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اللجنة التنفيـذية للصندوق المتعدد الأطراف
لتنفيــذ بروتوكـول مونتريــال
الاجتمـــــاع الخامس والسبعون
مونتريال، 16- 20 نوفمبر/ تشرين ثاني 2015

تقرير عن مؤشر الصندوق المتعدد الأطراف المتعلق بالأثر على المناخ (المقرر 65/73 (ب))

خلفية

1. استجابة للمقرر 23/69، قدمت للأمانة للاجتماع الثالث والسبعين رؤية محدثة وكاملة الإعدد بشأن مؤشر الصندوق المتعدد الأطراف عن الأثر على المناخ وهي الرؤية التي قام باستعراضها بصورة مستقلة ثلاثة من الخبراء المحنكين.[1]

2. وخلال الاجتماع الثالث والسبعين، طلبت اللجنة التنفيذية من الأمانة، *ضمن جملة أمور*، وضع الصيغة النهائية لمؤشر الصندوق المتعدد الأطراف المتعلقة بالأثر على المناخ مع مراعاة مايلي، حسب مقتضى الحال، التوصيات التي قدمها الخبراء التقنيون، وتقرير التقرير الخامس الصادر عن الفريق الحكومي الدولي المعني بتغير المناخ/ وتقاسم إدارة المؤشر مع هذا الفريق، والدعوة الى تقديم معلومات مسترجعة عن الأداة من البنك الدولي في سياق عمله مع مصارف التنمية الأخرى المتعددة الأطراف لتنسيق حساب غازات الاحتباس الحراري في إطار حوافظها الاستثمارية، وعمله عن إصلاح إعانات الطاقة. ويتوقع أن يقدم التقرير النهائي للجنة التنفيذية في موعد لا يتجاوز الاجتماع الخامس والسبعين (المقرر 65/73).

التعديلات التي أدخلت على مؤشر الصندوق المتعدد الأطراف

3. استنادا الى المعلومات المسترجعة التي وصلت اللجنة التنفيذية على النحو الذي يجسده المقرر 65/73، عدلت الأمانة المؤشر بإدراج التعليقات الواردة من خبراء الاستعراض المستقلين وقامت بتحديث قيم قدرات الاحترار العالمي في المؤشر استنادا الى تقرير اليونيب الخاص بعام 2014 الصادر عن لجنة الخيارات التقنية المعنية بالتبريد وتكييف الهواء ومضخات الحرارة، وهو التقرير الذي يعتمد على التقييم الخامس للفريق الحكومي الدولي المعني

[1] ركزت وجهات نظر الخبراء على مدى كفاية هيكل أداة مؤشرة الصندوق المتعدد الأطراف ومدى مواتيته للمستخدمين، والجوانب الإيجابية والسلبية للحسابات، ومدى كفاية توليد وعرض النتائج واتساق الأداة مع عمل هيئات الأمم المتحدة الأخرى العاملة في القضايا ذات الصلة بالمناخ.

بتغير المناخ مع بعض التصويبات بالاعتماد على البيانات المنقحة الخاصة بفترات الحياة. ويرد دليل التعليقات المستكمل في المرفق الأول بهذا التقرير.

المعلومات المسترجعة المتلقاة من الفريق الحكومي الدولي المعني بتغير المناخ والبنك الدولي

4. تقاسمت الأمانة أداة مؤشر الصندوق المتعدد الأطراف بشأن الأثر على المناخ مع الفريق الحكومي الدولي المعني بتغير المناخ والبنك الدولي للحصول على معلوماتها المسترجعة.

5. وفي 8 سبتمبر/ أيلول 2015، تلقت الأمانة معلومات مسترجعة من نائب أمين الفريق الحكومي الدولي المعني بتغير المناخ يشير الى أن "تقييم مقترحات السياسات النوعية سواء من الدول الأعضاء أو المؤسسات الدولية تتجاوز التفويض الحالي الممنوح للفريق الحكومي الدولي المعني بتغير المناخ كما أنه لا يملك الهيكل أو الموارد اللازمة لمعالجة هذه الطلبات".

6. وفي 21 أكتوبر/ تشرين الأول 2015، تلقت الأمانة معلومات مسترجعة من البنك الدولي تشير الى أنه يتقاسم أداة المؤشر قبل وبعد التعديل الأخير مع مصارف التنمية[2] المتعددة الأطراف العاملة بشأن الفريق العامل التابع للمؤسسة المالية الدولية عن محاسبة غازات الاحتباس الحراري[3]. كما قدم البنك الدولي المؤشر الى اجتماع الفريق العامل الذي عقد في واشنطن العاصمة من 23 الى 25 سبتمبر/ أيلول 2015 شارحا قدرة المؤشر على تقدير الأثر على المناخ (المباشر وغير المباشر) الناشيء عن معدات التبريد وتكييف الهواء. وقد وصلت المعلومات المسترجعة التالية "لاحظ الكثير من ممثلي مصارف التنمية المتعددة الأطراف الأثر المباشر وغير المباشر الناشيء عن انبعاثات غازات التبريد خلال تركيب وخدمة المعدات التي يمكن أن تكون كثيرة وخاصة في البلدان التي لديها كثافة كربون منخفضة في توليد الكهرباء. ويواصل الفريق العامل التابع للمؤسسة المالية الدولية العمل لوضع الصيغة النهائية لنهج مشترك لتقديمه لمؤتمر الأطراف في إتفاقية الأمم المتحدة بشان تغير المناخ في باريس. أما عن إطار التنسيق الخاص بالمؤسسات المالية الدولية، فإنها سوف تعتمد على أي مصدر للانبعاثات يتضمن النطاق الأول والنطاق الثاني بل وحتى النطاق الثالث للانبعاثات (حسب التحديد الوارد في بروتوكول محاسبة غازات الاحتباس الحراري)[4] والتي تعتبر كبيرة بصورة كافية، ولذا فإن من المحتمل أن تؤخذ هذه في الاعتبار في الحالات التي يعتبر فيها التبريد وتكييف الهواء مصدرا كبيرا لانبعاثات غازات الاحتباس الحراري، ، وسوف تكون أداة المؤشر في هذه الحالات مفيدة بالتأكيد في تقدير الأثر المباشر على المناخ من تسرب غازات التبريد من تشغيل معدات التبريد وتكييف الهواء والتخلص منها".

7. وتلاحظ الأمانة مع التقدير الرد الذي ورد من الفريق الحكومي الدولي المعني بتغير المناخ، والجهود التي بذلها البنك الدولي لتقاسم أداة مؤشر الصندوق المتعدد الأطراف مع الهيئات ذات الصلة العاملة في حساب غازات الاحتباس الحراري، والمعلومات المسترجعة التي تلقيت.

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[2] الوكالة الفرنسية للتنمية، ومصرف التنمية الأفريقي، ومصرف التنمية الألماني، والبنك الأوروبي للتعمير والتنمية وبنك الاستثمار الأوروبي، ومصرف التنمية للبلدان الأمريكية، والمؤسسة المالية الدولية، ومصرف الاستثمار لدول الشمال، ومؤسسة تمويل البيئة في دول الشمال والبنك الدولي.

[3] يعمل هذا الفريق على و ضع نهج مشترك لحساب انبعاثات غازات الاحتباس الحراري مع مشروعات كفاءة الطاقة وفقا لإطار المؤسسة المالية الدولية بشأن النهج المنسق لحساب غازات الاحتباس الحراري. وسيجري حساب التغيير الناشيء عن ذلك في انبعاثات ثاني أكسيد الكربون ذات الصلة من خلال مقارنة الانبعاثات المتوقعة بعد الاستثمار مع خط أساس سابق للاستثمار. ويشكل الفرق بين خط الأساس وانبعاثات المشروع ،انبعاثات غازات الاحتباس الحراري أو خفض هذه الانبعاثات أو ثلافيها. وسيطبق النهج على حساب غازات الاحتباس الحراري في المشروعات الاستثمارية لكفاءة الطاقة و/أو مكونات المشروع التابع للمؤسسة المالية المتعددة الأطراف. ويطبق كذلك على حساب أنشطة التخفيف من تغير المناخ والإبلاغ عنها بما في ذلك التحسينات على كفاءة الطاقة من خلال تركيب أجهزة ومعدات أكثر كفاءة في المباني والخدمات العامة والصناعة فى إطار المؤسسة المالية متعددة الأطراف للنهج المتساوق لحساب غازات الإحتباس
ا
http://www.worldbank.org/content/dam/Worldbank/document/IFI_Framework_for_Harmonized_Approach_to_Greenhouse_Gas_Accounting.pdf

[4] www.ghgprotocol.org

الخطوات التالية

8. وحسب الممارسات المعمول بها، ستواصل الأمانة استخدام مؤشر الصندوق المتعدد الأطراف المنقح بشأن الأثر على المناخ لتقدير الأثر على المناخ من المشروعات في قطاعات تصنيع التبريد وتكييف الهواء الواردة في المرحلة الثالثة والمراحل التالية من خطط إدارة إزالة المواد الهيدروكلوروفلوروكربونية.

التوصية

9. قد ترغب اللجنة التنفيذية فيمايلي:

(أ) أن تحاط علما بمؤشر الصندوق المتعدد الأطراف كامل الإعداد (المقرر65/73(ب)) الوارد في الوثيقة UNEP/OzL.Pro/ExCom/75/78؛

(ب) أن تلاحظ مع التقدير الرد الوارد من الفريق الحكومي الدولي المعني بتغير المناخ، والمعلومات المسترجعة الواردة من البنك الدولي في سياق عمله مع مصارف التنمية الأخرى المتعددة الأطراف لتنسيق حساب غازات الاحتباس الحراري عبر حوافظها الاستثمارية، وعمله في إصلاحات إعانات الطاقة؛

(ج) أن تأخذ علما بأن الأمانة سوف تواصل حساب الأثر على المناخ من المشروعات الاستثمارية في قطاعات تصنيع التبريد وتكيف الهواء التي تطبق نموذج مؤشر الصندوق، وللمشروعات الاستثمارية في جميع قطاعات التصنيع الأخرى التي تطبق المنهجيات الواردة في الفقرة 14 من الوثيقة UNEP/OzL.Pro/ExCom/73/54.

Annex I

MCII Model, Refrigeration and AC systems

MCII Model, Refrigeration and AC systems

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1 Introduction

1.1 General

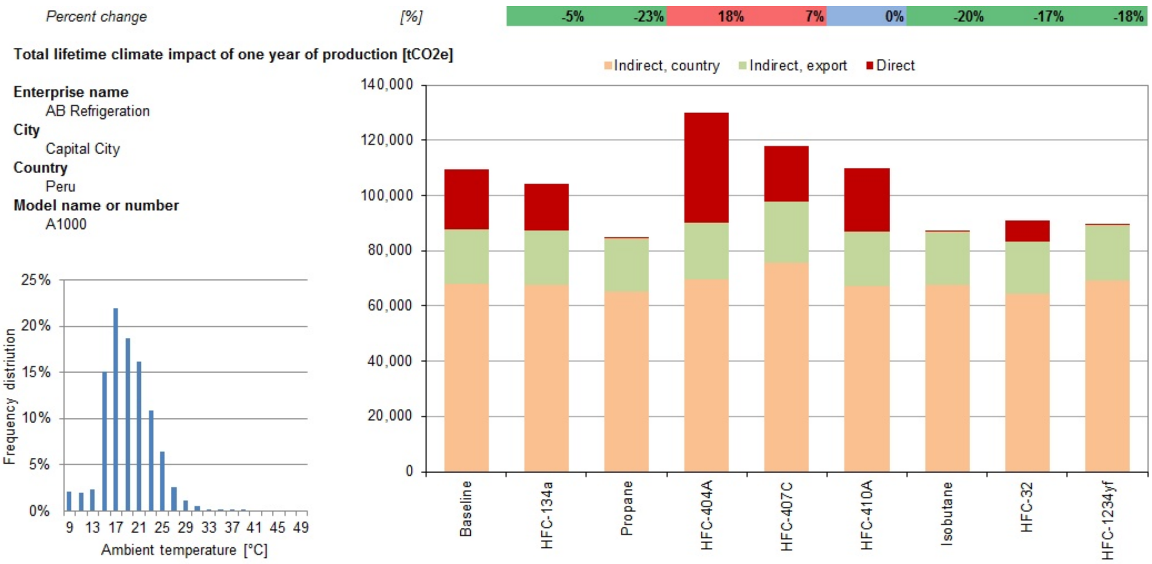
The Multilateral Fund Climate Impact Indicator (abbreviated to MCII) has been developed to allow an indication of the effect on the climate of future conversion projects in the refrigeration and air-conditioning manufacturing sectors from HCFCs (baseline) to alternative refrigerants funded by the Multilateral Fund. The MCII is not meant to replace any analysis undertaken on the basis of detailed performance information of specific equipment, such as a life cycle climate performance (LCCP) or a life cycle analysis (LCA).

The MCII is not a development tool for the refrigeration or air-conditioning system being studied. The internal model for calculating the energy consumption of the system is based on first principles for the thermodynamic circuit. It effectively calculates cycles based on average system characteristics, such as expected compressor efficiencies and heat exchanger performances. The performance of alternative refrigerants is then estimated based on thermodynamic and transport properties differences with HCFC-22. In principle it is assumed that the alternative refrigerants have no impact on the compressor efficiency and only to a small extent on heat exchanger performances, which in reality may not be the case as these components may be, or may need, optimization towards the alternative selected.

The MCII for refrigeration and air-conditioning activities takes into account:
1. the emissions of refrigerant during manufacturing, operation and at the end of life, called the direct emissions; as well as
2. the energy consumption of products using HCFC and their alternatives as refrigerants, called the indirect emissions. This requires an estimation of the energy consumption of the system of interest, which is handled in the so called cycle calculations.

In a first step the model calculates the emission of one refrigeration or air-conditioning unit over its lifetime as a sum of direct and indirect effects and multiplies the result with the amount of units produced in one year. This result represents the climate impact of the annual production for a given technology. For a qualitative comparison of different alternatives, the ratio between the baseline (HCFC) and the alternative refrigerants is used (percentage values). For aggregated, sector-or country-wide figures, the difference between the two is being used (absolute values in tonnes of CO2 equiv.). Negative values for the MCII denote a reduction in the climate impact as compared to the baseline, positive values an increase. The description of the emission model presents more detail.

An example of a part of a typical output sheet is shown below:



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1.2 Nomenclature

c_p	Specific heat [J/(kgK)]
c	Correction factor for conductance [-]
CI	Climate impact [t CO2e]
CL	Compressor clearance volume ratio = Dead volume / stroke volume [-]
COP	Coefficient of performance [-]
E	Energy consumption [kWh] or [GWh]
f	Frequency of occurrence [-],Generic function name, e.g. for refrigerant properties
h	Specific enthalpy [J/kg], Number of hours [h]
k	Polytropic exponent
LMTD	Logarithmic mean temperature difference [K]
M	Mass [kg]
$\dot{m}$	Mass flow rate [kg/s]
NTU	Number of transfer units [-]
n	Number of events [-]
p	Pressure [Pa] or [bar]
Q	Heat flow [W]
R	Refrigerant ratio's, e.g. leak, recharged, etc [-]
r	Ratio [-]
s	Entropy [J/(kgK)]
T	Temperature [K] or [°C]
UA	Conductance [W/K]
$\dot{V}$	Volumetric flow rate [m^3/s]
Δt	Time difference [y]
η	Heat exchanger effectiveness (temperature efficiency) [-]
η_v	Compressor volumetric efficiency [-]
η_i	Compressor isentropic efficiency [-]
ρ	Density [kg/m^3]
ϕ_v	Compressor displacement volume flow rate [m^3/s]

Indices

AC	air-conditioning
amb	ambient
air	air

annual per year
base,AC base condition for an air-conditioning unit
c condenser, condensation
comp compressor
design at design condition
e evaporator, evaporation
eol end of life
export export related part
in inlet
indoor indoor (inside the room)
L load
leak leakage
life life time (over entire life span)
mfg manufacturing
out outlet, exit
r refrigerant
R refrigeration, cooling
sat saturated refrigerant condition
sub subcooled refrigerant condition (liquid phase)
super superheated refrigerant condition (vapour phase)
sv saturated vapour
sl saturated liquid

1.3 Requirements

The model is entirely written as an MS Excel workbook using Visual Basic Macro's (VBA). The workbook is saved as an Excel "xlsm" file which requires Excel 2007 or higher to function. When the workbook is opened in MS Excel, the user may be requested to enable the macro's included in the model.

The workbook can be saved at any desired location and does not require any further files or settings. The workbook contains a number of hidden worksheets and also the VBA code is generally hidden. These sheets can be made visible when the appropriate password is given (Press ctrl-shift-U to unhide and ctrl-shift-P to hide and protect all files).

Further the workbook contains an expiration data, after which the model ceases to function. The user is requested to contact the Multilateral Fund secretariat for a model update.

When the workbook opens it may happen that Excel gives a warning about the presence of circular references. This warning can be neglected, once the macro's in the workbook are enabled the workbook itself will make the correct settings with respect to circular references automatically.

The calculation time depends on the PC which is used. The calculation may increase if also other workbooks are kept open next to the MCII model in Excel.

2 How to use the model

2.1 Introduction

This section aims for a fast introduction of the model, for readers interested in the background of the model, the chapters cycle model, emission model and model implementation are of interest.

The model allows the calculation of 6 different type of systems. For each of these systems a number of characteristic parameters have been defined which can not be changed by the user of the model. These are all discussed in the system topic.

For each system, a calculation is performed using HCFC-22 as the base line refrigerant at so called design conditions. This can be 32 or 40 °C ambient temperature depending on the country. Next calculations are performed with a series of selected alternative refrigerants.

The calculations are performed for a range of ambient temperatures (generally from 9 to 49 °C). The thermal load and the efficiency will vary over this ambient temperature range and consequently also the energy consumed by the system. Combining this energy consumption with the number of hours per year at which such ambient temperature is present, it is possible to calculate an annual energy consumption.

Once the annual energy consumption is known, it is possible to calculate the relevant CO2 emission for a manufacturer if the numbers of units annually produced is given and their expected life time. For this it is necessary to know the so called carbon intensity which is the ratio between the amount of CO2 emitted per kWh generated, which is a specific number per country. The resulting value represent the indirect emissions.

To calculate direct emissions, the user has to enter the typical charge of the system of interest. By using typical, fixed, numbers for leak rate, servicing frequency and end of life recovery, the model estimates the total refrigerant emitted over its lifetime. Combining this with the GWP value of the refrigerant of interest a direct CO2 emission is calculated.

Finally the model brings all these parameters together and presents the total climate impact for the base line systems as well as for the alternative refrigerants selected.

Some examples are included to illustrate the use of the model

2.2 Type of systems

The type of systems handled in the MCII model deal with three different applications:
1. Air-conditioning
2. Commercial cooling equipment
3. Commercial freezing equipment

The latter two systems are split as these have typically different characteristics due to the difference in temperature level.

For all these systems a difference is made between factory assembled system, where the charging of the units takes place in the factory and generally a more precise control and matching of components can be realized compared to on-site assembled systems. The latter typically deals with separate components, e.g. a condensing unit put on top of a cold store and an evaporator/fan assembly placed inside the cold store.

For each type of system a numbers of parameters are pre-set to typical values. E.g in stead of entering real compressor characteristics, a typical isentropic efficiency has been chosen. Another example is that instead of entering real condenser sizes in the model, a typical temperature difference between air and condensation temperature is chosen. The list of pre-set parameters is given below, where the blue values are actual fixed input parameters and the brown values are calculated:

Refrigeration/AC system settings								
System Type		***AC, factory assembly***	***AC, on site assembly***	***Commercial Cooling, factory assembly***	***Commercial Cooling, on site assembly***	***Commercial Frozen, factory assembly***	***Commercial Frozen, on site assembly***	***Symbol***
Evaporator								
Air evaporator inlet	*[°C]*	26	26	7	7	-16	-16	$T_{e,air,in}$
Air temperature difference	*[K]*	7	7	3	3	3	3	$_{,air,in} - T_{e,air,out}$
Air evaporator outlet	*[°C]*	19	19	4	4	-19	-19	$T_{e,air,out}$
Temperature Differential	*[°C]*	12	12	12	12	12	12	$T_{e,air,in} - T_e$
Evaporator superheat	*[K]*	5	10	8	10	8	10	ΔT_{super}
Evaporation temperature at design condition	*[°C]*	14	14	-5	-5	-28	-28	T_e
Condenser								
Temperature Differential	*[K]*	12	12	12	12	12	12	$T_c - T_{c,air,in}$
Condenser subcooling	*[K]*	5	5	5	5	5	5	ΔT_{sub}
Heat exchanger effectiveness	*[-]*	0.5	0.5	0.5	0.5	0.5	0.5	η_c
Condensation temperature at design condition	*[°C]*	38	38	38	38	38	38	T_c
Internal heat exchanger								
Heat exchanger effectiveness	*[-]*	0.5	0.5	0.5	0.5	0.5	0.5	η_{IHE}
Compressor								
Isentropic efficiency compressor	*[-]*	0.7	0.7	0.7	0.7	0.7	0.7	η_i
Clearance volume ratio	*[-]*	0.03	0.03	0.03	0.03	0.03	0.03	CL
Running time at design conditions	*[%]*	80	80	80	80	80	80	$R_{p,design}$
General								
Minimum Thermal Load	*[W]*	1000	1000	200	1000	200	1000	Q_L
Maximum Thermal Load	*[W]*	20000	200000	40000	200000	25000	100000	Q_L
Default Thermal Load	*[W]*	2000	3000	2000	3000	1000	5000	Q_L
Indoor (room) temperature at design	*[°C]*	26	26	26	26	26	26	$T_{indoor,design}$

condition								
Ambient Temperature at design condition	*[°C]*	32	32	32	32	32	32	$T_{amb,design}$
Refrigerant characteristics								
Leakage at manufacturing	*[%]*	2%	2%	2%	2%	2%	2%	R_{mfg}
Annual leakage	*[%]*	2%	5%	2%	25%	2%	25%	R_{leak}
Recharge level	*[%]*	55%	55%	55%	55%	55%	55%	$R_{rechargelevel}$
Recovery fraction	*[%]*	0%	0%	0%	0%	0%	0%	$R_{recovery}$

Some specific remarks to the table above:
1. The evaporator inlet air temperature for AC systems is the actual indoor temperature at design conditions. This temperature is a function of the design ambient temperature. See the room model for more details here. For the commercial systems the air inlet temperature represents the air return temperature to the evaporator. E.g. for display cases this is the air temperature after leaving the space being cooled, so it is generally higher than the actual product temperatures being cooled or kept frozen.
2. The air temperature difference is the air temperature before the evaporator coil minus the air temperature after the coil. If the cooling load is higher this typically means that more air needs to be transported (larger fans). Using this difference and the specified air inlet temperature, directly the outlet temperature can be found.
3. The temperature differential is the difference between the air inlet and the evaporation temperatures. Here a typical value of 12 K is used for all systems, which can be classified as a reasonably well designed system. In case such differential is high then a larger evaporator size should be considered (at the given air temperature difference).
4. The evaporator superheat are typical values to allow a proper control of the expansion devices. For on site installed systems, larger values are applied as these are typically not so well controlled. Note that the superheat value may not be larger than the temperature differential given. In the model a formula has been implemented for off-design point calculations, which sets the superheat at the maximum achievable value (being the air inlet temperature minus the evaporation temperature) in case this value is lower than the given design superheat value.
5. For the condenser the temperature differential shown is defined as the difference between condensation temperature and air inlet. Here also a value of 12 K is used which is fairly typical.
6. The condenser heat exchanger effectiveness is set to 0.5 which means that the air passing the condenser heats up to 50% of the theoretically maximally possible value, so 6 K.
7. The condenser subcooling is set to 5 K which is a fairly typical value for all these systems.
8. The internal heat exchanger effectiveness is set to 0.5 for all systems.
9. For the compressor an isentropic efficiency of 0.7 is applied. This is a fairly high efficient compressor.
10. The clearance volume ratio expresses dead volume versus swept volume (stroke volume) and influences the refrigerant flow delivered especially at high pressure ratio's which may occur at high ambient temperatures.
11. The running time at design is set at 80% leaving some additional capacity in case the thermal load exceeds the design condition
12. The thermal load has a minimum, a maximum and a typical (default) value. The actual value for the calculation is an input in the main part of the MCII model.
13. The indoor (room) temperature is a calculated parameter and has been made a function of the ambient temperature (see the room model).
14. The ambient temperature at design is a country specific parameter and is included in this list above, as it is used for the calculation of some other parameters in the table
15. The refrigerant leak rate at manufacturing is set at a fixed rate of 2% of the initial charge.
16. The annual leakage of sealed, factory assembled systems is typically small and set to 2%,

whereas on site assembled systems generally have significantly higher leak rates.

17. The recharge level indicates the minimum amount of refrigerant for a system to function still properly, see the direct impact model for more details.
18. The recovery fraction at the end of life has been implemented in the model, but is generally set to 0% as it is assumed that for most article 5 countries the recovery level is still fairly small.

2.3 Type of refrigerants

To date the model contains the following refrigerants:

Refrigerant Name	Boiling Point [°C]	ODP [kg R-11e/ kg]	GWP [kg CO2 e/kg]	Liquid Thermal conductivity [W/(mK)] [1])	Viscosity [µPas] [1])	dTsat/ dP [K/ Pa] [1])	Main characteristics
HCFC-22	-40.8	0.034	1780	0.0857	173.7	3.89	Base line refrigerant to be replaced due to its ODP
HFC-134a	-26.1	0	1360	0.0833	207.4	5.59	HFC-134a is used in a variety of equipment including heat pumps and chillers. It is classed as an A1 refrigerant (lower toxicity, non-flammable). Energy efficiency is good, provided that pipes and heat exchangers are suitably sized.
Propane (R-290)	-42.1	0	5	0.0961	102.3	4.49	HC-290 is flammable and has thermodynamic properties similar to HCFC-22. It is the most frequently used hydrocarbon refrigerant in air-conditioning applications.
R-404A	-46.2	0	4200	0.0655	137.5	3.32	R-404A is used widely in commercial refrigeration systems, and is classified as A1 (lower toxicity, non-flammable). The efficiency is acceptable. A major advantage of R-404A is the low discharge temperature which makes it possible to have a high temperature lift in a single stage system.
R-407C	-43.6	0	1700	0.0865	164.2	3.74	R-407C is a mixture of the which has been used widely in air-conditioning, chiller and heat pump systems, especially to help the transition from HCFC-22. It is classed as A1 (lower toxicity, non-flammable). The efficiency is acceptable and better than of the R-404A it is normally used to replace. However, temperature glide and higher discharge temperature needs to be taken into account.
R-410A	-51.4	0	2100	0.0917	125.8	2.49	R-410A is used widely in air-conditioning, chiller and heat pump systems, and is classified as A1 (low toxicity, non-flammable). The pressure of R-410A is higher than HCFC-22 or R-404A. Generally the efficiency is equivalent to HCFC-22 or better, especially at lower temperatures. This efficiency however deteriorates at higher ambient temperatures.
Isobutane (R-600a)	-11.7	0	20	0.0911	159.4	10.79	HC-600a is a flammable low pressure refrigerant. Its main use is in domestic refrigeration systems and smaller commercial refrigeration applications.
HFC-32	-51.7	0	704	0.1297	120.3	2.42	HFC-32 is used as a component of refrigerant blends such as R-404A and R-410A. As a single refrigerant the pressure and capacity are around 1.5 times higher than HCFC-22 and slightly higher than R-410A. It is classed as A2L (low toxicity, lower flammability). The efficiency of HFC-32 systems are

Refrigerant Name	Boiling Point [°C]	ODP [kg R-11e/ kg]	GWP [kg CO2 e/kg]	Liquid Thermal conductivity [W/(mK)] [1])	Viscosity [μPas] [1])	dTsat/ dP [K/ Pa] [1])	Main characteristics
							higher than R-410A and the theoretical COP is a few per cent better than R-410A at typical air-conditioning conditions. The capacity is approximately slightly higher (~ 5%) but it can be easily accommodated with slight adjustment of the compressor displacement in new systems. Its system charge is lower than for R-410A. It has better heat transfer properties and transport properties than R-410A due to lower molar mass. Discharge temperatures are higher than R-410A. Higher polarity of HFC-32 compared to R-4010A makes necessary the use of new lubricant oils. Some system adaptations may be necessary for handling the discharge temperature of the compressor especially at high ambient temperatures.
HFC-1234yf	-29.4	0	<1	0.0651	163.7	5.74	HFC-1234yf is an unsaturated HFC (HFO) and can replace HFC-134a in the same systems since the pressure-temperature characteristics are almost identical. It is classified as A2L (low toxicity, lower flammability). In general this refrigerant produces efficiency levels comparable to HFC-134a although the theoretical COP is a few percent below that of HFC-134a.

[1]) at 20 °C liquid temperature, source: "Refprop 9, NIST Reference Fluid Thermodynamic and Transport Properties", National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, Maryland 20899, USA.

GWP and ODP data has been taken from the UNEP 2014 Report of the Refrigeration, Air-Conditioning and Heat Pumps Technical Options Committee. In this UNEP RTOC report the data is based on the most recent data from WMO 2014 which is on itself based on IPCC, 5th assessment with corrections based on revised data for lifetimes. Further, in the UNEP RTOC report data for refrigerant mixtures has been obtained by means of calculation.

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The selection of a refrigerant has two main aspects on the performance of a refrigeration or air-conditioning systems:

a) Differences in thermodynamic properties (temperature/pressure relation, enthalpy etc) may lead to a higher or lower efficient cycle. This often depends on the operating condition so refrigerant A may be more efficient at certain pressure levels than refrigerant B while the opposite may be the case at other pressure levels. These principle differences in thermodynamic properties are included in the MCII model.

b) Differences in transport properties (viscosity, conductivity etc) may impact the heat transfer inside the evaporator and condenser. This is a complex issue and handled in a simplified way inside the MCII model.

There are further numerous other factors which influence the performance of a system, such as lubricant/refrigerant interaction, pressure drops etc. Any conversion project should properly deal with these aspects. Such aspects are considered to be beyond the scope of the MCII model.

2.3.1 Heat transfer correction factor

When comparing refrigerants inside refrigeration and air-conditioning systems the heat transfer in the heat exchangers play a large role. Inside the MCII model, the heat transfer characteristics of the condenser and evaporator are calculated for the base case HCFC-22 leading to conductance values for these two heat exchangers (the system assumes certain temperature differentials between refrigerant and air inlet for the base case). This is described in more detail in the design calculation with HCFC-22 for the condenser and the evaporator. For other refrigerants it is then simply assumed that the conductance (UA) remains the same.

However, it is known that changing refrigerants has an impact on the heat transfer. Several parameters are of relevance here:

a) Transport properties such as viscosity and thermal conductivity. Specifically higher thermal conductivities of liquid lead to a higher heat transfer coefficient (HTC).
b) Thermodynamic properties: latent heat. Higher latent heats (or specific refrigerating effect) result in lower mass flows at the same cooling capacity. This results in lower mass fluxes (mass flow rate divided by cross sectional area) if the same tube diameters are maintained, resulting generally in lower HTC values.
c) Thermodynamic properties: density. Low densities lead to high required volume flows at the same cooling capacities, which is typically compensated for by increasing the compressor stroke volume. High volume flows and hence high velocities (if the same tube diameters are used) lead to higher pressure drops.
d) Thermodynamic properties: dTsat/dp. The impact of pressure drop needs to be viewed in terms of drop in saturation temperature. If the pressure drop for refrigerant A is twice as large as for refrigerant B, the effect on system efficiency may still be lower for refrigerant A compared to B in case the drop in saturation temperature is lower (hence if dTsat/dp is less than 50 % of the value of refrigerant B).
e) Refrigerant composition and azeotropic or zeotropic behaviour. For zeotropic refrigerants, there is a negative effect on the HTC due to mass transport phenomena in the refrigerant. E.g. during evaporation, the most volatile component of the mixture will preferentially boil off. This also result in a non-constant temperature during evaporation or condensation, the so called temperature glide. This temperature glide can be used to its advantage if the heat exchanger is designed in a proper counter flow arrangement. Azeotropic mixtures will behave similar to pure refrigerants.

Concluding, estimating heat transfer effects when the refrigerant is changed is a complex task requiring very detailed modeling, which is far beyond the scope of the MCII model. It is further known that negative aspects such as lower mass fluxes, higher pressure drops, temperature glides can partly be negated by changing tube diameters, parallel paths and air flow arrangement in the heat exchanger or in other words, by heat exchanger redesign without significant costs impact (this excludes simply heat exchanger enlargement).

To compensate for these complex phenomena the model contains a simple correction factor to be applied to the UA values calculated for HCFC-22:

$$(UA)_{e,X} = c(UA)_{e,R22}$$

$$(UA)_{c,X} = c(UA)_{c,R22}$$

As all systems have air-to-refrigerant heat exchangers at both evaporator and condenser side, the same correction factor is applied to both sides.

An aspect which can not be neglected is the zeotropic effect on the HTC. This is especially relevant to R-407C. For this refrigerant a small literature survey was carried out and 7 sources revealed a wide range in change of inside tube heat transfer coefficient and reductions from 15 to 70 % compared to R-22 were reported depending on the heat exchanger characteristics. From this an average value of 40 % was assumed. As the internal tube heat transfer makes up only a part of the total heat transfer resistance this value has to be weighted with the air side heat transfer. In the systems included in the MCII model, typically the air side has the largest resistance. Assuming that 75 % of the heat transfer resistance is related to the air side and 25 % to the refrigerant side, the impact of the 40 % on the refrigerant side HTC reduces to 10 % on the total heat transfer or on the conductance. This results in a correction coefficient of 0.9.

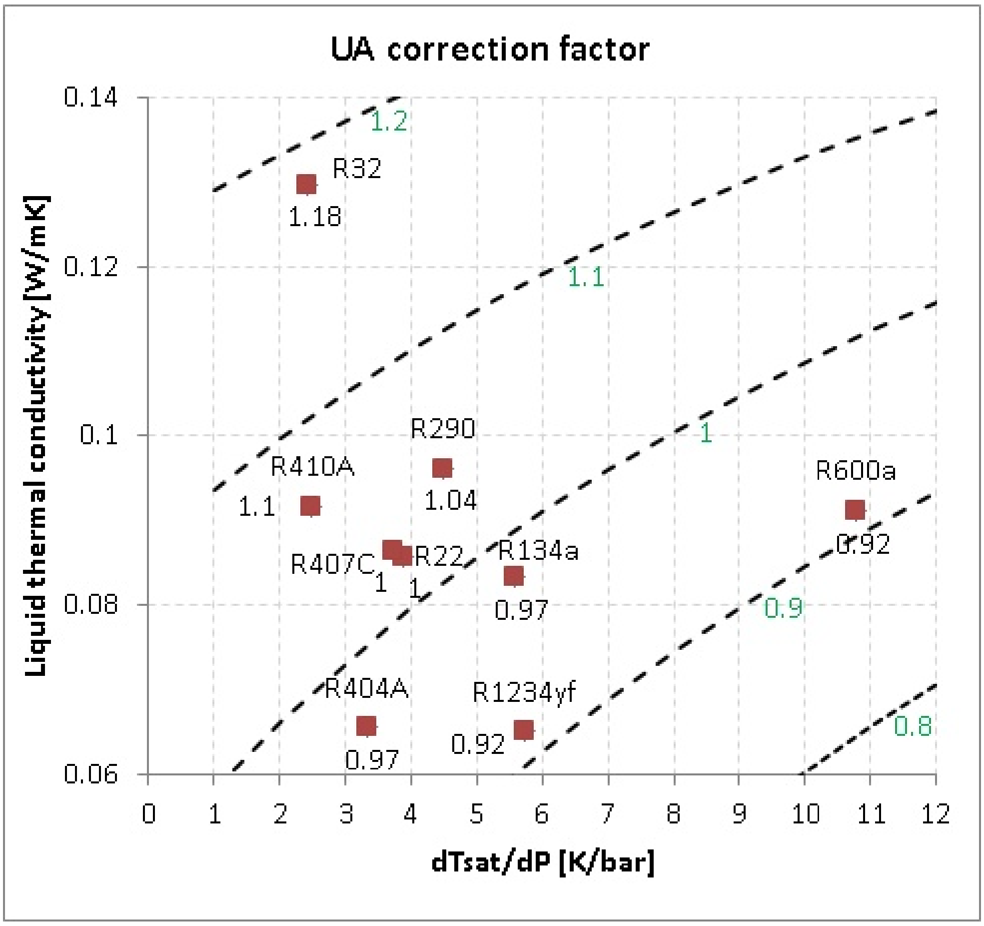
As mentioned, thermodynamic and transport properties do have an impact on the heat transfer. The two most important parameters are the liquid thermal conductivity and the saturation temperature dependency on pressure (dTsat/dp). From field studies it is very difficult to derive correction coefficients as these are widely influenced by the system being used for the conversion. A theoretical analysis presented by Domanski and Yashar was followed (Domanski, Yashar, "Comparable performance evaluation of HC and HFC refrigerants in an optimized system", 7th IIR Gustav Lorentzen Conference on Natural Working Fluids, Trondheim, Norway, May 28-31, 2006, further referred to as /Dom2006). This analysis proved useful as it made a comparison between ideal cycle calculations and optimized systems. The ideal cycle calculations are similar to the MCII calculations using purely thermodynamic relations. In the optimized systems, the heat exchangers were optimized for each refrigerant by changing tube circuitry. The latter has the consequence that excessive pressure drop is avoided (which would occur for low pressure refrigerants) by adding more parallel paths, but at the same type this has a negative impact on the inside tube heat transfer coefficients. Changes in compressor efficiency have been excluded by referring in each case to the same isentropic efficiency. The final results of the analysis include the COP of each system using a pure thermodynamic model and using the detailed model with optimized heat exchangers. The difference between these two results can be seen as the net effect of the heat exchange process only (on both cold and warm side simultaneously). This is then a combined result of liquid thermal conductivity, viscosity and pressure drop characteristics.

This difference in COP can also be simulated by the MCII tool using the heat transfer correction factor (or in other words, the correction factors can be tuned to get similar results as presented in / Dom2006/). This is not unrealistic as the heat exchangers studied are similar in nature to those of the applications included in the MCII tool. To investigate such possible correction factors, the MCII model was run at typical conditions for the /Dom2006/ study which was carried out for air-conditioning systems (system type was set to AC, Factory Assembled). The table below reports in column 2 the ratio between the COP of the alternative refrigerant and R-22 at the design condition if no correction factor for UA is applied (except for R-407C, which was set to 0.9). The third column presents the difference between optimized system calculations and ideal cycle calculations following /Dom2006/. If this difference is applied to the initial COP ratio's one can calculate target COP ratio's for the MCII model. Subsequently the heat transfer correction factors were adjusted until the COP ratio became close to the target COP ratio's (see the final COP ratio in column 6).

	COP ratio MCII model without UA correction factor	Change in efficiency compared to ideal cycle calculations following/ Dom2006/	Target COP Ratio's for MCII model	Heat transfer correction factor	Final COP ratio MCII model
	[-]	[%]	[-]	[-]	[-]
R-22	1.000			**1**	
R-134a	1.030	-3	1.000	**0.97**	1.009
R-290	1.010	3	1.040	**1.04**	1.037
R-404A	0.948			**0.97**	0.928
R-407C	0.915			**0.90**	0.915
R-410A	0.932	7.5	1.007	**1.10**	0.994
R-600a	1.069	-8.5	0.984	**0.92**	1.009
R-32	0.940	12	1.060	**1.18**	1.049
R-1234yf	1.018			**0.92**	0.960

As the study of /Dom2006/ did not include all alternative refrigerants included in the MCII model the following procedure has been used to estimate correction factors for the "missing refrigerants".

1. The heat transfer correction factors were plotted in a diagram (shown below) with dTsat/dp on the horizontal axis and the liquid thermal conductivity on the vertical axis. This shows a correlation with the highest correction factors in the left upper corner and the lowest in the right lower corner.
2. The data points for the known refrigerant where fitted to a cubic function (c1*dTsat/dp+c2*lambda +c3+c4*dTsat/dp*lambda). A support node was added to the bottom right corner to avoid strong extrapolation effects. The resulting fitting model is plotted in the graph by means of UA lines (the value given in a green font).
3. R-407C is very similar to R-22 with respect to pressure drop and liquid thermal conductivity, so the correction factor is left to 0.9 coming from the analysis based on the zeotropic refrigerant.
4. The correction factor for R-404A is estimated to be 0.97 as it has a lower dTsat/dp as R-22 and the dependency is quite strong (see e.g. the difference between R-410A and R-22), while the effect of lower liquid thermal conductivity is smaller.
5. The correction factor for R-1234yf is estimated to be 0.92 as it has a higher pressure drop effect and a lower liquid thermal conductivity compared to R-22.

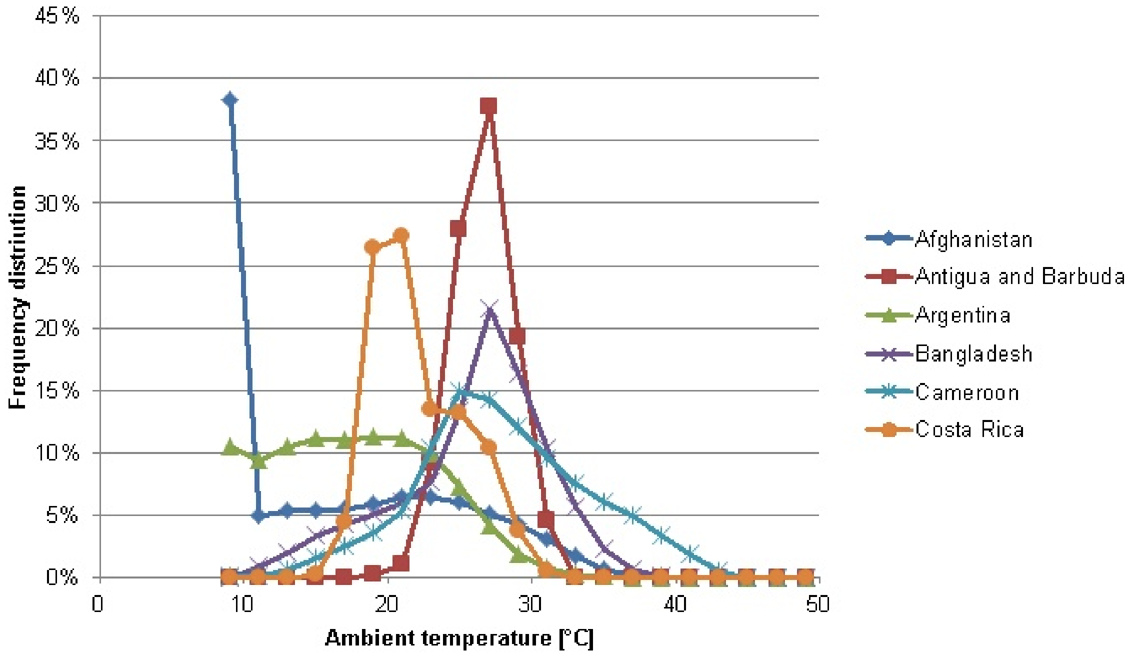


The resulting correction factors are included in the table above and inside the model as fixed parameters. This means that the values are applied for the various different system types and the different operating conditions. This is of course a large oversimplification of the reality. However, the system types are similar in nature (all air-to-air systems). Further, the use of these correction factors to other operating conditions, e.g. to the low temperature in freezers can still be realistic. Namely, the model includes all thermodynamic effects which may be quite different at the low temperatures. It is assumed that this is much less the case for the relative heat transfer figures so that the same correction on UA values is still reasonable also for other applications than air-conditioning systems.

Despite the rather crude approach used here, it is believed that the correction factors itself are based on physical factors and show a correct trend, namely that the higher pressure refrigerants used as alternative can compensate for their lower thermodynamic efficiencies by their better heat transfer characteristics. This therefore increases the accuracy of the MCII model. It must be noted however, that for any real system a more detailed study is needed as mentioned in the introduction.

2.4 Type of climate

The model contains climatic data for a large number of countries. The climatic data can be represented in a time, frequency chart for which an example for some countries is given below:



The temperature has been listed in increments of 2 K, this between 9 and 49°C ambient. In case a large fraction is plotted at 9°C, as is the case for Afghanistan, this means that during a large part of the year the ambient temperature is below this value (so it presents the aggregated values of all temperatures of 9°C and lower). The calculations of the systems do not run below this temperature. In praxis systems employing an outdoor condenser will not be able to drop the condensation temperature below a certain level, in order to prevent problems with operating the expansion devices and consequently the evaporator.

To obtain the climate data, the Secretariat has collected the frequency of occurrence of temperatures for a large number of countries. In case of countries with several climate zones, the occurrence has been calculated by weighting the different climate zones according to the population living in them, as a proxy to the number of refrigeration systems used.

2.5 Carbon intensity

The carbon intensity presents the amount of CO2 emitted for each kWh of electricity produced. This parameter may vary from country to country depending on the methodologies employed for generating electricity. Water powered, solar, wind or nuclear systems generate a low amount of CO2 whereas coal driven plants generally emit a large amount of CO2.

The emission of carbon dioxide are published for a number of Article 5 countries and have been estimated for the remainder according to information found in literature; however, for most countries with refrigeration manufacturing capacity, i.e. in the larger Article 5 countries, information has been published. In principle three sources have been used here where available. This data has been compared and an estimation of the reliability has been made. The resulting value is included in the

Countries worksheet of the MCII model.

2.6 Input data

The model contains a large number of preset parameters, as discussed under the type of systems.

The limited data needed is listed in the table below (note that all parameters which can be set by the user are given in a blue font, by using the tab key it is possible to jump to the next parameter). If a parameter has been changed which requires a recalculation, most calculated fields will show empty values and the button "Calculate" becomes active. Some of the parameters do not require a full calculation when changed, such as the number of units produced per year.

Parameter	Description	Symbol used in equations
Enterprise name	Textual description	
City	Location of the enterprise	
Country	Location of the enterprise, ambient temperature data and carbon intensity are a function of the country selected.	
Agency	Implementing agency	
Application	Selection between the different system types included in the model	
Model name or number	Textual description of the system investigated	
Number of units produced per year	Annual production figure used to calculate the total climate impact	n_{units}
Percentage exported [%]	Fraction of production exported to other countries, for this fraction the climate impact is calculated using a global average temperature distribution and carbon intensity	r_{export}
Refrigerant charge per unit [kg]	The amount of R-22 refrigerant added to the base line system at production or at field installation. Note that the charge needed for a unit is generally depending on the type of system, capacity (larger systems require larger charges) and the system design (e.g. size of liquid accumulators). The charge itself does not influence the system efficiency calculations, but it does impact the direct emissions.	$M_{initial}$
Product lifespan [y]	Average product lifetime from production or assembly until end of life.	Δt_{life}
Thermal load [W]	The heat load for which the system has been designed. Note that this thermal load will be matched with the refrigeration or AC system at 80 % running time ratio of the compressor in the design point.	Q_L
Alternatives to evaluate	List of refrigerants which can be compared with R-22. Clearing the "x" will exclude the selected refrigerant from the evaluation.	

2.7 Example cases

A typical example is included below:

Multilateral Fund Climate Impact Indicator (MCII)

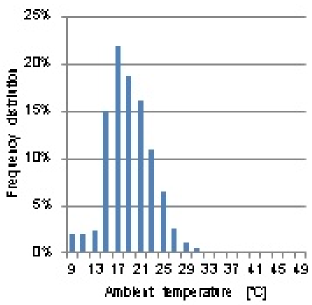
Version 3.4

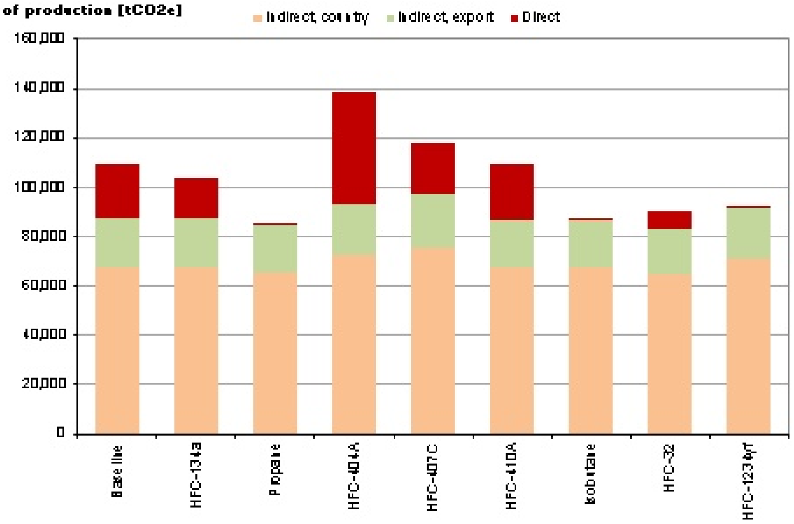
	Calculating	Done
General information	Elapsed Time [s]	45
Enterprise name		AB Refrigeration
City		Capital City
Country		Peru
Agency		UNIDO
Product information		
Application		Commercial Cooling, factory assembly
Model name or number		A1000
Number of units produced per year	[#/y]	10,000
Percentage exported	[%]	10%
Refrigerant charge per unit	[kg]	1.2
Product lifespan	[y]	10
Thermal load		
Minimum for this application type	[W]	200
Maximum for this application type	[W]	40,000
Thermal load (per unit)	[W]	2,000
Alternatives to evaluate		
HFC-134a		x
Propane		x
HFC-404A		x
HFC-407C		x
HFC-410A		x
Isobutane		x
HFC-32		x
HFC-1234yf		x

		Baseline	HFC-134a	Propane	HFC-404A	HFC-407C	HFC-410A	Isobutane	HFC-32	HFC-1234yf
Direct impact (over lifetime)										
ODS consumption (including service)	[t ODP]	0.67	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Climate impact of emissions	[t CO2e]	21,787	16,863	25	45,095	20,136	22,884	113	7,238	11
Indirect Impact, related to electricity production										
Country										
Design ambient temperature	[°C]	32								
Country carbon intensity	[kg CO2e/kWh]	0.251								
Electricity consumption, annual	[GWh/y]	27	27	26	29	30	27	27	26	28
Climate impact of lifetime emissions	[t CO2e]	67,927	67,586	65,331	72,404	75,679	67,281	67,477	64,499	70,869
Export										
Global design temperature	[°C]	32								
Global carbon intensity	[kg CO2e/kWh]	0.568								
Electricity consumption, annual	[GWh/y]	3	3	3	4	4	3	3	3	4
Climate impact of lifetime emissions	[t CO2e]	19,801	19,670	19,046	21,106	22,023	19,691	19,599	18,924	20,591
Total impact breakdown										
Change in direct impact	[t CO2e]		-4,924	-21,762	23,308	-1,651	1,097	-21,674	-14,549	-21,776
Change in indirect impact, country	[t CO2e]		-341	-2,596	4,477	7,752	-646	-450	-3,428	2,942
Change in indirect impact, global	[t CO2e]		-132	-756	1,305	2,222	-110	-202	-878	789
Total impact summary										
Total	[t CO2e]	109,516	104,119	84,402	138,605	117,838	109,857	87,189	90,661	91,471
Change	[t CO2e]		-5,397	-25,113	29,089	8,323	341	-22,326	-18,855	-18,044
Percent change	[%]		-5%	-23%	27%	8%	0%	-20%	-17%	-16%

Total lifetime climate impact of one year of production [tCO2e]

Enterprise name
AB Refrigeration
City
Capital City
Country
Peru
Model name or number
A1000





System efficiency effects can be seen in the chart by looking at the indirect bars (the combined values). The top bar represents the direct impact which is highly depending on the refrigerant selected as can be expected from the large differences in GWP. The relative magnitude of indirect versus direct is largely effect by the carbon intensity of the country, which is relatively low for Peru.

A second example is shown for on site air-conditioning systems of larger capacity in China, where the carbon intensity is higher.

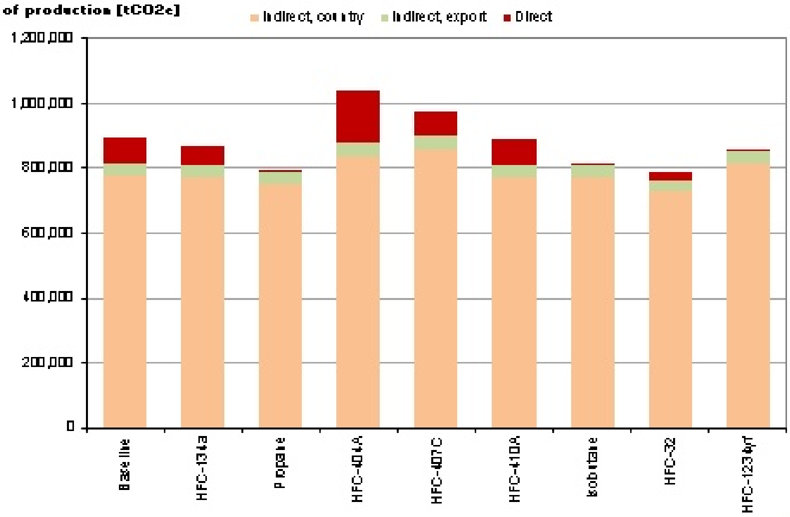
Multilateral Fund Climate Impact Indicator (MCII)

Version 3.4

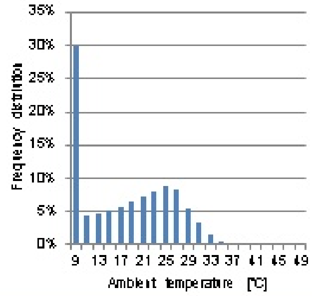
	Calculating	Done
General information	*Elapsed Time [s]*	45
Enterprise name		AB Refrigeration
City		Capital City
Country		China
Agency		UNIDO
Product information		
Application		AC, on site assembly
Model name or number		A1000
Number of units produced per year	*[#/y]*	10,000
Percentage exported	*[%]*	5%
Refrigerant charge per unit	*[kg]*	3
Product lifespan	*[y]*	12
Thermal load		
Minimum for this application type	*[W]*	1,000
Maximum for this application type	*[W]*	200,000
Thermal load (per unit)	*[W]*	15,000
Alternatives to evaluate		
HFC-134a		x
Propane		x
HFC-404A		x
HFC-407C		x
HFC-410A		x
Isobutane		x
HFC-32		x
HFC-1234yf		x

		Baseline	HFC-134a	Propane	HFC-404A	HFC-407C	HFC-410A	Isobutane	HFC-32	HFC-1234yf
Direct impact (over lifetime)										
ODS consumption (including service)	*[t ODP]*	2.43	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Climate impact of emissions	*[t CO2e]*	78,498	60,756	91	162,475	72,548	82,451	407	26,079	41
Indirect Impact, related to electricity production										
Country										
Design ambient temperature	*[°C]*	32								
Country carbon intensity	*[kg CO2e/kWh]*	0.827								
Electricity consumption, annual	*[GWh/y]*	78	78	76	85	87	78	78	73	82
Climate impact of lifetime emissions	*[t CO2e]*	778,505	773,681	750,832	839,402	860,275	772,779	774,208	728,954	814,593
Export										
Global design temperature	*[°C]*	32								
Global carbon intensity	*[kg CO2e/kWh]*	0.568								
Electricity consumption, annual	*[GWh/y]*	5	5	5	6	6	5	5	5	6
Climate impact of lifetime emissions	*[t CO2e]*	37,211	36,938	35,846	40,059	41,039	37,064	36,916	35,076	38,824
Total impact breakdown										
Change in direct impact	*[t CO2e]*		-17,742	-78,407	83,977	-5,950	3,953	-78,091	-52,419	-78,457
Change in indirect impact, country	*[t CO2e]*		-4,824	-27,673	60,897	81,770	-5,726	-4,296	-49,550	36,088
Change in indirect impact, global	*[t CO2e]*		-273	-1,365	2,849	3,828	-147	-295	-2,135	1,613
Total impact summary										
Total	*[t CO2e]*	894,214	871,375	786,769	1,041,936	973,862	892,293	811,531	790,109	853,458
Change	*[t CO2e]*		-22,839	-107,445	147,722	79,649	-1,920	-82,683	-104,104	-40,756
Percent change	*[%]*		-3%	-12%	17%	9%	0%	-9%	-12%	-5%

Total lifetime climate impact of one year of production [tCO2e]



Enterprise name
AB Refrigeration
City
Capital City
Country
China
Model name or number
A1000



3 Cycle model desciption

3.1 Introduction

Within the cycle model the refrigeration or air-conditioning system is calculated using various refrigerants and for various ambient conditions. These ambient conditions are taken from the climate data selected.

The cycle model contains two steps:
1. The calculation of the selected system using R-22 at the design conditions. As typical temperature differences of condenser and evaporators are predefined for the system, it is possible to calculate actual condenser and evaporator sizes and air flows through these heat exchangers.
2. The calculation of the same system with alternative refrigerants or at other ambient temperatures using the components defined in step 1. As the system may operate off-design, which means e.g. at a lower or higher ambient temperature several parameters may change. E.g. the air temperature entering the condenser will change with the ambient temperature if the condenser is located outdoor. The off-design calculations are performed for a range of ambient temperatures. This results in an actual cooling capacity and energy consumption for each condition. By multiplying the consumption with the number of hours in each temperature interval, it is possible to establish the total annual energy consumption of the system, which is discussed in the emission calculations

3.2 Design calculation with HCFC-22

To start a design calculation the following needs to be specified:
1. The selection of a refrigeration or air-conditioning system which automatically sets a large number of parameters
2. The design ambient temperature
3. The thermal load for which the system is designed (the amount of heat the cooling system must extract).

The design calculation follows the next structure, using as base refrigerant HCFC-22.

1. First the main refrigerant loop parameters are calculated: condensation and evaporation temperatures and outlet conditions of the evaporator as well as the condenser.
2. From the system cooling capacity, an evaporator analysis is carried out leading to the evaporator conductance used for further calculations at off-design conditions.
3. From the internal heat exchanger, the temperature at the exit of the suction line is determined and from this the temperature at the exit of the liquid line can also be determined (see the internal heat exchanger topic)
4. The refrigerant mass flow rate is determined
5. From the compression process the exit conditions at the compressor, which are equal to the inlet conditions of the condenser are derived.
6. Finally a condenser analysis can be made leading to the condenser conductance and the condenser air flow rate.
7. At the design condition the thermal load is specified. This can however be converted in a conductance value to allow later calculations at other conditions.

After the analysis of the R-22 system at design condition, the result is that evaporator and condenser sizes (not in terms of real dimensions but in the form of conductance's or UA values) are known as well as the air flows through evaporator and condenser. In addition also the compressor size needed for R-22 to match the thermal load supplied is calculated.

The evaporator and condenser info (UA and flow rate) is then applied to calculate the operation of the selected system with all alternative refrigerants or at other ambient temperatures.

3.2.1 Main circuit parameters

It is possible to derive the evaporation temperature directly from the air inlet temperature to the evaporator and the typical temperature differential:

$$T_e = T_{e,air,in} - \Delta T_e$$

From refrigerant saturation properties the evaporation pressure is subsequently calculated:

$$p_e = f_{sat}(T_e)$$

As the evaporator superheat is one of the parameters defined in the system selection (as a temperature differential), it is possible to calculate the evaporator exit temperature.

$$T_{e,out} = T_e + \Delta T_{super}$$

Using the pressure and the evaporator exit temperature the enthalpy is calculated using the appropriate refrigerant relation:

$$h_{e,out} = f(T_{e,out}, p_e)$$

For the condenser side, the condensation temperature can also directly be derived from the air temperature entering the condenser and the typical temperature differential given by the user:

$$T_c = T_{c,air,in} + \Delta T_c$$

The air temperature entering the condenser depends on the location of the condenser. If outdoor, then the ambient temperature at design condition is used. If indoor, the design temperature of the room is chosen.

Once the condensation temperature is calculated, the pressure can be derived from refrigerant saturation properties:

$$p_c = f_{sat}(T_c)$$

The condenser exit temperature can be found by subtracting the subcooling supplied by the system selection (as a differential temperature) from the condensation temperature:

$$T_{c,out} = T_c - \Delta T_{sub}$$

Using the appropriate refrigerant relations it is possible to calculate the condenser exit enthalpy:

$$h_{c,out} = f(T_{c,out}, p_c)$$

Knowing the conditions at evaporator and condenser exit, it is possible to calculate the internal heat exchanger performance, assuming an effectiveness of such heat exchanger:

$$T_{IHE,e,out} = T_{IHE,e,in} + \eta_{IHE}\left(T_{IHE,c,in} - T_{IHE,e,in}\right)$$

where the heat exchanger inlet at the low pressure side (e) is set equal to the evaporator exit and the heat exchanger inlet at the high pressure side (c) equal to the condenser exit.

From the heat exchange in this internal heat exchanger the outlet enthalpy at the high pressure side is calculated:

$$h_{IHE,c,out} = h_{IHE,c,in} - \left(h_{IHE,e,out} - h_{IHE,e,in}\right)$$

where again refrigerant property data has been used to calcuate enthalpies at heat exchanger inlet and outlets.

Assuming isenthalpic expansion in the throttling device in the circuit, the evaporator inlet enthalpy can now be set equal to the internal heat exchanger exit enthalpy at the high pressure side:

$$h_{e,in} = h_{IHE,c,out}$$

3.2.2 Evaporator

The cooling capacity of the system can be calculated from the thermal load given and the compressor run time:

$$Q_R = \frac{Q_L}{R_{p,design}}$$

For the evaporator air side, the temperature differential is specified during system selection (difference between air inlet and outlet). As the cooling capacity is known, it is possible to calculate the air mass flow rate (and hence also the air volumetric flow rate by dividing it with the density):

$$\dot{m}_{e,air} = \frac{Q_R}{c_{p,air}\left(T_{e,air,in} - T_{e,air,out}\right)}$$

$$\dot{V}_{e,air} = \frac{\dot{m}_{e,air}}{\rho_{air}}$$

As all temperatures are defined it is possible to calculate the logarithmic mean temperature difference for the evaporator:

$$LMTD_e = \frac{T_{e,air,in} - T_{e,air,out}}{\ln\left(\frac{T_{e,air,in} - T_e}{T_{e,air,out} - T_e}\right)}$$

which is used to calculate the evaporator conductance by:

$$(UA)_e = \frac{Q_r}{LMTD_e}$$

which means that the evaporator heat transfer characteristics at design conditions are fixed and can be used later for other temperature conditions or other refrigerants.

3.2.3 Refrigerant massflow

Knowing the cooling capacity of the system and the enthalpy difference over the evaporator, the refrigerant mass flow rate can be calculated from:

$$\dot{m}_r = \frac{Q_R}{h_{e,out} - h_{e,in}}$$

.

3.2.4 Compression process

To calculate the compression process, the isentropic efficiency is applied which is defined as:

$$\eta_i = \frac{h_{isentropic} - h_{comp,in}}{h_{comp,out} - h_{comp,in}}$$

This parameter can be seen as the work needed to compress the gas under constant entropy condition divided by the actual work and is defined by selecting the system.

By assuming the compressor inlet conditions to be equal to the exit conditions of the internal heat exchanger at the low pressure side:

$$h_{comp,in} = h_{IHE,e,out}$$

$$T_{comp,in} = T_{IHE,e,out}$$

the entropy at the inlet can be calculated:

$$s_{comp,in} = f(T_{comp,in}, p_e)$$

The isentropic end temperature and enthalpy can then be calculated from refrigerant property relations:

$$h_{isentropic} = f(s_{comp,in}, p_c)$$

$$T_{isentropic} = f(s_{comp,in}, p_c)$$

the compressor exit enthalpy is calculated using the formula for the isentropic efficiency listed above:

$$h_{comp,out} = h_{comp,in} + \frac{h_{isentropic} + h_{comp,in}}{\eta_i}$$

Finally the compressor input power can be calculated with:

$$P_{comp} = \dot{m}_r \left(h_{comp,out} + h_{comp,in} \right)$$

From the compressor volumetric relations it is possible to derive the compressor displacement volume needed to deliver the cooling capacity required under design conditions.

3.2.5 Condenser

For the warm side (the condenser) it is now possible to perform the heat transfer calculations. First it is assumed that the air entering the condenser coil is at the design condition discussed earlier. As the condensation temperature is known and the heat exchanger effectiveness is supplied by the system selected, it is possible to calculate the air exit temperature:

$$T_{c,air,out} = \eta_c \left(T_c - T_{c,air,in} \right) + T_{c,air,in}$$

Knowing all temperatures the logarithmic temperature difference can be calculated:

$$LMTD_c = \frac{T_{c,air,out} - T_{c,air,in}}{\ln\left(\frac{T_c - T_{c,air,in}}{T_c - T_{c,air,out}} \right)}$$

Here it is neglected that there are also non-isotherm parts in the condenser (the superheat and the subcooling). In a well designed system, these parts should be relatively small.

The condenser reject heat can be calculated as the refrigerant mass flow rate has already been established and the refrigerant state points at inlet and exit of the condenser are already known from the previous analysis:

$$Q_c = \dot{m}_r \left(h_{c,in} - h_{c,out} \right)$$

Knowing the condenser heat flow, it is possible to calculate the condenser conductance:

$$(UA)_c = \frac{Q_c}{LMTD_c}$$

It is then further possible to resolve the condenser air mass and volume air flow rate from:

$$\dot{m}_{c,air} = \frac{Q_c}{c_{p,air}\left(T_{c,air,out} - T_{c,air,in} \right)}$$

$$\dot{V}_{c,air} = \frac{\dot{m}_{c,air}}{\rho_{air}}$$

3.2.6 Thermal load

At the design condition, the thermal load for the system being studied is given as an input variable.

If the application is a commercial cooling or freezing unit, the thermal load origins from the heat flow through the walls and door of such a unit, which is a function of the temperature difference between the room and the air circulating in the product.

The heat conductance for a unit can thus be calculated at the design condition from:

$$(UA)_L = \frac{Q_L}{T_{indoor,design} - T_{e,air,in}}$$

For an air-conditioning application, the thermal load origins from the heat flow through the room walls itself and is a function of the temperature difference between the ambient (outdoor) and the room (indoor). However, even if the ambient and room temperature are equal, a load for the AC unit remains, due to internal heating in the room, presence of humans etc. If the ambient temperature drops below the indoor temperature a heat flow from indoor to outdoor would occur. The following formula represents this behaviour:

$$Q_L = (UA)_L \left(T_{amb} - T_{indoor}\right) + Q_{Base,AC}$$

The base load has a relation to the design load (as larger rooms will have a larger base load). It is however not a constant fraction, e.g. if the design ambient temperature is high, the base load should become a smaller fraction of the design heat load. To accommodate for this effect, a base heat load model has been implemented which takes a fraction of the design heat load using a base temperature condition (which is set in the model to 24°C, this means that if the design ambient temperature would be 24°C, the base load would become equal to the design load):

$$Q_{Base,AC} = Q_L \left(1 - 0.045(T_{amb,design} - T_{Base,AC})\right)$$

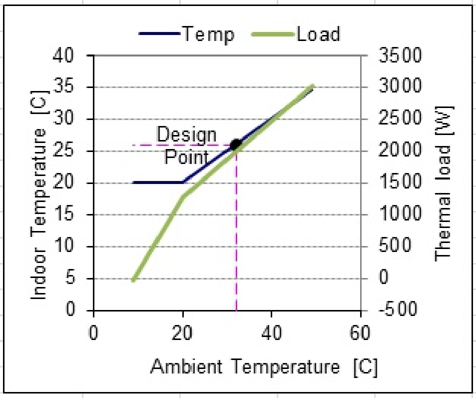
The factor 0.045 has been assigned to get a reasonable distribution between the constant heat load and the variable load coming from the ambient.

Assuming the base load above, a conductance of the room can be calculated from:

$$(UA)_L = \frac{Q_L - Q_{Base,AC}}{T_{amb,design} - T_{indoor,design}}$$

All together this generates an indoor temperature and a thermal load for an AC system as a function of the ambient temperature. An example is shown below (where the design base load was 2000 W at 32°C). The indoor temperature profile is generated from a model coupling the indoor temperature with the outdoor temperature.

Indoor temperature profile (for all systems)		
Minimum indoor temperature	*[°C]*	20
Indoor temperature increase / ambient temp. increase	*[-]*	0.5
Indoor temperature at design point	*[°C]*	26
Specific load calculation for AC systems		
Base load temperature	*[°C]*	24
Base thermal load	*[W]*	1280
UA for the room	*[W/K]*	120
Specific load calculation for other systems		
Air inlet temperature	*[°C]*	
UA for the application	*[W/K]*	
Isentropic efficiency compressor (default)	*[-]*	0.7



3.3 System calculations

Once the system has been selected and the calculation of the refrigeration system in the design point has been completed, it is possible to calculate the refrigeration cycle at other conditions or with other refrigerants. From the design point the air flow and thermal conductance (UA) of both the evaporator and condenser have been derived and are assumed to be the same in other operating conditions.

Other parameters, such as superheat, subcooling and isentropic compressor efficiency are all supposed to remain constant when the operating conditions of the system changes.

With this given set of data an iterative calculation of the system is needed. This is due to the fact that only the air entrance temperatures are given for both the condenser and evaporator, but the condensation temperature and evaporation temperature are unknown. If values are assumed for these parameters, the model equations can be evaluated, resulting finally in a revision of the evaporator and condenser temperature. This is repeated until convergence is achieved. The flow of (non-linear) equations is further explained in the iteration process, which is implemented in the Cycle sheets of the Excel workbook.

The equations which are used for all components in the system are described in separate chapters:
1. Compressor
2. Condenser
3. Evaporator
4. Internal heat exchanger
5. Room
6. Thermal load

There are some special situations in the system calculations:

1. If the compressor run time exceeds 100%, in general the system will not maintain the product temperature any more (e.g. the cooling unit will start to increase in temperature). In the model this is in principle not compensated for, so it is assumed that the compressor runs 100% at the same condition as when the thermal load would have been met. The program contains an internal (hidden) parameter which allows to perform the calculation in such a way that the product temperature will start to increase when the compressor has reached full load (since it is in general not preferred to calculate such situations, this option is not made available to the user).
2. At very low ambient temperatures the condensation temperature may drop below the evaporation temperature (e.g. for the cooling application). This is prevented by setting a minimum temperature differential between condenser and evaporator (in praxis this can be arranged with pressure regulators). As a result the running time of the compressor will remain constant at varying ambient temperatures in some cases.

3.3.1 Compressor

The compressor mass flow rate can be calculated as follows:

$$\dot{m}_r = \rho_{comp,in}\eta_v\phi_v \quad \text{[comp1]}$$

where the compressor volumetric efficiency is defined as follows (using the clearance volume ratio *CL*)

$$\eta_v = 1 - CL\left\|\left[\left(\frac{p_c}{p_e}\right)^{1/k} - 1\right]\right\| \quad \text{[comp2]}$$

and the compressor displacement volume is typically found as the product of the compressor swept volume and the operating frequency (rotational speed). In the model the compressor displacement volume flow rate Φ_v is used rather than swept volume in order to make systems independent of the operating frequency (which is for fixed speed compressors often linked to the frequency of the mains supply).

The compressor outlet conditions can typically be found using the isentropic efficiency given by the selection of the system:

$$\eta_i = \frac{h_{isentropic} - h_{comp,in}}{h_{comp,out} - h_{comp,in}} \quad \text{[comp3]}$$

if the inlet enthalpy to the compressor is known. The isentropic enthalpy is typically found using the appropriate refrigerant property relations.

The compressor input power can then be written as:

$$P_{comp} = \dot{m}(h_{comp,out} - h_{comp,in}) \quad \text{[comp 4]}$$

A total system efficiency can be expressed as a Coefficient of Performance:

$$COP = \frac{Q_R}{P_{comp}} \quad \text{[comp 5]}$$

The compressor run time can be calculated using the thermal load and the cooling capacity:

$$R_p = \frac{Q_L}{Q_R} \quad \text{[comp 6]}$$

3.3.2 Condenser

Basically three heat transfer relations are relevant for the condenser, for the air side, refrigerant side and the heat transfer between air and refrigerant, respectively:

$$Q = \dot{m}_{c,air}c_{p,air}\left(T_{c,air,out} - T_{c,air,in}\right) \quad \text{[cond1]}$$
$$Q = \dot{m}_r\left(h_{c,in} - h_{c,out}\right) \quad \text{[cond2]}$$
$$Q = (UA)_c LMTD_c \quad \text{[cond3]}$$

which must result in the same heat transfer in a stationary situation.

In this relation the logarithmic mean temperature difference is defined as:

$$LMTD_c = \frac{T_{c,air,out} - T_{c,air,in}}{\ln\left(\dfrac{T_c - T_{c,air,in}}{T_c - T_{c,air,out}}\right)} \quad \text{[cond 4]}$$

To evaluate the heat transfer for a coil type of heat exchanger, it is possible to use the classical Number of Transfer Units approach.This requires first the definition of the heat exchanger effectiveness:

$$\eta_c = \frac{T_{c,air,out} - T_{c,air,in}}{T_c - T_{c,air,in}} \quad \text{[cond 5]}$$

The number of transfer units is defined as the ratio of the conductance and the flow capacity:

$$NTU_c = \frac{(UA)_c}{\dot{m}_{c,air} c_{p,air}} \quad \text{[cond 6]}$$

Assuming a counter flow heat exchanger, it is now possible to relate the number of transfer units and the heat exchanger effectiveness with

$$\eta_c = 1 - e^{-NTU_c} \quad \text{[cond 7]}$$

Note that the above only holds for the single fluid refrigerants. For the mixed refrigerants using a glide, an extended model for the heat transfer effectiveness is integrated.

Here the temperature at the refrigerant side is not constant any more but follows a profile, corresponding with the temperature glide (the subcooled and superheated parts of the condenser are neglected). The logarithmic temperature difference formula can then be replaced with:

$$LMTD_c = \frac{\left(T_{c,sv} - T_{c,air,in}\right) - \left(T_{c,sl} - T_{c,air,out}\right)}{\ln\left(\dfrac{T_{c,sv} - T_{c,air,in}}{T_{c,sl} - T_{c,air,out}}\right)} \quad \text{[cond 4']}$$

The number of heat transfer units formula required here is a more complicated form of cond6 as this formula requires the minimum flow capacity following:

$$NTU_c = \frac{(UA)_c}{\min(c_{air}, c_{refr})} \quad \text{[cond 6']}$$

where the flow capacity formula's are defined as:

$$c_{air} = \dot{m}_{c,air} c_{p,air}$$

$$c_{refr} = \dot{m}_r \frac{\left(h_{c,sv} - h_{c,sl}\right)}{T_{c,sv} - T_{c,sl}}$$

For the condenser heat exchanger, counter flow is assumed for which the heat exchanger effectiveness is defined as:

$$\eta_c = \frac{1 - e^{-NTU_c(1-r)}}{1 - re^{-NTU_c(1-r)}} \quad \text{[cond 7']}$$

where r is defined as the ratio of the flow capacities:

$$if \quad c_{air} > c_{refr} \quad then$$
$$r = \frac{c_{refr}}{c_{air}}$$
$$else$$
$$r = \frac{c_{air}}{c_{refr}}$$

3.3.3 Evaporator

Basically three heat transfer relations are relevant for the evaporator, for the air side, refrigerant side and the heat transfer between air and refrigerant, respectively:

$$Q_R = \dot{m}_{e,air} c_{p,air} \left(T_{e,air,in} - T_{e,air,out} \right) \quad [\text{evap1}]$$
$$Q_R = \dot{m}_r \left(h_{e,out} - h_{e,in} \right) \quad [\text{evap2}]$$
$$Q_R = (UA)_e LMTD_e \quad [\text{evap3}]$$

which must result in the same heat transfer in a stationary situation.

In this relation the logarithmic mean temperature difference is defined as:

$$LMTD_e = \frac{T_{e,air,in} - T_{e,air,out}}{\ln\left(\dfrac{T_{e,air,in} - T_e}{T_{e,air,out} - T_e} \right)} \quad [\text{evap 4}]$$

To evaluate the heat transfer for a coil type of heat exchanger, it is possible to use the classical Number of Transfer Units approach.This requires first the definition of the heat exchanger temperature effectiveness:

$$\eta_e = \frac{T_{e,air,in} - T_{e,air,out}}{T_{e,air,in} - T_e} \quad [\text{evap 5}]$$

It is possible to express the number of transfer units as the ratio of the conductance and the flow capacity:

$$NTU_e = \frac{(UA)_e}{\dot{m}_{e,air} c_{p,air}} \quad [\text{evap6}]$$

Assuming a counter flow heat exchanger, it is now possible to relate the number of transfer units and the heat exchanger effectiveness with

$$\eta_e = 1 - e^{-NTU_e} \quad [\text{evap7}]$$

Note that the above only holds for the single fluid refrigerants. For the mixed refrigerants using a glide, an extended model for the heat transfer effectiveness is integrated.

Here the temperature at the refrigerant side is not constant any more but follows a profile, corresponding with the temperature glide (the superheated part of the evaporator is neglected). The logarithmic temperature difference formula can then be replaced with:

$$LMTD_e = \frac{\left(T_{e,air,in} - T_{e,sv}\right) - \left(T_{e,air,out} - T_{e,in}\right)}{\ln\left(\frac{T_{e,air,in} - T_{e,sv}}{T_{e,air,out} - T_{e,in}}\right)} \quad \text{[evap 4']}$$

as the inlet of the evaporator is typically somewhere between saturated liquid and saturated vapour its temperature needs a more precise estimation. This is handled by using a linear relation between temperature and enthalpy:

$$T_{e,in} = T_{e,sl} + \frac{h_{e,in} - h_{e,sl}}{h_{e,sv} - h_{e,sl}}\left(T_{e,sv} - T_{e,sl}\right) \quad \text{[evap 4']}$$

The number of heat transfer units formula required here is a more complicated from of evap6 as this formula requires the minimum flow capacity following:

$$NTU_e = \frac{(UA)_e}{\min(c_{air}, c_{refr})} \quad \text{[evap 6']}$$

where the flow capacity formula's are defined as:

$$c_{air} = \dot{m}_{e,air} c_{e,air}$$

$$c_{refr} = \dot{m}_r \frac{\left(h_{e,sv} - h_{e,sl}\right)}{T_{e,sv} - T_{e,sl}}$$

For the evaporator heat exchanger, counter flow is assumed for which the heat exchanger effectiveness is defined as:

$$\eta_e = \frac{1 - e^{-NTU_e(1-r)}}{1 - re^{-NTU_e(1-r)}} \quad \text{[evap 7']}$$

where r is defined as the ratio of the flow capacities:

$$if \quad c_{air} > c_{refr} \quad then$$

$$r = \frac{c_{refr}}{c_{air}}$$

$$else$$

$$r = \frac{c_{air}}{c_{refr}}$$

3.3.4 Internal Heat Exchanger

Many systems contain a heat exchanger between the suction line (after the evaporator) and the liquid line (after the condenser). An internal heat transfer between these two parts of the cycle may increase the COP of the system. The reason is that due to the internal heat exchanger the liquid cools down further so that, after the expansion process, more refrigerant in liquid form enters the evaporator (resulting in a lower refrigerant quality which is defined as the ratio between the vapour mass flow rate and the total refrigerant mass flow rate). The negative impact of the internal heat exchanger is that the suction gas will heat further up before arriving at the compressor which has a negative impact on the compressor efficiency.

To calculate the cycle including such heat exchanger the effectiveness of such heat exchanger must be supplied. This is defined as:

$$\eta_{IHE} = \frac{T_{IHE,e,out} - T_{IHE,e,in}}{T_{IHE,c,in} - T_{IHE,e,in}} \quad \text{[ihe 1]}$$

Further it is assumed that the heat released in the liquid line enters completely into the suction line. This can be expressed by the following enthalpy relation:

$$h_{IHE,e,out} - h_{IHE,e,in} = h_{IHE,c,in} - h_{IHE,c,out} \quad \text{[ihe 2]}$$

where the necessary enthalpies can be obtained from the temperatures using the refrigerant property relations.

In general these equations must be coupled to the condenser, evaporator and compressor equation, e.g. the outlet of the heat exchanger at the suction side is the inlet of the compressor.

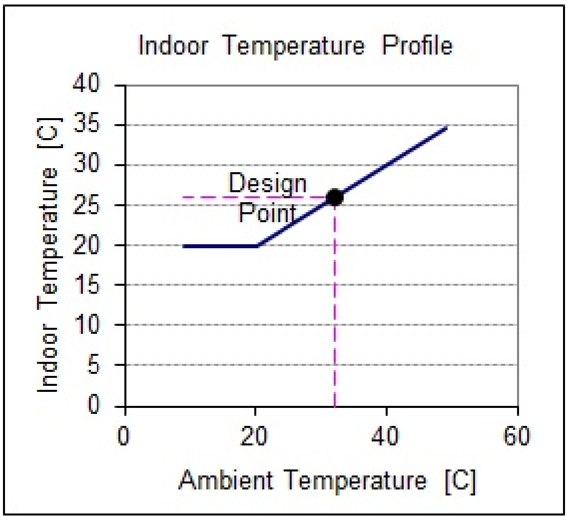
3.3.5 Room

The room (indoor) temperature often depends on the ambient temperature. The ambient temperature at design is a generic input parameter to the model.

To handle the link between indoor and ambient temperature, the model includes a calculation of the indoor design temperature where:

$$\begin{aligned}
&if \quad T_{amb,design} < T_{indoor,\mathrm{minimum}} \quad then \\
&T_{indoor,design} = T_{indoor,\mathrm{minimum}} \\
&else \\
&T_{indoor,design} = (T_{amb,design} - T_{indoor,\mathrm{minimum}})r + T_{indoor,\mathrm{minimum}} \quad \text{[room1]}
\end{aligned}$$

which means that if the ambient temperature is below a given minimum indoor temperature (e.g. 20 °C], then the indoor temperature is equal to this minimum value. Otherwise, the indoor follows the ambient temperature with a certain fraction *r* (if *r*=1 it increases just as much as the ambient temperature increase, with *r*=0 the indoor temperature stays constant). An example is shown below for *r*=0.5. This value is also used as the default value in the model.



3.3.6 Thermal load

The thermal load for a commercial cooling or refrigeration unit is calculated from:

$$Q_L = (UA)_L\left(T_{indoor} - T_{e,air,in}\right) \quad \text{[therm 1]}$$

for an AC unit the thermal load is expressed as:

$$Q_L = (UA)_L\left(T_{amb} - T_{indoor}\right) + Q_{Base,AC} \quad \text{[therm 2]}$$

3.3.7 Iteration process

The flow of the equations is described in the following:

1. The air inlet temperatures for both evaporator and condenser are taken from the system selection or from the calculated indoor temperature for air-conditioning systems. The air mass flow rates and the conductance's are obtained from the calculation at the design condition for R-22. Note that conductance's are scaled with a correction factor depending on the refrigerant.
2. The ambient temperature is given from the climate data.
3. The room (indoor) temperature follows the ambient temperature given a certain profile [room1].
4. The thermal load is calculated using the conductance value derived from the R-22 calculation [therm1] or [therm2]
5. The condensation temperature is assumed to an initial value of 20 K above the air inlet temperature, but is in each next iteration step calculated from the condenser heat exchanger effectiveness formula [cond5].
6. The evaporation temperature is assumed to be equal to the value in the R-22 design calculation as an initial value, but is in the next steps calculated from the evaporator heat exchanger effectiveness formula [evap5].
7. The evaporator air outlet temperature is calculated using the heat transfer relation for the air side [evap1].
8. The evaporator logarithmic mean temperature difference can be calculated from all know temperatures [evap4]
9. The number of transfer units for the evaporator is calculated using the conductance and the air mass flow rate [evap6] and from this the heat exchanger effectiveness [evap7].
10. The condenser exit temperature is calculated from the condensation temperature and the given subcooling;
    $$T_{c,out} = T_c - \Delta T_{sub}$$
11. From this temperature and the refrigerant properties, the enthalpy at the outlet is derived.
12. The evaporator exit temperature is calculated using the evaporation temperature and the given superheat:
    $$T_{e,out} = T_e + \Delta T_{super}$$
13. and from the refrigerant properties the enthalpy at evaporator exit is found.
14. From the internal heat exchanger formula [ihe1] the exit temperature at the low pressure side is obtained.
15. Using appropriate refrigerant relations and the heat flow balance [ihe2] the enthalpy at the exit of the internal heat exchanger at the high pressure side is obtained.
16. Assuming an adiabatic expansion process the enthalpy at evaporator inlet can be assumed to be the same as the enthalpy at internal heat exchanger exit:
    $$h_{e,in} = h_{IHE,c,out}$$
17. The refrigerant mass flow rate now follows from the heat transfer relation of the refrigerant side in the evaporator [evap2].
18. The refrigerant inlet temperature for the compressor is assumed to be equal to the internal heat

exchanger exit temperature at the low pressure side:

$$T_{comp,in} = T_{IHE,e,out}$$

19. and from refrigerant properties the enthalpy and entropy at the compressor inlet are calculated.
20. The isentropic compressor exit entropy is by definition equal to the inlet entropy.
21. From this isentropic exit entropy and the condensation temperature the isentropic enthalpy is calculated using refrigerant properties.
22. From the isentropic efficiency given by the user in the parameter list, the real compressor exit enthalpy is calculated [comp3]
23. From the condensation and evaporation pressures (calculated from their respective saturation temperatures) the volumetric efficiency is calculated [comp2]
24. From the compressor mass flow rate equation, the required volumetric flow rate is calculated [comp1].
25. From the enthalpy difference over the compressor and the mass flow rate the compressor power is calculated [comp4]
26. Using the compressor power and the total cooling capacity the COP is calculated [comp5]
27. The number of transfer units for the condenser is calculated using the conductance and the air mass flow rate [cond6] and from this the heat exchanger effectiveness [cond7]
28. The condenser inlet enthalpy is taken from the compressor outlet enthalpy.
29. The heat rejected from the condenser follows from the heat transfer relation of the refrigerant side [cond2]
30. The air outlet temperature of the condenser is calculated using the heat transfer relation for the air side [cond1]
31. The compressor run time is calculated using the thermal load and the cooling capacity [comp6]

4 Emission model

4.1 Introduction

The emission model calculates the emission of CO2 (or CO2 equivalents) for the selected refrigeration or AC system over its lifetime. In principle this emission constitutes of two parts:

1. Direct emissions, which results from refrigerant emissions during manufacturing of the systems, operation and end of life. Of direct importance is here the initial refrigerant charge. This is a user input parameter which depends highly on the type of system, capacity (larger systems require larger charges) and the system design (e.g. size of liquid accumulators). The charge to be given is that for the base line R-22 systems. Charges for alternative refrigerants are calculated based on liquid density differences.
2. Indirect emissions, which results from the electricity generation needed to drive the system over its lifetime. To calculate these the cycle model is needed for estimating the system power input at various ambient temperatures. Climatic data is then subsequently needed to supply an ambient temperature distribution over the year.

The totals of these emissions are expressed in metric tonnes of CO2 equivalent and present the climate impact of the systems being investigated.

The carbon dioxide emissions of a system depends on the size, quality of the components, quality of design, application and the operating conditions (chiefly the ambient temperature), and, finally, the CO2 emission related to the production of electricity. In order to take the different factors into account, a number of assumptions were made and procedures were developed:

1. It is assumed that the principle quality of components and quality of the design remain constant; reflecting the content of decision 61/44 of the Executive Committee, asking the Secretariat to “maintain the established practice when evaluating component upgrades in HCFC conversion projects for the refrigeration and air-conditioning sectors, such that after conversion the defining characteristics of the components would remain largely unchanged or, when no similar component was available, would only be improved to the extent necessary to allow the conversion to take place […]” ;
2. The parameters entered as input values are also assumed to remain constant; in particular the capacity of the system, the application and whether a unit is factory assembled or assembled in the field, as well as the country and the share of export;
3. The load of the system is estimated depending on the design load = capacity of the unit, and an estimated deviation for different temperatures. A more detailed description can be found in the thermal load model.
4. The energy efficiency varies, depending on the refrigerant used, for different outdoor temperatures; two refrigerants having the same energy efficiency at one outdoor temperature and otherwise identical operating conditions will show a difference in energy consumption at other conditions. The climate data is an important factor here.
5. The emission of carbon dioxide during electricity production are published for a number of Article 5 countries and have been estimated for the remainder according to information found in literature;

4.2 Direct impact

The direct emissions of HCFCs and alternatives take into account a large number of factors related to the lifetime of each unit manufactured, and aims to use general assumptions to quantify them. This quantification is carried out for the lifetime of the equipment and relates to:

1. The HCFC charge, being an input value, and the potentially different charge of the alternatives.
2. An emission at the time of manufacturing for systems assembled and charged in a factory.
3. Typical annual emissions for an average unit, depending on the type of refrigeration or air-conditioning equipment and on assembly in a factory or on site.
4. An average lifetime for each unit depending on the various types of refrigeration and air-conditioning

equipment as well as on assembly in a factory or on site.
5. Recovery at the end of life, currently, in line with practices typical for Article 5 countries assumed to be zero.
6. The climate impact of the substance, calculated on the basis of the substances Greenhouse Warming Potential (GWP) for a 100-year time horizon.

A quantification of system specific parameters is given in the discussion of the type of systems.

To calculate the direct impact of a refrigeration or air-conditioning system, the emission of refrigerants over its entire lifetime needs to be estimated. This can be done using the following formula (expressed as a fraction of the initial charge):

$$R_{life} = 1 + R_{rechargelife} + R_{mfg} - R_{recovery}$$

which takes into account the initial charge of the unit (set to 1), the charge added over its lifespan during servicing, the refrigerant lost during manufacturing as well as the amount of refrigerant recovered at the end of life (all given as a fraction of the initial charge).

The recharged fraction over its lifetime can be calculated from:

$$R_{rechargelife} = n_{recharge} R_{recharge}$$

which is a multiplication of the number of recharge events over lifetime and the ratio of the initial refrigerant actually recharged. As most systems contains a form of refrigerant accumulator which allows some leakage before the system stops functioning properly a recharge level has been introduced (as a fraction of the initial charge). Using this level and the annual leak fraction it can be calculated what the time interval is between recharge events:

$$\Delta t_{recharge} = \frac{1 - R_{rechargelevel}}{R_{leak}}$$

E.g. if there is an annual leak rate of 2% and the level can safely reduce to 80% then the system needs to be recharged every 10 year.

To calculate how many recharges are made during the life span of the unit, the following formula is applied:

$$n_{recharge} = floor\left(\frac{\Delta t_{life} - 1}{\Delta t_{recharge}}\right)$$

E.g. if the product lifetime is 10 years and the recharge interval time is 5 years, the number of recharges is 1 (the floor function is a round down to the nearest integer here). The minus 1 year construction avoids that the system would just be charged before the end of its life. It can also be that there is no recharge at all during lifetime.

The amount of refrigerant added during each recharge event can now be calculated from the annual leakage and the time between recharge events:

$$R_{recharge} = R_{leak} \Delta t_{recharge}$$

If annual service would be present and the system would be topped up to the original charge level each time, then the amount recharged would be equal to the leak percentage. However, with the above construction of calculating the number of recharge events a more realistic estimation of the amount charged each time, and hence over the lifetime, is found.

To calculate how much of the initial charge is recovered at the end of life, then an estimation is needed of the charge fraction present at the end of life (eol) and the percentage of the refrigerant which is typically recovered:

$$R_{recovery} = R_{eol} R_{recovery,rate}$$

The refrigerant ratio present at end of life does depend on how much refrigerant has been added and the leaked refrigerant, both over the entire lifetime:

$$R_{eol} = 1 + R_{recharge,life} - R_{leak,life}$$

where the charge added over lifetime has been discussed before and the leak over the lifetime is simply:

$$R_{leak,life} = R_{leak} \Delta t_{life}$$

To calculate the total charge over life time the initial charge needs to be multiplied with the charge ratio used over lifetime and multiplied with the number of units produced per year.

$$M_{life} = R_{life} M_{initial} n_{units}$$

The direct climate impact is then:

$$CI_{life} = \frac{M_{life} GWP_r}{1000} \quad [\text{t CO2e}]$$

If the Greenhouse Warming Potential of the refrigerant is expressed in kg CO2 equivalent per kg refrigerant then the climate impact is obtained in metric tonnes of CO2 equivalence.

In the model, the calculation of the refrigerant over lifetime is only made for R-22, assuming that all leakages, etc, remain the same for all refrigerants. The only correction made is for the difference in liquid density between the alternative refrigerant and R-22:

$$M_{life,r} = M_{life,R22} \frac{\rho_r}{\rho_{R22}}$$

E.g. for hydrocarbons the liquid density is less than half the value of R-22 and consequently the charge used over lifetime reduces. To obtain the climate impact of alternative refrigerants the GWP of the respective refrigerant is applied.

4.3 Indirect impact

Indirect CO2 emissions result from the electricity generation needed to drive the system over its lifetime. To calculate the indirect emissions the following steps are needed:

1. The temperature distribution over the year must be known
2. The cycle needs to be calculated for the refrigerant of interest and for each of the ambient temperatures occurring at the location of the system
3. The system energy consumption needs to be integrated over all the ambient temperatures and the hours these occur during the year
4. The total yearly energy consumption needs to be multiplied with the typical CO2 emission per kWh electricity use (the country carbon intensity) in order to get the total equivalent CO2 emission.

The model contains a dataset of temperature distributions for a large range of countries. These are divided in temperature bins (intervals) ranging from 9 to 49 °C with a width of 2 K. For each of these temperature bins the model calculates the performance of the cycle, resulting in a compressor run time and an input power. The actual annual energy consumption for all units operating in the country of interest and for each temperature bin can be found by:

$$E_{T,country} = h_T R_p P_{comp} n_{units} \left(1 - r_{export}\right) / 1000 \quad \text{[kWh]}$$

where the hours per year at temperature *T* can be found from:

$$h_T = f_T \cdot 365 \cdot 24 \quad \text{[h]}$$

using the ratio of hours at temperature T per year.
Internally in the cycle model it is possible that for each temperature bin a different thermal load occurs. To compensate for this load the compressor will have to run a certain part of its time (the running time ratio) during which the power is calculated. The product of these two gives the average power used at the temperature T.

To obtain the total energy consumption the energy consumption per temperature bin needs to be integrated over all temperature bins and multiplied by the life time in years. To avoid large numbers, the results is expressed in GWh:

$$E_{life,country} = \Delta t_{life} \sum_{T=\min}^{T=\max} E_T \cdot 10^{-6} \quad \text{[GWh]}$$

The model contains also the option that units manufactured in a country are actually exported and used elsewhere. The basic energy consumption for the fraction of units exported is:

$$E_{T,export} = h_T R_p P_{comp} n_{units} r_{export} / 1000 \quad \text{[kWh]}$$

For the temperature distribution, a global temperature distribution is used as it would be too complicated to track where this fraction of units is operated.

5 Model implementation

5.1 Workbook structure

The model is entirely developed as a spreadsheet tool, which is able to calculate refrigeration and AC system performances under a variety of ambient conditions and compare the results with HCFC-22 base cases. This comparison does include both energy consumption as well as the related CO2 emissions for which regional data is included in the model.

The spreadsheet model is structured as follows:

1. The **MCII** sheet, which contains the user input data (such as refrigeration system to be studied, climatic zone, country of application, etc.). Also the main output data is shown here, such as annual energy consumption and CO2 emission for HCFC-22 and all the alternatives included. The results are shown in tabular format and can be printed as a single sheet.
2. A **Details** sheet which contains some of the main results calculated. It shows the system performance at the design point as well as a diagram of system efficiencies and compressor run time over the various ambient temperatures.
3. A set of **Cycle_x** ("x" representing the name of the refrigerant) sheets containing the refrigeration cycle calculations, based on ideal loop calculations extended with isentropic efficiencies of the compression process. The cycle calculations are automatically performed for all relevant ambient temperatures (using a bin approach with temperature intervals).
4. A set of **x**-sheets ("x" representing the name of the refrigerant), each containing refrigerant property data in tabular form.
5. A **Settings** sheet which contains predefined data for the refrigeration/AC systems which can be studied.
6. A **Countries** data sheet which contain temperature/time information for a large number of countries, as well as carbon intensity data. In addition, for each country a design temperature is included.
7. A **WorkArea** sheet which is used for some background calculations and preparation for graphs.

The spreadsheet model further contains some code modules (using VBA), which is used for the necessary user interfacing.

To solve an iteration in Excel the iterative calculation method has been activated which allows circular dependencies between cells. Special care has been taken to start up this process. The iterative procedure in Excel has been accompanied by a VBA macro which handles the iteration process and checks progress in the convergence. Further the procedures have been optimised to reduce calculation speed, e.g. by making sure that initial values are reasonably guessed or are based on a previous calculation (e.g. when switching from one temperature bin to the next one).

5.2 Refrigerant calculation

The cycle model contains worksheets with a full set of thermodynamic tabular data for each refrigerant and uses Excel VBA routines to interpolate between the values and to calculate on this basis a refrigeration cycle. The same as the data sheet could also be obtained by using a property data subroutine (available, inter alia, from NIST in Gaithersburg); however, such routines cannot be disseminated freely and complicates distribution of the model, therefore the use of the data sheets and interpolation methods. A secondary benefit of this approach is that the calculation runs significantly faster. The thermodynamic tabular data is based itself on Refprop version 9.0 from NIST.

An example of the thermodynamic tabular data is shown in next figure which shows a part of the data collected for R-407C:

Refrigerant	R407c.mix	Based on Refprop 9.0						
						Enthalpy Table [J/kg]		
						dT superheat [K]		
	Tsat [K]	psat [Pa]	Hsl [J/kg]	Tbubble [	Rhosl [kg/	0	10	
	220.0	41929	128224	**212.7**	1410	379953	387266	394
	226.7	61131	136981	**219.5**	1389	383972	391489	399
	233.4	86837	145804	**226.4**	1369	387938	395674	403
	240.1	120498	154703	**233.2**	1348	391836	399805	407
	246.8	163718	163689	**240.0**	1326	395651	403870	412
	253.5	218246	172773	**246.9**	1304	399368	407855	416
	260.3	285966	181966	**253.7**	1282	402971	411744	420
	267.0	368896	191284	**260.6**	1258	406441	415524	424
	273.7	469180	200740	**267.4**	1234	409759	419177	428
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Inside the cycle worksheets the calculation of a model takes place and refrigerant property functions are called which access these thermodynamic data table. The property functions are basically lookup and interpolation functions. The functions available are listed below:

```
' All parameters transfer in SI
' Temperature [K]
' Pressure [Pa]
' Enthalpy [J/kg]
' Entropy [J/kgK]
'
' Dew point pressure as a function of saturation temperature
'-----------------------------------------------------------------------
Function I_Pdew_T(Refr As String, Tsat As Double) As Variant

' Saturated liquid enthalpy as a function of saturation temperature (dew point temperature  !)
'-----------------------------------------------------------------------
Function I_Hsl_Tsat(Refr As String, Tsat As Double) As Variant

' Saturated liquid temperature as a function of saturated liquid enthalpy
'-----------------------------------------------------------------------
Function I_Tbubble_Hsl(Refr As String, Hsl As Double) As Variant

' Bubble point temperature as a function of saturation temperature given at the dewpoint
'-----------------------------------------------------------------------
Function I_Tbubble_Tsat(Refr As String, Tsat As Double) As Variant

' calculation of gas superheat as a function of saturation temperature and entropy
' -----------------------------------------------------------------------
Function I_dT_Tsat_s(Refr As String, Tsat As Double, s As Double) As Variant

' calculation of a certain property (can be enthalpy, etc) as a function of gas superheat and saturation temperature
' -----------------------------------------------------------------------
Function VapourProp_Tsat_dT(Refr As String, Prop As String, Tsat As Double, Dt As Double) As Variant
```