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EXECUTIVE COMMITTEE OF
THE MULTILATERAL FUND FOR THE
IMPLEMENTATION OF THE MONTREAL PROTOCOL
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**REPORT ON THE MULTILATERAL FUND CLIMATE IMPACT INDICATOR
(DECISION 69/23)**

Background

1. The document on revised analysis of relevant cost considerations surrounding financing of HCFC phase-out¹ included a section on environmental issues and an approach for evaluating climate-relevant emissions during the life cycle of a product containing HCFCs. In its discussions, the Executive Committee considered to further analyse if the approach outlined in the document provided a satisfactory and transparent basis for the prioritization of HCFC phase-out technologies to minimize other impacts on the environment, including on the climate as envisaged in decision XIX/6. The Executive Committee requested the Secretariat to continue with its evaluation in order to report in a more detailed fashion at a subsequent meeting (decision 55/43(g)).

2. At its 57th meeting, the Executive Committee considered the document on prioritization of HCFC phase-out technologies to minimize other impacts on the environment², which contained a status report on the further analysis of a functional unit approach for the prioritization of HCFC phase-out technologies to minimize other impacts on the environment. The Executive Committee was informed³ *inter alia* that the Secretariat, with the support of experts, was undertaking a technical consultation to enable the development of functional units for the refrigeration and foam sectors. Further to a discussion, the Executive Committee requested the Secretariat to prepare a document presenting four examples of the application of the methodology to the 58th meeting (decision 57/33(b)).

3. In response to decision 57/33(b), at its 59th meeting the Executive Committee considered the document on prioritization of HCFC phase-out technologies to minimize other impacts on the environment⁴. In this document, the Secretariat introduced the term “Multilateral Fund Climate Impact Indicator” (MCII) for a simplified concept of what was originally functional units. Further to a

¹ UNEP/OzL.Pro/ExCom/55/47.

² UNEP/OzL.Pro/ExCom/57/59.

³ Paragraphs 170 to 173 of document UNEP/OzL.Pro/ExCom/57/69.

⁴ UNEP/OzL.Pro/ExCom/59/51 and Add.1.

discussion⁵, the Executive Committee *inter alia* requested the Secretariat to demonstrate the application of the MCII to a sample of project submissions from the 60th meeting onwards; to finalize the development of the MCII, and to make the underlying data, the methodology used, and preliminary working models of the software available on the intranet to bilateral and implementing agencies and to members of the Executive Committee (decision 59/45(c), (d) and (e)).

4. Subsequently, the MCII has been further developed focussing on the refrigeration and air-conditioning manufacturing sector; it has been made available to Executive Committee members and implementing agencies; and the Secretariat has reported to several Executive Committee meetings on the progress made and experience gained with the MCII⁶.

5. The MCII has also been used in the calculation of the climate impact associated with the conversion of refrigeration and air-conditioning manufacturing enterprises associated with stage I of approved HPMPs and the results had been included in relevant project evaluation document submitted to the Executive Committee⁷. The experience gained in applying the MCII had shown that the potential climate impact of the conversion of refrigeration and air-conditioning equipment from the baseline (HCFC-22) to alternative refrigerants can be measured objectively and systematically, and enables the Executive Committee to monitor and account for the potential climate impact of the projects supported by the Multilateral Fund.

6. At the 69th meeting, the Executive Committee discussed the report on the MCII⁸ submitted by the Secretariat and, *inter alia*, requested the Secretariat to provide a progress report to the last meeting in 2014, and the results of an independent review to be undertaken (decision 69/23(b)).

Actions taken to optimize the MCII tool

7. Based on feedback from and discussions with Executive Committee members⁹ and implementing agencies, the Secretariat undertook additional work¹⁰ that concluded with a fully-developed MCII tool that incorporated the following additional features:

- (a) A simplified user interface has been developed including the format for inputting the minimum required information; a simplified output format with values and graphics that facilitate the analysis of the results; and a disclosure of the weather conditions for each country;
- (b) A user-friendly instructions manual comprising: a succinct explanation of the MCII tool, the basic design parameters, the outcomes and how they should be interpreted; simplified instructions on how to use the MCII, including the minimum information required; and a detailed explanation of all the assumptions, principles, databases and algorithms used in the model. The instruction manual is contained in Annex I to the present report;

⁵ Paragraphs 219 to 227 of document UNEP/OzL.Pro/ExCom/59/59.

⁶ As reported in the following documents: UNEP/OzL.Pro/ExCom/59/51 and Add.1, UNEP/OzL.Pro/ExCom/62/56 and Add.1, UNEP/OzL.Pro/ExCom/63/58, UNEP/OzL.Pro/ExCom/64/50, UNEP/OzL.Pro/ExCom/65/54, UNEP/OzL.Pro/ExCom/66/52, UNEP/OzL.Pro/ExCom/67/34, UNEP/OzL.Pro/ExCom/69/34, UNEP/OzL.Pro/ExCom/72/43.

⁷ The MCII has been applied in the following projects (numbers in parenthesis refer to pre-session documents UNEP/OzL.Pro/ExCom/...): Algeria HPMP (66/26); Argentina room and unitary air-conditioning equipment manufacturing sector project (61/28); Bahrain HPMP (68/22); Bosnia HPMP (66/30); China HPMP - industrial and commercial refrigeration and air-conditioning sector (64/29); China HPMP - room air-conditioner manufacturing sector (64/29); Indonesia HPMP (64/34); Iran (Islamic Republic of) HPMP (63/35); Jordan Phase-out of HCFC-22 and HCFC-141b from the manufacture of unitary air-conditioning equipment at Petra Engineering Industries Co. (60/31); Lebanon HPMP (64/37); Nigeria HPMP (62/43); Serbia HPMP (62/47); Thailand HPMP (68/41).

⁸ UNEP/OzL.Pro/ExCom/69/34.

⁹ In response to a letter sent by the Secretariat inviting Executive Committee members on the use of the MCII for inclusion in a document to be presented to the 73rd meeting, the Government of Australia submitted its comments on 19 June 2014. The Secretariat notes with appreciation the comments received which had been taken into account.

¹⁰ The Secretariat sought expert advice to modify the background programming of the MCII tool.

- (c) Inclusion of two additional refrigerants, namely HFC-32 and HFO-1234yf, in the list of refrigerants already available within the MCII. Several other alternative refrigerants were considered, however they are not yet standardized as indicated by a refrigerant number issued by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), or were not related to the applications covered by the MCII. New alternative refrigerants may be incorporated into the MCII in the future as needed; and
- (d) Inclusion of transport properties of refrigerants to take into account differences between technologies in heat exchange and pressure loss, using a heat transfer correction coefficient based on the specific characteristics of each refrigerant¹¹.

8. In response to decision 69/23(c), where the Secretariat was requested to undertake an independent review of the MCII, the Secretariat had sent the fully-developed MCII to three qualified experts. The report by the experts will be included in an addendum to the present document prior to the 73rd meeting. Suggestions provided by the experts to continue improving the MCII may also be added to the Secretariat's recommendation.

The MCII and its purpose

9. The MCII is a tool that:

- (a) Provides an indication of the effect on the climate associated with the conversion of enterprises manufacturing air-conditioning, commercial cooling and commercial freezing equipment from HCFC-22 (baseline) to alternative refrigerants¹²;
- (b) Operates on the basis of the following parameters which are included in the project proposals on the conversion of equipment manufacturing enterprises submitted to the Executive Committee: country¹³, type of equipment¹⁴, number of units produced per year, refrigerant charge per unit, product lifetime (years); cooling capacity per unit (W); and number of units exported¹⁵;
- (c) Takes into account the emissions of refrigerant during manufacturing, operation and at the end of life (direct emissions), and the emissions of greenhouse gases occurring as a result of energy consumption of refrigeration and air-conditioning equipment based on HCFC-22 (baseline) and alternative refrigerants (indirect emissions);
- (d) Calculates for a refrigeration or air-conditioning unit the CO₂ equivalent emissions over its lifetime as the sum of direct and indirect emissions, and multiplies the result with the amount of units produced in one year. This interim result represents the climate impact of the annual production for a given technology over the lifetime of the equipment produced; and

¹¹ All correction factors are set to 1 except for HFC-407C, which is set to 0.9. A detailed explanation is presented in section 2.3.1 of the manual attached as Annex I to the present document.

¹² These refrigerants are: R-290 (propane), R-600a (isobutane), HFC-134a, HFC-32, HFC-404A, HFC-407C, HFC-410A, and HFO-1234yf.

¹³ The MCII contains climate and carbon intensity data for a large number of countries. Carbon intensity refers to the amount of CO₂ emitted for each kWh of electricity produced. Calculation of climate impact of a refrigeration unit operating in a specified country will take into consideration the specific frequency of occurrence of temperatures under which the equipment will be operating and the carbon intensity of the energy produced in the country where the equipment will be operating.

¹⁴ The MCII deals with air-conditioning equipment, commercial cooling equipment and commercial freezing equipment. A difference is made between factory-assembled systems and on-site assembled systems (e.g., a condensing unit put on top of a cold store and an evaporator/fan assembly placed inside the cold store).

¹⁵ The climate impact of equipment exported is calculated based on average values on climate and carbon intensity data of all countries in the MCII.

- (e) Compares the climate impact of the refrigeration or air-conditioning unit selected using several technologies including the baseline technology (HCFC). The ratio between the baseline (HCFC) and the alternative is used to obtain a qualitative comparison of different alternatives (percentage values). Negative values for the MCII denote a reduction in the climate impact as compared to the baseline, positive values denote an increase. The description of the emission model provides more detail.

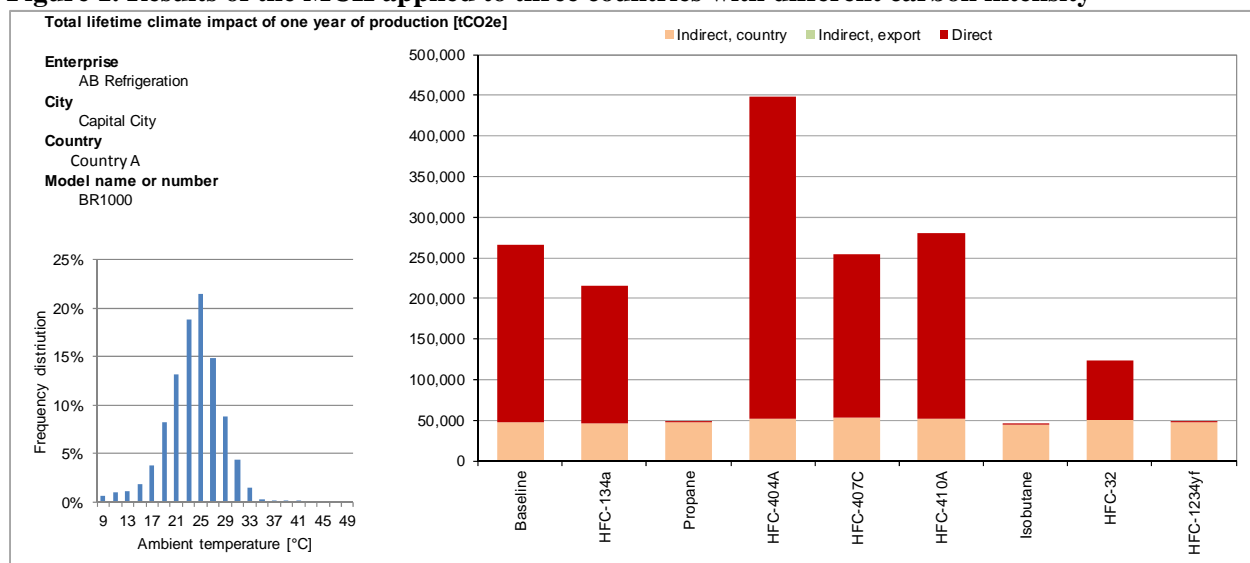
10. However, the objective of the MCII is only to offer an indication of the climate impact prior to any conversion activities based on limited data, not to replace any analysis that could be undertaken on the basis of more detailed information on the performance of specific refrigeration and air-conditioning equipment, such as a life-cycle climate performance or a life-cycle analysis.

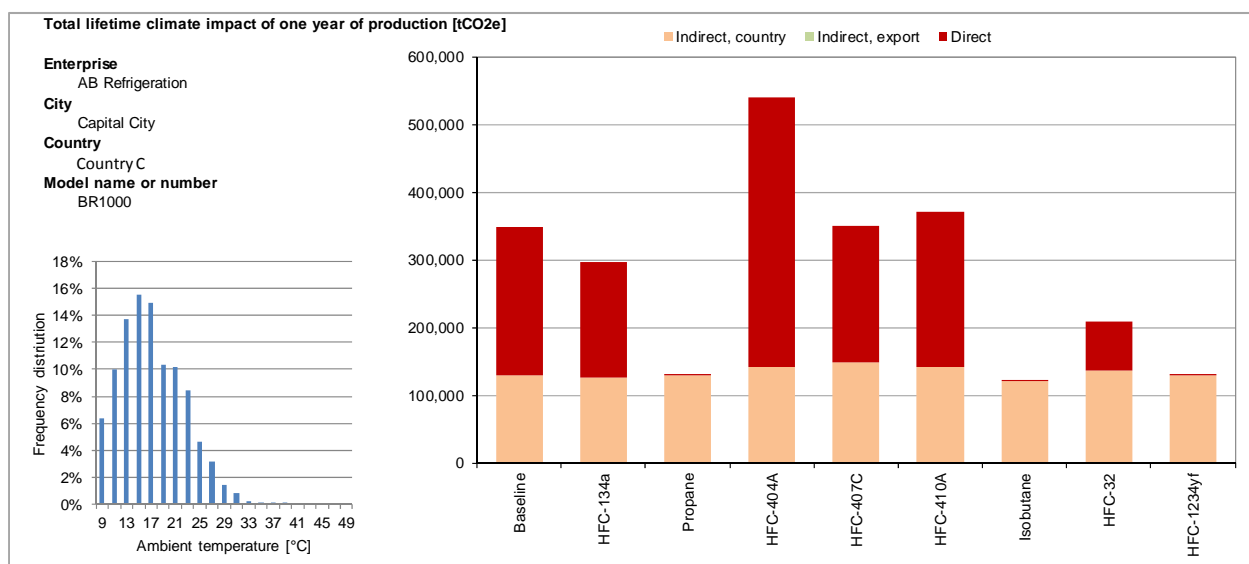
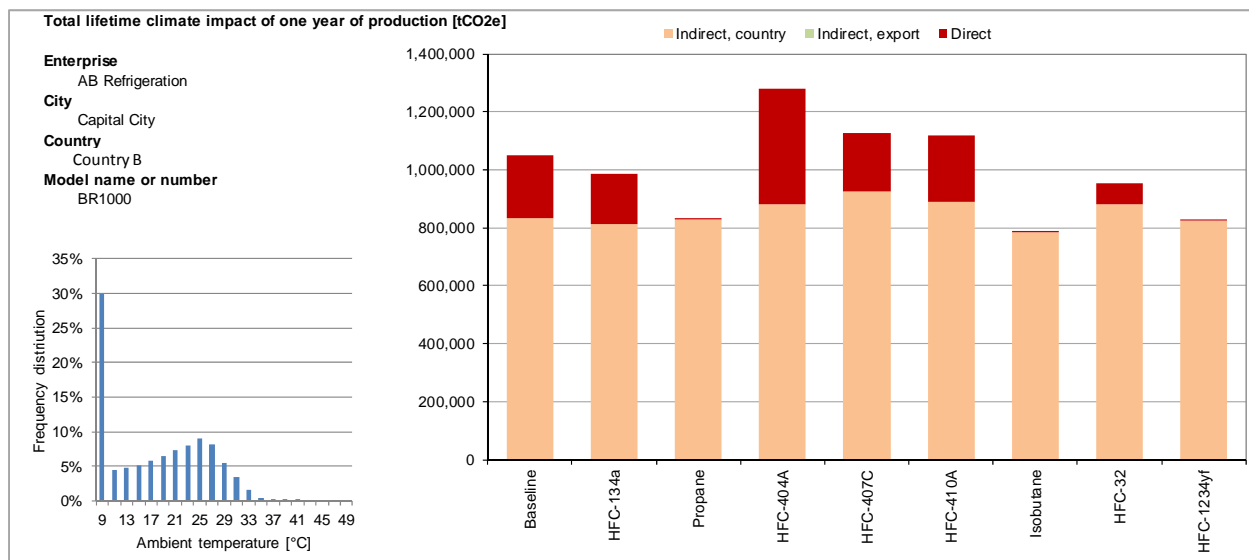
11. To undertake its calculations, the MCII contains an internal model that calculates the energy consumption of the system based on first principles for the thermodynamic circuit. It effectively calculates cycles based on average system characteristics, such as expected compressor efficiencies and heat exchanger performances. The performance of alternative refrigerants is then estimated based on first principles based on the thermodynamic differences with HCFC-22. The model assumes that the alternative refrigerants have no impact on compressor efficiency and heat exchanger performance, which in reality may not be the case as these components may be, or may need to be, optimized for the alternative selected.

Example of the use of the fully-developed MCII

12. In order to illustrate the type of information provided by the fully-developed MCII, the tool was applied for 100,000 units of air-conditioning equipment, with a cooling capacity of 2,000 W and a product lifetime of 10 years, manufactured in three countries with different energy sources, namely: Country A (emitting 0.057 kg of CO₂ for each kWh of electricity produced), Country B (emitting 0.827 kg of CO₂ for each kWh of electricity produced) and Country C (emitting 0.478 kg of CO₂ for each kWh of electricity produced). The results are provided in Figure 1.

Figure 1. Results of the MCII applied to three countries with different carbon intensity





13. The following observations are of relevance:

- (a) In Country A the weight on potential climate impact of direct emission is larger than that of indirect emissions due to its carbon intensity (0.057 kg of CO₂ emitted for each kWh of electricity produced). Baseline emissions of 266,145 tonnes CO₂-equivalent are 83 per cent direct and 17 per cent indirect. By selecting a low global warming potential (GWP) refrigerant (propane, isobutane or HFC-1234yf), total CO₂-equivalent emissions could be reduced by more than 80 per cent;
- (b) In Country B, given a larger carbon intensity associated to the source of energy (0.827 kg of CO₂ emitted for each kWh of electricity produced), indirect emissions are a larger portion of the total emissions. Baseline emissions of 1,051,341 tonnes CO₂-equivalent are four times larger than in Country A. In this case, direct emissions represent only 21 per cent of the total emissions. By selecting a low-GWP technology, total CO₂-equivalent emissions could be reduced by 25 per cent in the best-case scenario; and

- (c) In Country C, with a carbon intensity associated to the source of energy larger than that in Country A but lower than that in Country B (0.478 kg of CO₂ emitted for each kWh of electricity produced), results in baseline emissions of 347,934 tonnes CO₂-equivalent, of which 63 per cent are direct emissions. By introducing low-GWP technologies, total CO₂-equivalent emissions could be reduced by 65 per cent in the best-case scenario.

Calculation of the climate impact for other manufacturing sectors

14. After a thorough assessment of the methodologies currently used to calculate the climate impact for other manufacturing sectors, and the feasibility of integrating them into the MCII, the Secretariat concluded that those calculations should remain unchanged, as described in document UNEP/OzL.Pro/ExCom/69/34:

- (a) For the foam sector, the most straightforward approach is to continue calculating the climate impact based on a comparison of the amount of HCFC used for foam blowing, the amounts of alternative blowing agents used after conversion, and the GWP of both substances. This approach has so been used in all climate impact calculations presented to the Executive Committee as part of HPMP submissions;
- (b) For the aerosol and solvent sectors and, where applicable, in the process agent sector, it is assumed that all substances are emitted into the atmosphere in the same year in which the solvent or process agent has been produced or imported. Calculations of climate impact will continue to be done on the basis the amounts of solvents used before (HCFCs) and after conversion, and the GWP of both substances; and
- (c) For the fire-fighting sector the release of fire-fighting agents is delayed as compared to initial installation. Currently, no pattern is available to clearly classify the applications and allow for emission analysis. Therefore, the calculation of the climate impact will be done based on an assessment of amounts used, alternative technologies selected and use pattern.

Conclusions

15. The Secretariat will continue applying the MCII model to calculate the impact on the climate of investment projects in the refrigeration and air-conditioning manufacturing sectors. The calculation on the impact to the climate of investment projects in other manufacturing sectors will continue to be done based on the methodologies described in paragraph 14 above.

16. There is no possibility for integrating the MCII to the multi-year agreement database given the fact that operating properties of the two applications are different.

17. The MCII Microsoft is available on the Secretariat's website.

Recommendation

18. The Executive Committee may wish:

- (a) To note the Secretariat's report on the fully-developed Multilateral Fund Climate Impact Indicator (MCII) (decision 69/23) contained in document UNEP/OzL.Pro/ExCom/73/54; and

- (b) To note that the Secretariat will continue to calculate the impact on the climate of investment projects in the refrigeration and air-conditioning manufacturing sectors applying the MCII model, and for investment projects in all other manufacturing sectors to apply the methodologies described in paragraph 14 of document UNEP/OzL.Pro/ExCom/73/54.

Annex I

MCII Model, Refrigeration and AC systems

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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).

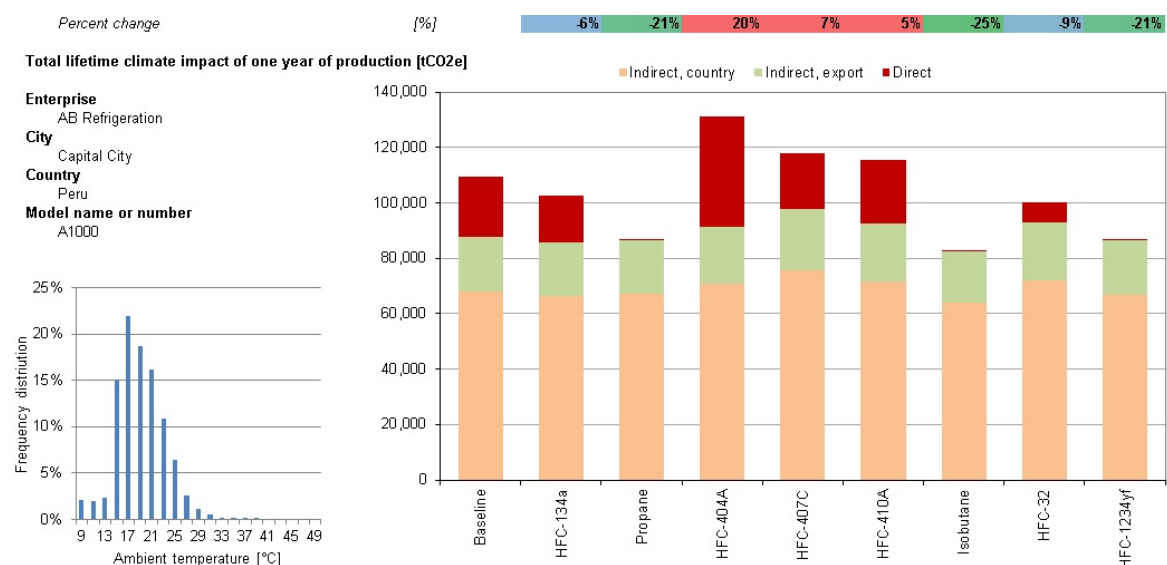
The MCII is not a development tool for the refrigeration or air conditioning system being studied. The internal model for calculating the energy consumption of the system is based on first principles for the thermodynamic circuit. It effectively calculates cycles based on average system characteristics, such as expected compressor efficiencies and heat exchanger performances. The performance of alternative refrigerants is then estimated based on thermodynamic 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:



1.2 Nomenclature

c_p	Specific heat [J/(kgK)]
c	Correction factor for conductance [-]
CI	Climate impact [t CO ₂ e]
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 [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 efficiency (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

<i>AC</i>	Air Conditioning
<i>amb</i>	ambient
<i>air</i>	air
<i>annual</i>	per year
<i>base, AC</i>	base condition for an air conditioning unit
<i>c</i>	condenser, condensation
<i>comp</i>	compressor
<i>design</i>	at design condition
<i>e</i>	evaporator, evaporation
<i>eol</i>	end of life
<i>export</i>	export related part
<i>in</i>	inlet
<i>indoor</i>	indoor (inside the room)
<i>L</i>	load
<i>leak</i>	leakage
<i>life</i>	life time (over entire life span)
<i>mfg</i>	manufacturing
<i>out</i>	outlet, exit
<i>r</i>	refrigerant
<i>R</i>	refrigeration, cooling
<i>sat</i>	saturated refrigerant condition

<i>sub</i>	subcooled refrigerant condition (liquid phase)
<i>super</i>	superheated refrigerant condition (vapour phase)
<i>sv</i>	saturated vapour
<i>sl</i>	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.

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
Evaporator							
Air evaporator inlet	[°C]	26	26	7	7	-16	-16
Air temperature difference	[K]	7	7	3	3	3	3
Air evaporator outlet	[°C]	19	19	4	4	-19	-19
Temperature Differential	[°C]	12	12	12	12	12	12
Evaporator superheat	[K]	5	15	8	20	8	20
Evaporation temperature at design condition	[°C]	14	14	-5	-5	-28	-28
Condenser							
Temperature Differential	[K]	12	12	12	12	12	12
Condenser subcooling	[K]	5	5	5	5	5	5
Heat exchanger effectiveness	[-]	0.5	0.5	0.5	0.5	0.5	0.5
Condensation temperature at design condition	[°C]	38	38	38	38	38	38
Internal heat exchanger							
Heat exchanger effectiveness	[-]	0.5	0.5	0.5	0.5	0.5	0.5
Compressor							
Isentropic efficiency compressor	[-]	0.7	0.7	0.7	0.7	0.7	0.7
Clearance volume ratio	[-]	0.03	0.03	0.03	0.03	0.03	0.03
Running time at design conditions	[%]	80	80	80	80	80	80
General							
Min Cooling Capacity	[W]	1000	1000	200	1000	200	1000
Max Cooling Capacity	[W]	20000	200000	40000	200000	25000	100000
Default Cooling Capacity	[W]	2000	3000	2000	3000	1000	5000
Indoor (room) temperature at design condition	[°C]	26	26	26	26	26	26
Ambient Temperature at design condition	[°C]	32	32	32	32	32	32
Refrigerant characteristics							
Leakage at manufacturing	[%]	2%	2%	2%	2%	2%	2%
Annual leakage	[%]	2%	5%	2%	25%	2%	25%
Recharge level	[%]	55%	55%	55%	55%	55%	55%
Recovery fraction	[%]	0%	0%	0%	0%	0%	0%

Some specific remarks to the table above:

1. The air evaporator inlet 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.
5. For the condenser the temperature differential presents 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 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 cooling capacity 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 parameters 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 used 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 R11e/kg]	GWP [kg CO2e/kg]	Main characteristics
HCFC-22	-40.8	0.04	1790	Base line refrigerant to be replaced due to its ODP
HFC-134a	-26.1	0	1370	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.

Refrigerant Name	Boiling Point [°C]	ODP [kg R11e/kg]	GWP [kg CO ₂ e/kg]	Main characteristics
Propane (R-290)	-42.1	0	20	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	3700	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	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	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	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	716	HFC-32 was originally used as a component of refrigerant blends such as R-404A and R-410A. 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 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	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

Refrigerant Name	Boiling Point [°C]	ODP [kg R11e/kg]	GWP [kg CO2e/kg]	Main characteristics
				efficiency levels comparable to HFC-134a although the theoretical COP is a few percent below that of HFC-134a.

The data above has been taken from UNEP 2010 Report of the Refrigeration, Air Conditioning and Heat Pumps Technical Options Committee report.

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

- 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.
- 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:

- Transport properties such as viscosity and thermal conductivity. Specifically higher thermal conductivities lead to a higher heat transfer coefficient (HTC).
- Thermodynamic properties such as the 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 divided by cross sectional area) if the same tube diameters are maintained, resulting generally in lower HTC values.
- Thermodynamic properties such as 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.
- 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 boil off first followed by the higher boiling point components. 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 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). These aspects are therefore not further considered.

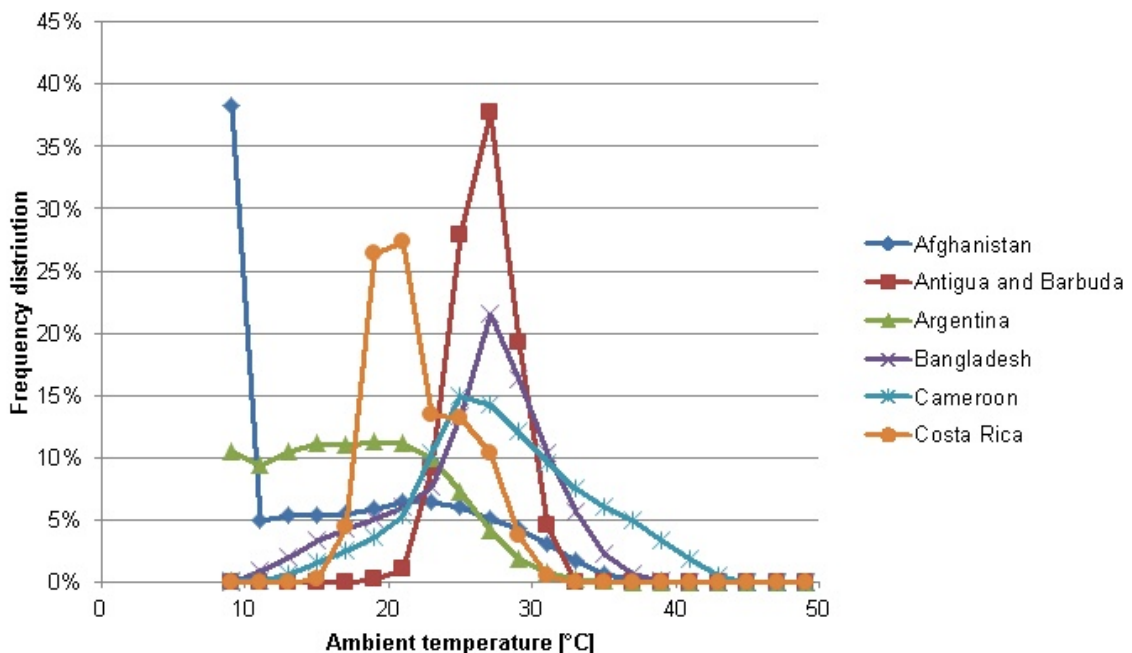
A refrigerant aspect which can not be negated is the zeotropic effect on the HTC and possibly also the thermal conductivity. To compensate the model contains a simple correction factor to be applied to the UA value calculated for HCFC-22 (example for the evaporator):

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

To date all correction factors are set to 1 except for R-407C which is set to 0.9. For R407C a small literature survey was carried out and 7 sources revealed a wide range in change of heat transfer coefficient and reductions from 15 to 70 % 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 the correction coefficient of 0.9.

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 remains below 10 °C, as is the case of Afghanistan, this indicates that during a large part of the year the ambient temperature is below this value. 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 collect 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 Example cases

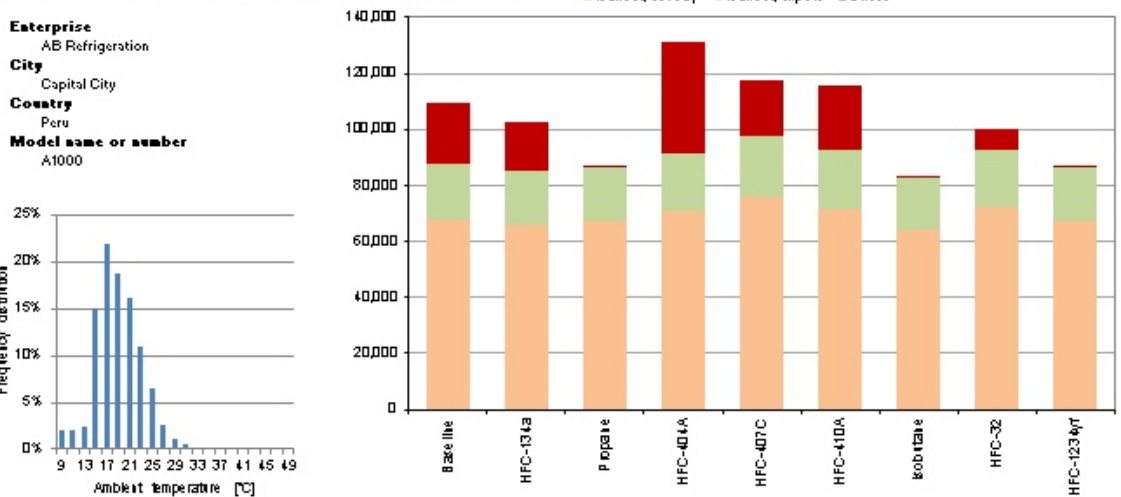
A typical example is included below:

Multilateral Fund Climate Impact Indicator (MCII)

Version 3.1

		Calculating	Date							
General information		Elapsed Time [s]	50							
Enterprise		AB Refrigeration								
City		Capital City								
Country		Peru								
Agency		UNEP								
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								
Cooling capacity per unit										
Minimum for this application type	[W]	200								
Maximum for this application type	[W]	40,000								
Cooling capacity 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]	21,910	16,987	101	33,727	20,136	22,884	113	7,362	11
Indirect impact, related to electricity production										
Country										
Design ambient temperature	[°C]	32								
Electricity consumption, annual	[GWh/y]	27	26	27	28	30	29	25	29	27
Climate impact of lifetime emissions	[t CO2e]	67,927	66,239	67,038	70,788	75,684	71,637	63,907	71,886	66,929
Export										
Global design temperature	[°C]	32								
Electricity consumption, annual	[GWh/y]	3	3	3	4	4	4	3	4	3
Climate impact of lifetime emissions	[t CO2e]	19,801	19,290	19,524	20,655	22,024	20,926	18,592	21,003	19,477
Total impact breakdowns										
Change in direct impact	[t CO2e]		-4,323	-21,808	17,817	-1,774	975	-21,797	-14,548	-21,898
Change in indirect impact, country	[t CO2e]		-1,688	-883	2,861	7,757	3,710	-4,020	3,959	-938
Change in indirect impact, global	[t CO2e]		-511	-277	853	2,223	1,124	-1,209	1,202	-324
Total impact summary										
Total	[t CO2e]	109,638	102,517	86,664	131,163	117,845	115,447	82,612	100,250	86,417
Change	[t CO2e]		-7,121	-22,974	21,531	8,207	5,809	-27,026	-9,388	-23,221
Percent change	[%]		-6%	-21%	20%	7%	5%	-25%	-9%	-21%

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



3 Cycle model description

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 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 R22 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 R22 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 a temperature efficiency of such heat exchanger:

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

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:

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

where again refrigerant property data has been used to calculate 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 (and hence also the air volumetric flow rate by dividing it with the density):

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

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

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

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

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 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:

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

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 temperature efficiency is supplied by the [system selected](#), it is possible to calculate the air exit temperature:

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

Knowing all temperatures the logarithmic temperature difference can be calculated:

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

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 has already been established and the refrigerant state points at inlet and exit of the condenser are already known from the previous analysis:

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

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:

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

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

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 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 defined as:

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

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 and the room. However, even if ambient temperature and room temperature are equal, in general a load for the AC unit remains due to internal heating in the room, presence of humans etc.

To accommodate for this a base heat load has been expressed as a fraction of the design heat load at a base temperature condition which is set in the model to 24 C:

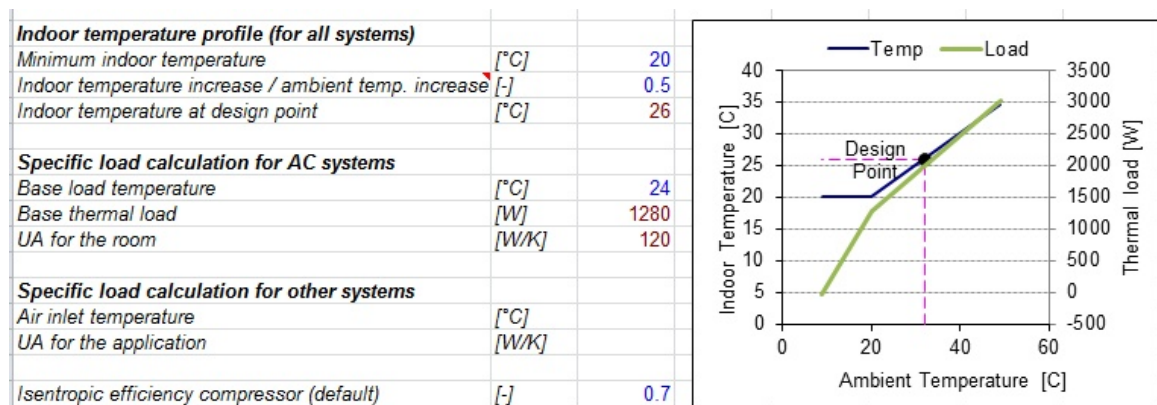
$$Q_{Base,AC} = Q_R (1 - 0.045(T_{amb,design} - T_{Base,AC}))$$

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

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

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

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:



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 generally made available.).
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 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)

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

and the compressor displacement volume is typically found as the product of the compressor swept volume and the operating frequency. In the model the compressor displacement volume flow ϕ_v is used rather than swept volume in order to make systems independent on operating frequency as this is generally linked to the mains supply frequency.

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:

$$COP = \frac{Q_R}{P_{comp}} \quad [\text{comp5}]$$

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:

$$Q = \dot{m}_{c,air} c_{p,air} (T_{c,air,out} - T_{c,air,in}) \quad [\text{cond1}]$$

$$Q = \dot{m}_r (h_{c,in} - h_{c,out}) \quad [\text{cond2}]$$

$$Q = (UA)_c LMTD_c \quad [\text{cond3}]$$

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

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

$$LMTD_c = \frac{T_{c,air,in} - T_{c,air,out}}{\ln\left(\frac{T_c - T_{c,air,in}}{T_c - T_{c,air,out}}\right)} \quad [\text{cond4}]$$

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 efficiency:

$$\eta_c = \frac{T_c - T_{c,air,out}}{T_c - T_{c,air,in}} \quad [\text{cond5}]$$

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

$$NTU_c = \frac{(UA)_c}{\dot{m}_{c,air} c_{p,air}} \quad [\text{cond6}]$$

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

$$\eta_c = 1 - e^{-NTU_c} \quad [\text{cond7}]$$

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:

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

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:

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

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

For the condenser heat exchanger, cross flow is assumed for which the heat exchanger efficiency is defined as:

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

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

if $c_{air} > c_{refr}$ 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:

$$Q_R = \dot{m}_{e,air} c_{p,air} (T_{e,air,in} - T_{e,air,out}) \quad [\text{evap1}]$$

$$Q_R = \dot{m}_r (h_{e,out} - h_{e,in}) \quad [\text{evap2}]$$

$$Q_R = (UA)_e LMTD_e \quad [\text{evap3}]$$

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

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

$$LMTD_e = \frac{T_{e,air,out} - T_{e,air,in}}{\ln\left(\frac{T_{e,air,in} - T_e}{T_{e,air,out} - T_e}\right)} \quad [\text{evap4}]$$

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 efficiency:

$$\eta_e = \frac{T_{e,air,out} - T_e}{T_{e,air,in} - T_e} \quad [\text{evap5}]$$

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 cross flow heat exchanger, it is now possible to relate the number of transfer units and the heat exchanger efficiency 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:

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

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:

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

The number of heat transfer units formula required here is a more complicated form 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:

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

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

For the evaporator heat exchanger, cross flow is assumed for which the heat exchanger efficiency is defined as:

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

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

if $c_{air} > c_{refr}$ *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 liquid and total flow). 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 efficiency of such heat exchanger must be supplied. This (temperature) efficiency 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:

if $T_{amb,design} < T_{indoor,minimum}$ *then*

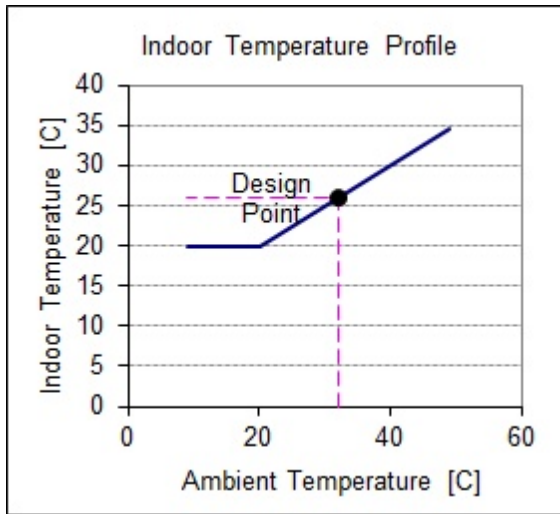
$$T_{indoor,design} = T_{indoor,minimum}$$

else

$$T_{indoor,design} = (T_{amb,design} - T_{indoor,minimum})r + T_{indoor,minimum} \quad [\text{room1}]$$

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 $r=0.5$.



3.3.6 Thermal load

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

$$Q_R = (UA)_L (T_{indoor} - T_{e,air,in}) \quad [\text{therm1}]$$

for an AC unit the thermal load is expressed as:

$$Q_R = (UA)_L (T_{amb} - T_{indoor}) + Q_{Base,AC} \quad [\text{therm2}]$$

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](#) and the air mass flow rates and the conductance's are obtained from the calculation at the design condition for R22. 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 R22 calculation [therm1] or [therm2]
5. The required refrigeration capacity is obtained from the thermal load and the required compressor run time by:

$$Q_R = \frac{Q_L}{R_p}$$

6. 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].
7. 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].

8. The evaporator air outlet temperature is calculated using the heat transfer relation for the air side [evap1].

9. The evaporator logarithmic mean temperature difference can be calculated from all know temperatures [evap4]

10. The number of transfer units for the evaporator is calculated using the conductance and the air mass flow [evap6] and from this the heat exchanger efficiency [evap7].

11. The condenser exit temperature is calculated from the condensation temperature and the given subcooling;

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

12. From this temperature and the refrigerant properties, the enthalpy at the outlet is derived.

13. The evaporator exit temperature is calculated using the evaporation temperature and the given superheat:

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

14. and from the refrigerant properties the enthalpy at evaporator exit is found.

15. From the internal heat exchanger formula [ihe1] the exit temperature at the low pressure side is obtained.

16. 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.

17. 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:

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

15. The refrigerant mass flow now follows from the heat transfer relation of the refrigerant side in the evaporator [evap2].

16. The refrigerant inlet temperature for the compressor is assumed to be equal to the internal heat exchanger exit temperature at the low pressure side:

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

20. and from refrigerant properties the enthalpy and entropy at the compressor inlet are calculated.

21. The isentropic compressor exit entropy is by definition equal to the inlet entropy.

22. From this isentropic exit entropy and the condensation temperature the isentropic enthalpy is calculated using refrigerant properties.

23. From the isentropic efficiency given by the user in the parameter list, the real compressor exit enthalpy is calculated [comp3]

24. From the condensation and evaporation pressures (calculated from their respective saturation temperatures) the volumetric efficiency is calculated [comp2]

25. From the compressor mass flow equation, the required volumetric flow rate is calculated [comp1].

26. From the enthalpy difference over the compressor and the mass flow the compressor power is calculated [comp4]

27. Using the compressor power and the total cooling capacity the COP is calculated [comp5]

28. The number of transfer units for the condenser is calculated using the conductance and the air mass flow [cond6] and from this the heat exchanger efficiency [cond7]

29. The condenser inlet enthalpy is taken from the compressor outlet enthalpy.

30. The heat rejected from the condenser follows from the heat transfer relation of the refrigerant side [cond2]

31. The air outlet temperature of the condenser is calculated using the heat transfer relation for the air side [cond1]

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.
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):

$$R_{life} = 1 + R_{recharge\,life} + R_{mfg} - R_{recovery}$$

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:

$$R_{recharge\,life} = n_{recharge} R_{recharge}$$

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:

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

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:

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

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:

$$CI_{life} = \frac{M_{life} GWP_r}{1000} \quad [\text{t CO}_2\text{e}]$$

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 R22, 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 R22:

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

E.g. for hydrocarbons the liquid density is less than half the value of R22 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:

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

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:

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

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:

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

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	Based on Refprop 9.0							
	Enthalpy Table [J/kg]							
	dT superheat [K]							
	Tsat [K]	psat [Pa]	Hsl [J/kg]	Tbubble [K]	Rhosl [kg/m ³]	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

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