REFRIGERATOR AND HEAT PUMP EQUIPMENT:
CHECK-UPS AND PERFORMANCE DATA INFERRED FROM MEASUREMENTS UNDER FIELD CONDITIONS IN THE REFRIGERANT SYSTEM

1 SCOPE AND FIELD OF APPLICATION

1.1 Scope
The scope of this Nordtest method is to provide instructions and recommendations regarding onsite performance check-ups of heat pumps and refrigeration equipment. The method describes the way in which the performance of heat pump and refrigeration equipment is to be inferred by means of measurements in the refrigerant system and the performance compared with data provided by the manufacturer or supplier of the equipment. The objective of the method is to keep the uncertainty of measurement of the coefficient of performance below ±15 %.

1.2 Field of application
The primary field of application of this Nordtest method pertains to electrically driven compression heat pumps and refrigeration equipment. The equipment is assumed to use the basic reversed-rankine cycle with either a single component refrigerant or an azeotropic blend. However, relevant parts may also be applied to the testing of equipment using the same type of refrigeration cycle with other types of prime movers.

The method described applies to installations which are equipped with suitably placed pressure taps for the condensing and evaporating pressures and a known heat loss from the compressor. The method is particularly suited to small unitary equipment with reasonably well insulated hermetic compressors.

2 REFERENCES
The following Nordtest method is a necessary, controlling document when NT VVS 116 (this method) is to be used:

2.1 NT VVS 115. Refrigeration and heat pump equipment
- General conditions for field testing and presentation of performance.
NT VVS 115 controls the following specific test methods:

2.1.1 NT VVS 116. Refrigeration and heat pump equipment
- Check-ups and performance data inferred from measurements in the refrigerant system (this method).

2.1.2 NT VVS 076. Large heat pumps
- Field testing presentation of performance.
Reference 2.1 also lists the relevant European standards and, in an appendix, some other useful documents.
3 DEFINITIONS

Reference 2.1 defines the technical terms that are essential for the application of this method. It also provides further references to other definitions which are specific to heat pumps and refrigerating equipment and may be of use within the field of application of this method.

3.1 System boundaries

Figure 1 summarises the system boundaries for heat pump/refrigerating equipment, heat pump/refrigerating plants and heating plants respectively, which are used in the definitions of coefficients of performance in Reference 2.1.

![Diagram of system boundaries](image)

Figure 1. System boundaries for determination of coefficients of performance.

3.2 Refrigerant states

Figure 2 summarises the symbols used to define and designate the refrigerant states which are necessary to calculate performance data in accordance with this Nordtest method.

![Diagram of refrigerant states](image)

Figure 2. Designation of refrigerant states.

4 SYMBOLS

References 2.1 provides the symbols that are essential for the application of this method. It also provides further references to other symbols which are specific to heat pumps and refrigerating equipment and may be of use within the field of application of this method.

4.1 General symbols

In this Nordtest method the index "kb" is consistently used for the heat transfer medium on the cold side, "vb" for the heat transfer medium on the hot side and "1" for air. In some cases "in" or "ut" are added to indicate inlet or outlet. The high pressure side (condenser) is designated by the index "1" and the low pressure side (evaporator) by the index "2".

4.2 Symbols for performance check-ups

The following symbols pertain specifically to performance check-ups or other situations where performance data are inferred from measurements in the refrigerant system. For further information, see Figures 1 and 2.

- **COP** Coefficient of performance (-)
- **COP₁** Coefficient of performance in the heating mode (inferred from refrigerant states)
- **COP₁kB** Compressor coefficient of performance in the heating mode
- **COP₁p** Motor coefficient of performance in the heating mode
- **COP₁pₐ** Total coefficient of performance in the heating mode
- **COP₂** Coefficient of performance in the cooling mode (inferred from refrigerant states)
- **COP₂kB** Compressor coefficient of performance in the cooling mode
- **COP₂p** Motor coefficient of performance in the cooling mode
- **COP₂pₐ** Total coefficient of performance in the cooling mode
- **h** Specific enthalpy (J/kg)
- **h₁** Specific enthalpy of the saturated refrigerant vapour in the condenser at temperature T₁ and pressure p₁
- **h₂** Specific enthalpy of the saturated refrigerant vapour in the evaporator at temperature T₂ and pressure p₂
- **h₃** Specific enthalpy of the superheated refrigerant vapour in the compressor outlet at temperature T₃ and pressure p₃ (p₃ ≥ or ~ p₁)
- **h₄** Specific enthalpy of the superheated refrigerant vapour in the compressor inlet at temperature T₄ and pressure p₄ (p₄ ≤ or ~ p₂)
- **h₅** Specific enthalpy of the subcooled liquid refrigerant after the condenser at temperature T₅ and pressure p₅ (p₅ ≤ or ~ p₁)
- **h₆** Specific enthalpy of the refrigerant after the expansion device (in a mixed state) at temperature T₆ and pressure p₆ (p₆ ≥ or ~ p₂)
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h7 Specific enthalpy of the saturated refrigerant liquid in the condenser at temperature T7 and pressure
P7 ≤ or ~ P1)
p Pressure (Pa or bar)
P1 Pressure of the saturated refrigerant vapour in the condenser at temperature T1
P2 Pressure of the saturated refrigerant vapour in the evaporator at temperature T2
P3 Pressure of the superheated refrigerant vapour in the compressor outlet (p3 ≥ or ~ P1)
P4 Pressure of the superheated refrigerant liquid in the compressor inlet (P4 ≤ or ~ P2)
P5 Pressure of the subcooled refrigerant liquid after the condenser (P5 ≥ or ~ P1)
P6 Pressure of the refrigerant, in a mixed state, after the expansion device (P6 ≥ or ~ P2)
P7 Pressure of the saturated refrigerant liquid in the condenser (p7 ≤ or ~ P1)
P8 Pressure of the saturated vapour (p1 or p2)
P Power (W)
P1 Thermal power released from the heat pump to the heat transfer medium
Pvpa Total thermal power released from a heat pump installation to the heat transfer medium
Pvpa = P1 + Pef + Pdistr or Pvpa = P1 + Pef + Pdistr
P2 Net thermal power absorbed by the refrigerant circuit
P2kma Net thermal power absorbed by the cold side heat transfer medium
P2kma = P2 - Pef - Pdistr or P2kma = P2 - Pef - Pdistr
Pdistr Thermal power loss from the distribution system (a heat loss is given a negative sign and a heat gain a positive sign)
Pltr Thermal power loss from the compressor
Pk Shaft drive power to compressor
Pem Drive power to compressor motor
Pef Drive power to fan(s)
Pep Drive power to pump(s)
q Flowrate (m^3/s or m^3/h)
qrpm Mass flowrate of the refrigerant
T Thermodynamic temperature (K)
T1 Temperature of the saturated refrigerant vapour in the condenser at pressure p1
T2 Temperature of the saturated refrigerant vapour in the evaporator at pressure p2
T3 Temperature of the superheated refrigerant vapour in the compressor outlet
T4 Temperature of the superheated refrigerant vapour in the compressor inlet
T5 Temperature of the subcooled refrigerant liquid after the condenser
T6 Temperature of the refrigerant after the expansion device
T7 Temperature of the saturated refrigerant liquid after the condenser
T Celsius temperature (°C)
t1 Condensing temperature
t2 Evaporating temperature
t3 - t7 See the designations for thermodynamic temperatures in Figure 2 above
t6 Temperature of the saturated refrigerant vapour (t1 or t2)
tsup Temperature of the superheated refrigerant vapour (t3 or t4)
tbin Temperature, incoming cold side heat transfer medium to the heat pump
tbint Temperature, outgoing cold side heat transfer medium from the heat pump
tbin T Temperature, incoming hot side heat transfer medium to the heat pump
tbut Temperature, outgoing hot side heat transfer medium from the heat pump
θ Temperature difference (K)
θsub Subcooling (θsub = t1 - t5)
θsup Superheat (θsup = t1 - t3)
θsup1 Superheat at the compressor outlet (θsup1 = t3 - t4)
θsup2 Superheat at the compressor inlet (θsup2 = t4 - t2)
Miscellaneous
f Thermal loss factor of the compressor, f = Pib - Pem

5 GENERAL CONDITIONS

Nordtest method NT VVS 115 (Reference 2.1) describes the general conditions to be applied when using this test method.

5.1 Environmental and safety aspects

This test method requires the measurement of two refrigerant pressures. In addition to the general conditions stipulated by NT VVS 115, it must therefore be ascertained that there is absolutely no leakage introduced by the fitting of pressure taps and measuring devices. This must be checked both before and after the actual test by means of appropriate methods for leakage detection. Special care must be taken when the refrigerant is flammable or toxic.

5.2 Documentation

When documenting the tested system and measuring equipment in accordance with NT VVS 115, special care shall be taken in describing the exact location of the temperature and pressure sensors. In particular the location with respect to long sections of piping or additional heat exchangers such as hot gas coolers, subcoolers, intercoolers etc. shall be documented. The design of pressure taps and the methods of fitting and insulating temperature sensors must be described.

6 PRESENTATION OF PERFORMANCE

Nordtest method NT VVS 115 (Reference 2.1) presents the general principles to be adhered to by manufacturers and suppliers in their presentation of performance.

When the purpose of the test is to directly compare measured and stated performance data (NT VVS 115, Section 5.2.4) the following data shall be presented prior to
the use of this test method. The presentation may be in the form of diagrams, equations or sufficiently detailed tables.

6.1 Coefficient of performance

Coefficients of performance shall be presented according to the type of compressor and equipment. For heat pumps one or both of the following coefficients of performance in the heating mode shall be given:
- The motor coefficient of performance, COP_{vp}
- The compressor coefficient of performance, COP_{1k}

For refrigerating equipment one or both of the following coefficients of performance in the cooling mode shall be given:
- The motor coefficient of performance, COP_{2km}
- The compressor coefficient of performance, COP_{2k}

6.2 Power input

Power inputs shall be presented according to the type of compressor. For heat pumps and refrigerating equipment one or both of the following power inputs shall be given:
- The drive power to the compressor motor, P_{em}
- The shaft drive power to the compressor, P_{k}

6.3 Thermal capacities

Thermal capacities shall be presented according to the type of equipment. For heat pumps and refrigerating equipment the following thermal capacities shall be given:
- The thermal power released from the refrigerant to the hot side heat transfer medium, P_{1}
- The thermal power absorbed by the refrigerant from the cold side heat transfer medium, P_{2}
- The thermal losses from the compressor to the ambient atmosphere, P_{för}

6.4 Refrigerant temperatures and pressures

For heat pumps and refrigerating equipment the following temperatures and pressures in the refrigerant shall be stated together with their respective exact locations:
- Temperature of the superheated refrigerant vapour in the compressor outlet, t_{3}
- Temperature of the superheated refrigerant vapour in the compressor inlet, t_{4}
- Temperature of the subcooled refrigerant liquid after the condensor, t_{5}
- Pressure of the superheated refrigerant vapour in the compressor outlet, p_{3}
- Pressure of the superheated refrigerant vapour in the compressor inlet, p_{4}

6.5 Operating conditions

The external operating conditions of the equipment shall be stated in terms of the temperatures of the heat transfer media:
- Temperature of the cold side heat transfer medium, t_{kbin}
- Temperature of the hot side heat transfer medium, t_{vbin}
- Temperature of the cold side heat transfer medium, t_{vbut}
- Temperature of the hot side heat transfer medium, t_{vbin}

7 METHOD OF TEST

This Nordtest method describes the methodology to be used in performance check-ups to infer performance data from measurements in the refrigerant system. The method can be used in situations where a measuring uncertainty of ±15 % is satisfactory. Performance check-ups comprise:
- Determination of the coefficient of performance in the heating mode (COP_{1}),
- Determination of the coefficient of performance in the cooling mode (COP_{2}),
- Determination of the electric power input to the compressor motor (P_{em}).

From these results the thermal capacities can be inferred:
- The thermal power released from the refrigerant to the hot side heat transfer medium (P_{1})
- The thermal power absorbed by the refrigerant from the cold side heat transfer medium (P_{2}).

7.1 Basic principles

It is important to understand the basic principle of inferring performance data from measurements in the refrigerant circuit in order to appreciate the limitations of the methodology. The method relies on specific assumptions regarding the refrigeration cycle, the availability of accurate thermodynamic data for the refrigerant and a reasonable estimate of the thermal losses from the compressor. Thermodynamic data of the refrigerant can be used in the form of pressure-enthalpy charts, as illustrated by Figure 2, or in the form of equations of state (see Appendix A).

Using the assumptions of Figure 3 the thermal balance of the compressor can be expressed as:

\[ P_{em} = q_{Rm} \cdot (h_{3} - h_{4}) + P_{för} = q_{Rm} \cdot (h_{3} - h_{4}) + f \cdot P_{em} \]

where the losses have been expressed as a fraction f of the power input to the compressor.

![Figure 3. Thermal balance of the compressor.](Image)
The thermal output at the condenser can be written as:
\[ P_1 = q_{Rm} \cdot (h_3 - h_5) \]
and the thermal input at the evaporator as:
\[ P_2 = q_{Rm} \cdot (h_4 - h_5) \]
Thus, by elimination of \( q_{Rm} \) the coefficients of performance may be written as:
\[ \text{COP}_1 = \frac{(h_3 - h_5)}{(h_3 - h_4)} (1 - f) \]
and, assuming isenthalpic expansion (i.e. \( h_6 = h_3 \)),
\[ \text{COP}_2 = \frac{(h_4 - h_5)}{(h_3 - h_4)} (1 - f) \]
for \( \text{COP}_2 \) = \( \text{COP}_1 - 1 + f \)

Finally, the heating and cooling capacities may be inferred from the calculated values of COP, if the power input to the compressor is measured,
\[ P_1 = \text{COP}_1 \cdot P_{em} \quad \text{and} \quad P_2 = \text{COP}_2 \cdot P_{em} \]

If equations of state are used to calculate the specific enthalpies from measured values of temperature and pressure, different relations must be used for the different sections of the refrigeration cycle in Figure 2:
- Superheated vapour (3 \( \rightarrow \) 1, 2 \( \rightarrow \) 4, 4 \( \rightarrow \) 3),
- Subcooled condensate (7 \( \rightarrow \) 5)
- Latent heat of vaporisation (1 \( \rightarrow \) 7, 6 \( \rightarrow \) 2).

Often pressure drops in heat exchangers and pipework can be neglected in comparison with the pressure differences at the compressor and expansion device. Then \( p_3 \approx p_1 \approx p_5 \) and \( p_4 \approx p_2 \approx p_6 \). In this case the 4 points necessary for establishing the refrigeration cycle in Figure 2, or the 7 points required when equations of state are used, can be determined by the following measurements:
- Temperature and pressure at the compressor outlet (point 3),
- Temperature and pressure at the compressor inlet (point 4),
- Temperature of the subcooled liquid refrigerant (point 5).

7.2 Measurements

7.2.1 General requirements
The aim of the method of test is to achieve an overall uncertainty of measurement which is less than ±15 % (Level 3 according to NT VVS 115). The general requirements of NT VVS 115 Clause 7.2.1 apply.
The method described in 7.1 assumes that sensors have been positioned in the appropriate positions, that the refrigerant after the condenser is properly subcooled, and that the vapour at the compressor inlet is dry, i.e. properly superheated. Therefore, always check the specifics of the refrigeration system such as the use of intercoolers, reversing valves, liquid separators, long piping, liquid injection in the compressor, oil cooling etc. Also, with each test check the items such as liquid subcooling, suction gas superheat, and the credibility of calculated values of compressor efficiency.

7.2.2 Requirements for stability
The general requirements of NT VVS 115 Clause 7.2.2, accuracy Level 3, apply.

7.2.3 Requirements for measuring instruments
The general requirements for NT VVS 115 Clause 7.2.3 apply.

7.2.4 Measurement of different quantities
For the presentation of performance data and operating points according to Section 6 (6.1-6.5), measurement of some or all of the following quantities is required (see NT VVS 115, Clause 7.2.4, for general references).

<table>
<thead>
<tr>
<th>Quantities always to be measured:</th>
<th>Quantities measured for specific purposes:</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Incoming temperature of heat transfer medium, cold side</td>
<td>- Flowrate of heat transfer medium, hot side</td>
</tr>
<tr>
<td>- Incoming temperature of heat transfer medium, hot side</td>
<td>- Flowrate of heat transfer medium, cold side</td>
</tr>
<tr>
<td>- Outgoing temperature of heat transfer medium, hot side</td>
<td>- Outgoing temperature of heat transfer medium, cold side</td>
</tr>
<tr>
<td>- Power input to compressor(s)</td>
<td>- State of humid air</td>
</tr>
<tr>
<td>- Outlet pressure of the compressor (condensing pressure)</td>
<td>- Pressure difference in the heat transfer system, cold side</td>
</tr>
<tr>
<td>- Inlet pressure of the compressor (evaporating pressure)</td>
<td>- Pressure difference in the heat transfer system, hot side</td>
</tr>
<tr>
<td>- Outlet temperature of the compressor</td>
<td>- Power input to ancillary devices</td>
</tr>
<tr>
<td>- Inlet temperature of the compressor</td>
<td>- Refrigerant temperatures before and after specific heat exchangers</td>
</tr>
<tr>
<td>- Subcooled liquid refrigerant temperature</td>
<td></td>
</tr>
</tbody>
</table>

Appendix E shows some examples of suitable sensor positions for different configurations of refrigeration system.

The general requirements for measuring equipment in NT 115, Clauses 7.2.4.1-7.2.4.6, accuracy Level 3, apply. However, the requirements regarding measurement of refrigerant temperatures and pressures presented in 7.2.4.2-7.2.4.3 take precedence whenever they are more restrictive than those of NT VVS 115 7.2.4.1 and 7.2.4.3.2.

7.2.4.1 General requirements for the uncertainty of individual measurands
The required accuracy of measured refrigerant temperatures and pressures will depend on the type of refrigerant and the operating conditions. As a basic rule, the uncertainty of a measured variable \( x_j \) shall be smaller than...
\[ \Delta x_j < \frac{3}{B_j} \% \]

where \( B_j \) is the propagation coefficient, as described in 7.4, relating the uncertainty of measured variables to the uncertainty of the calculated COP. This ascertains that individual uncertainties will not exceed 3 % and that the probable uncertainty of the calculated COP will not exceed 15 %. Methods of calculating the propagation coefficients may be found in Appendix B and some typical values of required uncertainties are presented in Annex C.

### 7.2.4.2 Measurement of refrigerant pressures

Refrigerant pressures shall be measured with an uncertainty not exceeding the values given in the table below. Propagation coefficients \( B_1 \) and \( B_2 \) apply in the calculation of COP. If the saturation temperatures \( t_1 \) and \( t_2 \) are to be calculated, coefficients \( C_1 \) and \( C_2 \) apply, see 7.2.4.2. The latter requirement is much the stricter of the two.

Tabulated requirements for measurement of refrigerant pressures (%):

<table>
<thead>
<tr>
<th>Item</th>
<th>Refrigerant pressures ( \Delta p/p ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall uncertainty of measurement for pressure</td>
<td>( \frac{3}{B_1} ) or ( \frac{3}{C_1} )</td>
</tr>
<tr>
<td>Calibration of the measuring instrument</td>
<td>1</td>
</tr>
<tr>
<td>Reproducibility of the measuring instrument</td>
<td>0.5</td>
</tr>
<tr>
<td>Resolution of the indicator</td>
<td>0.5</td>
</tr>
</tbody>
</table>

* Irrespective of the tabulated values the overall uncertainty should always be less than ±5 %.

General recommendations regarding measurement of pressures are given in NT VVS 115 Appendix C1.3.

### 7.2.4.3 Measurement of refrigerant temperatures

Refrigerant temperatures shall be measured with an uncertainty not exceeding the values given in the table below.

Tabulated requirements for measurement of refrigerant temperatures in order to calculate COP (±K):

<table>
<thead>
<tr>
<th>Item</th>
<th>Refrigerant temperatures, ±T (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing, ( \Delta t )</td>
<td>Evaporating, ( \Delta t )</td>
</tr>
<tr>
<td>Overall uncertainty of measurement for saturation temperatures</td>
<td>1</td>
</tr>
<tr>
<td>Calibration of the measuring instrument</td>
<td>0.5</td>
</tr>
<tr>
<td>Reproducibility of the measuring instrument</td>
<td>0.2</td>
</tr>
<tr>
<td>Resolution of the indicator</td>
<td>0.2</td>
</tr>
</tbody>
</table>

When refrigerant saturation temperatures are inferred from measurements of pressure, then 7.2.4.1 gives the maximum uncertainties in the pressure measurement.

### 7.2.4.4 Thermal loss factor of the compressor

The thermal loss factor must be known with an uncertainty not exceeding

\[ \Delta f < \frac{3}{B_6} \% \]

Appendix D provides some typical examples of thermal loss factors.

### 7.2.4.5 Measurement of electric power input to the compressor

NT VVS 115, Clause 7.2.4.5, Level 3, applies to the measurement of electric power input to the compressor and other components using electricity.

### 7.3 Evaluation

The general requirements of NT VVS 115 Clause 7.3 apply. The results shall include the condensing pressure and temperature, the evaporating pressure and temperature, the compressor inlet and outlet temperatures, the refrigerant liquid temperature and the temperatures of the hot and cold

side heat transfer media. As a check, the results shall also include the subcooling of the liquid refrigerant, the superheating of the refrigerant vapour at the compressor inlet, the difference between the condensing and outgoing hot side heat transfer medium temperatures, the difference between the evaporating and incoming cold side heat transfer medium temperatures, the calculated Carnot efficiency of the process and the stability of the measured data.

Coefficients of performance shall be calculated in accordance with the basic principles laid out in 7.1. The value of the thermal loss factor that was used shall be stated. The operating conditions in terms of the heat transfer media temperatures on both the condenser and the evaporator sides must always be given.

If measured, the power inputs to the compressor and ancillary equipment shall be stated. Thermal capacities shall be calculated in accordance with the basic principles laid out in 7.1.

Uncertainties of measured and calculated results shall be stated in accordance with the principles described in 7.4.

7.4 Uncertainties

The general requirements of NT VVS 115 Clause 7.4.1-7.4.2 apply.

In this method the absolute values of maximum expected uncertainties are combined instead of using the more complex method of distinguishing between type A and type B uncertainties (see NT VVS 115) and calculating the root mean square uncertainty.

Note: The installation effects for the most important measurements all tend to overestimate the calculated COP. Hence it is preferable to use estimates of maximum uncertainties when using this method.

The required uncertainties of measurement are stated in 7.2.4. The uncertainty of the calculated COP is based on the propagation equation in Appendix B,

\[
\frac{\Delta \text{COP}}{\text{COP}} = \left| B_1 \frac{\Delta P_1}{P_1} \right| + \left| B_2 \frac{\Delta P_2}{P_2} \right| + \left| B_3 \Delta t_3 \right| + \left| B_4 \Delta t_4 \right| + \left| B_5 \Delta t_5 \right| + \left| B_6 \frac{\Delta f}{f} \right|
\]

Methods of calculating the propagation coefficients \( B_1 \) - \( B_6 \) are given in Appendix B. Appendix C provides examples of calculated values and the required measuring uncertainties.

The uncertainties of calculated thermal capacities are estimated according to

\[
\frac{\Delta P_1}{P_1} = \frac{\Delta \text{COP}_1}{\text{COP}_1} + \frac{\Delta P_{\text{em}}}{P_{\text{em}}}
\]

and

\[
\frac{\Delta P_2}{P_2} = \frac{\Delta \text{COP}_2}{\text{COP}_2} + \frac{\Delta P_{\text{em}}}{P_{\text{em}}}
\]

respectively

8 TESTREPORT

The test report shall include the following information, if relevant:

a) Name and address of the testing organisation,

b) Identification number of the test report,

c) Name and address of the organisation or person who ordered the test,

d) Purpose of the test,

e) Make, type designation and manufacturing number of the tested heat pump or refrigeration equipment (see 5.2 and NT VVS 115),

f) List of components such as compressor, refrigerant, expansion device, heat exchangers, pumps, fans (see 5.2 and NT VVS 115),

g) A sketch of the heat pump or refrigeration equipment including connecting points of measuring equipment and sensor locations (see 5.2 and NT VVS 115),

h) Date of the test,

i) Test results in Sl-units (see 7.4),

j) Uncertainty and stability of the test results (see 7.4),

k) Reference to the test method (NT VVS 116),

l) Any deviations from the test method,

m) Date and signature.
APPENDIX A: EXAMPLES OF EQUATIONS OF STATE FOR REFRIGERANTS

This appendix provides examples of equations of state which have been proposed by Cleland (reference A6.1).

A1 RELATION BETWEEN TEMPERATURE AND PRESSURE OF THE SATURATED REFRIGERANT VAPOUR

\[ t_s = \frac{-a_2}{\ln(p_s) - a_1} \cdot a_3 \]

where

\( t_s \) = the temperature of the saturated vapour in °C,
\( p_s \) = the pressure of the saturated vapour in Pa.

A2 ENTHALPY OF THE LIQUID REFRIGERANT

\[ h_5 = a_4 + a_5 \cdot t_5 + a_6 \cdot t_5^2 + a_7 \cdot t_5^3 \]

where

\( h_5 \) = the specific enthalpy of the refrigerant liquid in J/kg,
\( t_5 \) = the temperature of the refrigerant liquid in °C.

A3 ENTHALPY OF THE SATURATED VAPOUR

\[ h_{ml} = a_8 + a_9 \cdot t_s + a_{10} \cdot t_s^2 + a_{11} \cdot t_s^3 \]
\[ h_s = h_{ml} + a_{12} \]

where \( h_{ml} \) is an intermediate result, in J/kg, which is used also in the calculations for superheated vapour (see A4), and \( h_s \) = the specific enthalpy of the saturated vapour in J/kg.

A4 ENTHALPY OF THE SUPERHEATED VAPOUR

In the case of superheated vapour, with a temperature \( t_{sup} \) and a pressure \( p_s \), the superheat \( \theta_{sup} \), i.e. the difference between actual vapour temperature and the saturation temperature \( t_s \), may be calculated according to:

\[ \theta_{sup} = t_{sup} - t_s \]

where

\( \theta_{sup} \) = the superheat in K.

To calculate the enthalpy of superheated vapour the following intermediate step is taken,

\[ h_{m2} = h_{ml} \cdot g \]

where

\[ g = [1 + a_{13} \cdot \theta_{sup} + a_{14} \cdot \theta_{sup}^2 + a_{15} \cdot t_s + a_{16} \cdot \theta_{sup} \cdot t_s + a_{17} \cdot \theta_{sup}^2 + a_{18} \cdot \theta_{sup}^2 \cdot t_s^2] \]

Finally, the specific enthalpy, in J/kg, of the superheated vapour can be derived from

\[ h_{sup} = h_{m2} + a_{12} \]

A5 COEFFICIENTS FOR THE EQUATIONS OF STATE

The following coefficients derive from reference A6.1. Used in the equations of state above, the calculated results are correct within ±0.5 % in the temperature range -60 °C to +60 °C. In the table the IIR standard of setting the specific enthalpy of liquid refrigerant to 200 000 J/kg at a temperature of 0 °C has been used (coefficient \( a_4 \)).

<table>
<thead>
<tr>
<th>Coefficient</th>
<th>R12</th>
<th>R22</th>
<th>R114</th>
<th>R502</th>
<th>R717</th>
</tr>
</thead>
<tbody>
<tr>
<td>a2</td>
<td>2033.5646</td>
<td>2025.4518</td>
<td>2235.3078</td>
<td>1924.9516</td>
<td>2233.8226</td>
</tr>
<tr>
<td>a3</td>
<td>248.30</td>
<td>248.94</td>
<td>238.18</td>
<td>248.46</td>
<td>244.20</td>
</tr>
<tr>
<td>a4</td>
<td>200 000</td>
<td>200 000</td>
<td>200 000</td>
<td>200 000</td>
<td>200 000</td>
</tr>
<tr>
<td>a5</td>
<td>923.88</td>
<td>1170.36</td>
<td>954.99</td>
<td>1114.60</td>
<td>4751.63</td>
</tr>
<tr>
<td>a6</td>
<td>0.83716</td>
<td>1.68674</td>
<td>1.24882</td>
<td>2.12743</td>
<td>2.04493</td>
</tr>
<tr>
<td>a7 (10^-2)</td>
<td>5.3772</td>
<td>5.2703</td>
<td>-0.9671</td>
<td>-1.7679</td>
<td>-37.875</td>
</tr>
<tr>
<td>a8</td>
<td>187 565</td>
<td>250 027</td>
<td>173 522</td>
<td>187 890</td>
<td>1 441 467</td>
</tr>
<tr>
<td>a9</td>
<td>428.992</td>
<td>367.265</td>
<td>622.742</td>
<td>406.454</td>
<td>920.154</td>
</tr>
<tr>
<td>a10</td>
<td>-0.75152</td>
<td>-1.84133</td>
<td>0.21971</td>
<td>-1.59402</td>
<td>-10.20556</td>
</tr>
<tr>
<td>a11 (10^-3)</td>
<td>-5.6695</td>
<td>-1.14556</td>
<td>-5.3029</td>
<td>-13.601</td>
<td>-26.512</td>
</tr>
<tr>
<td>a12</td>
<td>163 994</td>
<td>155 482</td>
<td>163 856</td>
<td>158 898</td>
<td>15 689</td>
</tr>
<tr>
<td>a13 (10^-3)</td>
<td>3.43263</td>
<td>2.85446</td>
<td>3.92169</td>
<td>3.80815</td>
<td>1.68973</td>
</tr>
<tr>
<td>a14 (10^-7)</td>
<td>7.27473</td>
<td>4.0129</td>
<td>35.0776</td>
<td>14.4572</td>
<td>-3.47675</td>
</tr>
<tr>
<td>a15 (10^-4)</td>
<td>7.27759</td>
<td>13.3626</td>
<td>-5.29945</td>
<td>16.5858</td>
<td>8.55325</td>
</tr>
<tr>
<td>a16 (10^-6)</td>
<td>-6.63650</td>
<td>-8.11617</td>
<td>-2.40700</td>
<td>-12.5256</td>
<td>-3.04755</td>
</tr>
<tr>
<td>a17 (10^-9)</td>
<td>6.95693</td>
<td>14.1194</td>
<td>5.79432</td>
<td>20.5676</td>
<td>9.79201</td>
</tr>
<tr>
<td>a18 (10^-10)</td>
<td>-4.17264</td>
<td>-9.53294</td>
<td>-2.32032</td>
<td>-15.5967</td>
<td>-3.62549</td>
</tr>
</tbody>
</table>

A6 REFERENCES

APPENDIX B: PROPAGATION OF UNCERTAINTIES

Differentiating the basic equation of the inferred value of COP,
\[
\text{COP} = \frac{(h_3 - h_5)}{(h_3 - h_4)} \cdot (1 - f)
\]
we have
\[
\frac{\Delta \text{COP}}{\text{COP}} = \frac{(h_5 - h_4)}{(h_3 - h_4)} \cdot \Delta h_3 + \frac{1}{(h_3 - h_4)} \cdot \Delta h_4 - \frac{1}{(h_3 - h_5)} \cdot \Delta h_5 - \frac{1}{(1 - f)} \cdot \Delta f
\]
with
\[
\begin{align*}
A_1 &= \frac{(h_5 - h_4)}{(h_3 - h_4) \cdot (h_3 - h_5)}, \\
A_2 &= \frac{1}{(h_3 - h_4)}, \\
A_3 &= \frac{1}{(1 - f)} \quad \text{and} \\
A_4 &= \frac{1}{(1 - f)}
\end{align*}
\]

Differentiation of \( h = h(p, t) \) yields:
\[
\begin{align*}
\Delta h_3 &= \frac{\partial h_3}{\partial p_1} \Delta p_1 + \frac{\partial h_3}{\partial t_1} \Delta t_1, \\
\Delta h_4 &= \frac{\partial h_4}{\partial p_2} \Delta p_2 + \frac{\partial h_4}{\partial t_2} \Delta t_2, \\
\Delta h_5 &= \frac{\partial h_5}{\partial p_1} \Delta p_1 + \frac{\partial h_5}{\partial t_5} \Delta t_5 = \frac{\partial h_5}{\partial t_5} \Delta t_5
\end{align*}
\]
(the liquid enthalpy is practically independent of pressure).

Combination of the relations above gives:
\[
\frac{\Delta \text{COP}}{\text{COP}} = \frac{\Delta p_1}{p_1} + \frac{\Delta p_2}{p_2} + B_3 \cdot \Delta t_3 + B_4 \cdot \Delta t_4 + B_5 \cdot \Delta t_5 + B_6 \cdot \Delta f
\]
with
\[
\begin{align*}
B_1 &= \frac{-(h_4 - h_5) \cdot p_1}{(h_3 - h_4) \cdot (h_3 - h_5) \cdot \frac{\partial h_3}{\partial p_1} [-]}, \\
B_2 &= \frac{p_2}{(h_3 - h_4) \cdot \frac{\partial h_4}{\partial p_2} [-]}, \\
B_3 &= \frac{-(h_4 - h_5) \cdot \frac{\partial h_3}{\partial t_3} [K^{-1}]}{(h_3 - h_4) \cdot (h_3 - h_5) \cdot \frac{\partial h_3}{\partial t_3} [K^{-1}]}, \\
B_4 &= \frac{1}{(h_3 - h_4) \cdot \frac{\partial h_4}{\partial t_4} [K^{-1}]}, \\
B_5 &= \frac{-1}{(h_3 - h_5) \cdot \frac{\partial h_5}{\partial t_5} [K^{-1}]}, \quad \text{and} \\
B_6 &= \frac{f}{(1 - f)} [-]
\end{align*}
\]

The partial derivatives may be estimated using the relationships from Appendix A,
\[
B_1 = \frac{\partial h_3}{\partial p_1} \left[ \frac{1}{(\ln p_1 - a_1)^2} \right] \left[ g_1 \left( a_9 + 2a_{10} \cdot t_1 + 3a_{11} \cdot t_1^2 \right) + h_{n1}(-a_{13} - 2a_{14} \cdot \theta_{\text{sup}} + a_{15}(\theta_{\text{sup}} - t_1) \\
+ a_{16} \cdot \theta_{\text{sup}}(\theta_{\text{sup}} - 2t_1) + a_{17} \cdot t_1(2\theta_{\text{sup}} - t_1) \\
+ 2a_{18} \cdot \theta_{\text{sup}} \cdot t_1(\theta_{\text{sup}} - t_1)) \right],
\]
\[
B_2 = \frac{\partial h_4}{\partial p_2} \left[ \frac{1}{(\ln p_2 - a_2)^2} \right] \left[ g_2 \left( a_9 + 2a_{10} \cdot t_2 + 3a_{11} \cdot t_2^2 \right) + h_{n2}(-a_{13} - 2a_{14} \cdot \theta_{\text{sup}} + a_{15}(\theta_{\text{sup}} - t_2) \\
+ a_{16} \cdot \theta_{\text{sup}}(\theta_{\text{sup}} - 2t_2) + a_{17} \cdot t_2(2\theta_{\text{sup}} - t_2) \\
+ 2a_{18} \cdot \theta_{\text{sup}} \cdot t_2(\theta_{\text{sup}} - t_2)) \right],
\]
\[
B_3 = h_{n3}(a_{13} + 2a_{14} \cdot \theta_{\text{sup}} + a_{15} \cdot t_1 \\
+ 2a_{16} \cdot \theta_{\text{sup}} \cdot t_1 + a_{17} \cdot t_1^2 + 2a_{18} \cdot \theta_{\text{sup}} \cdot t_1^2),
\]
\[
B_4 = h_{n4}(a_{13} + 2a_{14} \cdot \theta_{\text{sup}} + a_{15} \cdot t_2 \\
+ 2a_{16} \cdot \theta_{\text{sup}} \cdot t_2 + a_{17} \cdot t_2^2 + 2a_{18} \cdot \theta_{\text{sup}} \cdot t_2^2),
\]
\[
B_5 = a_5 + 2a_6 \cdot t_5 + 3a_7 \cdot t_5^2
\]

Often the first order terms in \( t \) and \( \theta \) are sufficient in order to estimate the propagation coefficients.

When saturation temperatures, \( t_s \), are inferred from measured saturation pressures, \( p_s \) the uncertainty of the temperature may be expressed as
\[
\Delta t_s = \frac{a_2}{(\ln p_s - a_1)^2} \cdot \frac{\Delta p_s}{p_s} = C_9 \cdot \frac{\Delta p_s}{p_s}
\]

Specifically, for the condensing and evaporating temperatures, we have
\[
\Delta t_1 = C_1 \cdot \frac{\Delta p_1}{p_1} \quad \text{and} \quad \Delta t_2 = C_2 \cdot \frac{\Delta p_2}{p_2}
\]

The table below provides examples of derived propagation coefficients for a simple process between condensation and evaporation temperatures of \( t_1 = 40 ^\circ \text{C}, t_2 = -10 ^\circ \text{C} \). The subcooled liquid temperature is assumed to be \( t_5 = 35 ^\circ \text{C} \) \( \theta_{\text{sup}} = 5 ^\circ \text{K} \) and the superheated suction vapour is assumed to be \( t_4 = +5 ^\circ \text{C} \theta_{\text{sup}} = 15 ^\circ \text{K} \). The thermal loss factor of the compressor is \( f = 0.68 \) (i.e. 8 %) and the refrigeration cycle has a Carnot efficiency of \( \eta_{C2} = 0.45 \) (i.e. 45 %; \( \text{COP}_{C2} = 5.85 \) between the given temperatures).
### Table of typical values regarding propagation coefficients for uncertainties in the calculation of COP

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Refrigerant</th>
<th>R12</th>
<th>R22</th>
<th>R114</th>
<th>R502</th>
<th>R717</th>
</tr>
</thead>
<tbody>
<tr>
<td>p1 (bar)</td>
<td></td>
<td>9.61</td>
<td>15.34</td>
<td>3.37</td>
<td>16.77</td>
<td>15.56</td>
</tr>
<tr>
<td>p2 (bar)</td>
<td></td>
<td>2.19</td>
<td>3.54</td>
<td>0.58</td>
<td>4.14</td>
<td>2.91</td>
</tr>
<tr>
<td>t3 (°C)</td>
<td>87.9</td>
<td>108.6</td>
<td>65.4</td>
<td>78.7</td>
<td>199.9</td>
<td></td>
</tr>
<tr>
<td>t4 (°C)</td>
<td>5.0</td>
<td>5.0</td>
<td>5.0</td>
<td>5.0</td>
<td>5.0</td>
<td></td>
</tr>
<tr>
<td>t5 (°C)</td>
<td>35.0</td>
<td>35.0</td>
<td>35.0</td>
<td>35.0</td>
<td>35.0</td>
<td></td>
</tr>
</tbody>
</table>

#### Specific enthalpies

| h3 (kJ/kg)   | 403.3       | 475.9  | 381.7 | 395.1 | 1904.9 |
| h4 (kJ/kg)   | 356.5       | 411.8  | 341.3 | 352.8 | 1481.6 |
| h5 (kJ/kg)   | 233.6       | 243.3  | 234.9 | 241.5 | 367.2  |

#### Derivates

| \( \frac{\partial h_3}{\partial p_1} \) (kJ/kg/bar) | -1.1 | -0.9 | -1.2 | -1.1 | -1.4 |
| \( \frac{\partial h_4}{\partial p_2} \) (kJ/kg/bar) | -1.9 | -1.8 | -1.9 | -1.4 | -8.7 |
| \( \frac{\partial h_3}{\partial p_3} \) (kJ/kg/K)   | 0.7  | 0.8  | 0.8  | 0.9  | 2.2  |
| \( \frac{\partial h_4}{\partial p_4} \) (kJ/kg/K)   | 0.6  | 0.7  | 0.7  | 0.7  | 2.3  |
| \( \frac{\partial h_5}{\partial p_5} \) (kJ/kg/K)   | 1.0  | 1.3  | 1.0  | 1.3  | 4.8  |

#### Propagation coefficients

| B1 (−)   | 0.157 | 0.157 | 0.070 | 0.322 | 0.038 |
| B2 (−)   | -0.087| -0.101| -0.028| -0.138| -0.060|
| B3 (K⁻¹) | -0.011| -0.009| -0.014| -0.015| -0.004|
| B4 (K⁻¹) | 0.013 | 0.011 | 0.017 | 0.016 | 0.005 |
| B5 (K⁻¹) | -0.006| -0.006| -0.007| -0.008| -0.003|
| B6 (−)   | -0.087| -0.087| -0.087| -0.087| -0.087|
| C1 (K⁻¹) | 40.87 | 41.21 | 34.62 | 43.21 | 36.15 |
| C2 (K⁻¹) | 27.93 | 28.19 | 23.29 | 29.54 | 24.55 |
APPENDIX C: EXAMPLES OF REQUIRED MEASURING UNCERTAINTIES FOR DIFFERENT TYPES OF REFRIGERANTS

The table below provides examples of the maximum permissible measuring uncertainties of temperatures and pressures for some common types of refrigerants. The values are based on the relation regarding propagation of uncertainties given in 7.6, assuming equal contributions of 15/6 = 2.5 < 3 % from each of the 6 contributors to the total uncertainty of 15 %. The requirements apply for operation conditions according to the example in Appendix B.

<table>
<thead>
<tr>
<th>Type of refrigerant</th>
<th>Maximum permissible uncertainty (±)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\Delta p_1$ (P)</td>
</tr>
<tr>
<td>R12</td>
<td>19.1</td>
</tr>
<tr>
<td>R22</td>
<td>19.1</td>
</tr>
<tr>
<td>R114</td>
<td>42.6</td>
</tr>
<tr>
<td>R502</td>
<td>9.3</td>
</tr>
<tr>
<td>R134a</td>
<td></td>
</tr>
<tr>
<td>R290</td>
<td></td>
</tr>
<tr>
<td>R717</td>
<td>78.2</td>
</tr>
</tbody>
</table>
APPENDIX D: EXAMPLES OF TYPICAL COMPRESSOR THERMAL LOSS FACTORS

The following table provides information regarding typical loss factors for different types of units and precautions to be taken in specific cases.

<table>
<thead>
<tr>
<th>Size and type of compressor</th>
<th>Type of installation</th>
<th>Loss factor (%)</th>
<th>Precautions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Typical</td>
<td>Reported range</td>
</tr>
<tr>
<td>Small hermetic ((P_{em} &lt; 5 \text{ kW}))</td>
<td>Normal insulation; indoors</td>
<td>7</td>
<td>5–15</td>
</tr>
<tr>
<td></td>
<td>Poor insulation; outdoors</td>
<td>15</td>
<td>10–25</td>
</tr>
<tr>
<td>Large hermetic ((5 \text{ kW} &lt; P_{em} &lt; 50 \text{ kW}))</td>
<td>Normal insulation; indoors</td>
<td>5</td>
<td>3–10</td>
</tr>
<tr>
<td>Small semi hermetic ((P_{em} &lt; 10 \text{ kW}))</td>
<td>Poor insulation; outdoors</td>
<td>7</td>
<td>5–15</td>
</tr>
<tr>
<td>Large semi hermetic ((10 \text{ kW} &lt; P_{em} &lt; 100 \text{ kW}))</td>
<td>Normal insulation; indoors</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Poor insulation; outdoors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Screw; dry</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Screw; oil injection</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centrifugal</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX E: EXAMPLES OF HOW TO POSITION SENSORS IN DIFFERENT TYPES OF REFRIGERATION SYSTEMS

In the basic refrigeration system (Figure E1) sensors \( p_1, p_2, t_3, t_4, \) and \( t_5 \) suffice to map the refrigeration cycle in the enthalpy-pressure diagram. Hence COP\(_1\) and COP\(_2\) may be calculated. Adding a measurement of \( P_{em} \) also makes it possible to calculate \( P_1 \) and \( P_2 \). To record the operating conditions, sensors for \( t_{kbin} \) and \( t_{vbut} \) (in the case of air cooled condensers \( t_{vbin} \)) are also required.

E1 BASIC REFRIGERATION SYSTEM

![Figure E1. Sensor positions in a basic refrigeration cycle.](image)

E2 REFRIGERATION SYSTEM INCLUDING DESUPERHEATERS AND SUBCOOLERS

In addition to the basic measurements it may be useful to measure also the inlet and outlet temperatures of the extra heat exchangers plus the refrigerant temperatures between these heat exchangers and the condenser.

![Figure E2. Sensor positions in a refrigeration/heat pump cycle including separate heat exchangers for desuperheating and subcooling of the refrigerant.](image)

E3 REFRIGERATION SYSTEM INCLUDING A SUCTION INTERCOOLER

In addition to the basic measurements it may be useful to measure also the refrigerant temperatures at the inlets and outlets on both sides of the suction intercooler.

![Figure E3. Sensor positions in a refrigeration/heat pump cycle including a suction intercooler.](image)

E4 REVERSIBLE REFRIGERATION SYSTEM

In addition to the basic measurements it may be useful to measure also the refrigerant temperatures at the three outlets of the 4-way reversing valve.

![Figure E4. Sensor positions in a reversible refrigeration/heat pump cycle including a 4-way reversing valve (e.g. air-conditioning heat pumps).](image)