | 32 | 23 |
| 40°C | 70 | 47 | 34 | 25 |
| 50°C | 70 | 48 | 35 | 27 |
| 60°C | 68 | 47 | 36 | 28 |

## Condenser data

Table A5: Capacity, $Q_c$ /kW

| $Q_c = \phi\left(T_{c,dp(in)}, \Delta T_{dsh}\right); T_{HTF(in)} = c$ | | | | | |
|---|---|---|---|---|---|
| $T_{c,dp(in)}$ | Discharge superheat, $\Delta T_{dsh}$ | | | | |
| | 30 K | 50 K | 70 K | 90 K | 110 K |
| 30°C | 1.57 | 1.49 | 1.47 | 1.46 | 1.47 |
| 40°C | 4.61 | 4.36 | 4.26 | 4.22 | 4.22 |
| 50°C | 8.25 | 7.84 | 7.65 | 7.58 | 7.56 |
| 60°C | 11.63 | 11.06 | 10.80 | 10.70 | 10.67 |

Table A6: Outlet temperature, $T_{c,dp(out)}$ /°C

| $T_{c,dp(out)} = \phi\left(T_{c,dp(in)}, \Delta T_{dsh}\right); T_{HTF(in)} = c$ | | | | | |
|---|---|---|---|---|---|
| $T_{c,dp(in)}$ | Discharge superheat, $\Delta T_{dsh}$ | | | | |
| | 30 K | 50 K | 70 K | 90 K | 110 K |
| 30°C | 30.0 | 30.0 | 30.0 | 30.0 | 30.0 |
| 40°C | 38.9 | 39.0 | 39.1 | 39.2 | 39.2 |
| 50°C | 46.6 | 47.1 | 47.4 | 47.6 | 47.8 |
| 60°C | 53.3 | 54.3 | 55.0 | 55.6 | 56.0 |

## Evaporator data

Table A7: Capacity, $Q_e$ /kW

| $Q_e = \phi\left(T_{e,dp(out)}, T_{c,dp(out)}\right); T_{HTF(in)} = c$ | | | | |
|---|---|---|---|---|
| $T_{e,dp(out)}$ | Cond. outlet temperature, $T_{c,dp(out)}$ | | | |
| | 30°C | 40°C | 50°C | 60°C |
| 0°C | 1.50 | 1.44 | 1.40 | 1.38 |
| -10°C | 3.53 | 3.32 | 3.13 | 2.92 |
| -20°C | 5.45 | 5.00 | 4.57 | 4.10 |
| -30°C | 6.84 | 6.14 | 5.47 | 4.77 |

Table A8: Inlet temperature, $T_{e,dp(in)}$ /°C

| $T_{e,dp(in)} = \phi\left(T_{e,dp(out)}, T_{c,dp(out)}\right); T_{HTF(in)} = c$ | | | | |
|---|---|---|---|---|
| $T_{e,dp(out)}$ | Cond. outlet temperature, $T_{c,dp(out)}$ | | | |
| | 30°C | 40°C | 50°C | 60°C |
| 0°C | 0.1 | 0.1 | 0.1 | 0.2 |
| -10°C | -9.4 | -9.2 | -8.8 | -8.7 |
| -20°C | -17.4 | -16.6 | -15.5 | -14.7 |
| -30°C | -22.6 | -21.0 | -18.9 | -17.6 |