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COMITÉ EJECUTIVO DEL FONDO MULTILATERAL
 PARA LA APLICACIÓN DEL
 PROTOCOLO DE MONTREAL
Septuagésima tercera Reunión
París, 9 – 13 de noviembre de 2014

INFORME SOBRE EL INDICADOR DE IMPACTO
CLIMÁTICO DEL FONDO MULTILATERAL (DECISIÓN 69/23)

Antecedentes

1.   El documento sobre el análisis revisado de las consideraciones relativas a los costos pertinentes en torno a la financiación de la eliminación[1] de los HCFC incluía una sección sobre cuestiones medioambientales y un enfoque para evaluar las emisiones relativas al clima durante el ciclo de vida de un producto que contiene HCFC. En sus debates, el Comité Ejecutivo estudió seguir analizando si el enfoque descrito en el documento ofrecía una base satisfactoria y transparente para la determinación de las prioridades entre las tecnologías de eliminación de los HCFC para minimizar otros impactos sobre el medio ambiente, incluso sobre el clima, tal y como se prevé en la decisión XIX/6. El Comité Ejecutivo solicitó a la Secretaría que prosiga con su evaluación y que informe al respecto, de manera más detallada, en una reunión posterior (decisión 55/43(g)).

2.   En su 57[a] reunión, el Comité Ejecutivo estudió el documento sobre la determinación de las prioridades entre las tecnologías de eliminación de los HCFC para minimizar otros impactos en el medio ambiente[2], que contenía un informe de situación sobre el análisis más detenido de un enfoque de unidades funcionales para la determinación de las prioridades entre las tecnologías de eliminación de los HCFC para minimizar otros impactos en el medio ambiente. El Comité Ejecutivo fue informado[3], entre otras cosas, de que la Secretaría, con el apoyo de expertos, había emprendido una consulta técnica para permitir la creación de unidades funcionales para los sectores de la refrigeración y de las espumas. A raíz de un debate, el Comité Ejecutivo pidió a la Secretaría que preparara un documento para presentar cuatro ejemplos de la aplicación de la metodología en la 58ª reunión (decisión 57/33(b)).

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[1] UNEP/OzL.Pro/ExCom/55/47.

[2] UNEP/OzL.Pro/ExCom/57/59.

[3] Párrafos 170 a 173 del documento UNEP/OzL.Pro/ExCom/57/69.

3.   En respuesta a la decisión 57/33(b), el Comité Ejecutivo, en su 59ª reunión, estudió el documento sobre la determinación de las prioridades entre las tecnologías de eliminación de los HCFC para minimizar otros impactos en el medio ambiente[4]. En este documento, la Secretaría introdujo la expresión "Indicador de Impacto Climático del Fondo Multilateral" (IICFM) para un concepto simplificado de lo que originalmente eran las unidades funcionales. A raíz de un debate[5], el Comité Ejecutivo, entre otras cosas, pidió a la Secretaría que demostrara la aplicación del IICFM a una muestra de presentaciones de proyectos desde la 60ª reunión en adelante; que finalizara la preparación del IICFM y que pusiera en la intranet los datos subyacentes, la metodología usada y los modelos de trabajo preliminares del programa informático a disposición de los organismos bilaterales y de ejecución y de los miembros del Comité Ejecutivo (decisión 59/45(c), (d) y (e)).

4.   Posteriormente, el IICFM se desarrolló aún más haciendo hincapié en el sector de fabricación de aparatos refrigeración y aire acondicionado; se puso a disposición de los miembros del Comité Ejecutivo y de los organismos de ejecución; y la Secretaría ha informado en varias reuniones del Comité Ejecutivo sobre los avances realizados y la experiencia adquirida con el IICFM[6].

5.   El IICFM también se usó para el cálculo del impacto climático asociado con la conversión de empresas de fabricación de aparatos de refrigeración y aire acondicionado asociadas a la etapa I de los planes de gestión de la eliminación de los HCFC aprobados y los resultados se incluyeron en el documento de evaluación del proyecto correspondiente presentado al Comité Ejecutivo[7]. La experiencia adquirida en la aplicación del IICFM demostró que el impacto climático potencial de la conversión de equipos de refrigeración y aire acondicionado desde el nivel básico (HCFC-22) a refrigerantes alternativos puede medirse objetiva y sistemáticamente y permite al Comité Ejecutivo supervisar y justificar el impacto climático potencial de los proyectos apoyados por el Fondo Multilateral.

6.   En la 69ª reunión, el Comité Ejecutivo trató del informe sobre el IICFM[8] presentado por la Secretaría y, entre otras cosas, solicitó a ésta que presentara un informe sobre la marcha de las actividades en la última reunión de 2014, y los resultados de un examen independiente que debería llevarse a cabo (decisión 69/23(b)).

Medidas tomadas para optimizar la herramienta del IICFM

7.   Sobre la base de los comentarios y discusiones con miembros del Comité Ejecutivo[9] y los organismos de ejecución, la Secretaría emprendió el trabajo adicional[10] que concluyó con una herramienta de IICFM totalmente desarrollada que incorporó las siguientes características adicionales:

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[4] UNEP/OzL.Pro/ExCom/59/51 y Add.1.

[5] Párrafos 219 a 227 del documento UNEP/OzL.Pro/ExCom/59/59.

[6]Tal y como se recoge en los siguientes documentos: UNEP/OzL.Pro/ExCom/59/51 y Add.1, UNEP/OzL.Pro/ExCom/62/56 y 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] El IICFM se ha aplicado a los siguientes proyectos (los números entre paréntesis se refieren a los documentos UNEP/OzL.Pro/ExCom/…previos al periodo de sesiones): plan de gestión de la eliminación de los HCFC de Argelia (66/26); proyecto del sector de la fabricación de equipos unitarios de aire acondicionado y para habitaciones de Argentina (61/28); plan de gestión de la eliminación de los HCFC de Bahrein (68/22); plan de gestión de la eliminación de los HCFC de Bosnia (66/30); plan de gestión de la eliminación de los HCFC de China, sector de refrigeración y aire acondicionado industrial y comercial (64/29); plan de gestión de la eliminación de los HCFC de China, sector de fabricación de aparatos de aire acondicionado para habitaciones (64/29); plan de gestión de la eliminación de los HCFC de Indonesia (64/34); plan de gestión de la eliminación de los HCFC de Irán (República Islámica de) (63/35); Jordania, eliminación de HCFC-22 y HCFC-141b en la fabricación de equipos unitarios de aire acondicionado en Petra Engineering Industries Co. (60/31); plan de gestión de la eliminación de los HCFC de Líbano (64/37); plan de gestión de la eliminación de los HCFC de Nigeria (62/43); plan de gestión de la eliminación de los HCFC de Serbia (62/47); plan de gestión de la eliminación de los HCFC de Tailandia (68/41).

[8] UNEP/OzL.Pro/ExCom/69/34.

[9] En respuesta a una carta enviada por la Secretaría en la que invitaba a los miembros del Comité Ejecutivo a usar el IICFM para incluirlo en un documento a presentar a la 73ª reunión, el Gobierno de Australia presentó sus comentarios el 19 de junio de 2014. La Secretaría toma nota con beneplácito de los comentarios recibidos, que han sido tenidos en cuenta.

[10] La Secretaría buscó asesoramiento de expertos para modificar la programación de antecedentes de la herramienta IICFM.

a) Se preparó una interfaz simplificada del usuario que incluía el formato para introducir la información mínima exigida; un formato simplificado de salida con valores y gráficos que facilitan el análisis de los resultados; y una información sobre las condiciones meteorológicas de cada país;

b) Un manual de instrucciones fácil de usar que consta de: una explicación breve de la herramienta IICFM, los parámetros básicos de diseño, los resultados y la forma en que deberían ser interpretados; instrucciones simplificadas sobre la forma de usar el IICFM, incluyendo en ello la información mínima necesaria; y una explicación detallada de todas las hipótesis, principios, bases de datos y algoritmos usados en el modelo. El manual de instrucciones se encuentra en el Anexo I del presente informe;

c) Inclusión de dos refrigerantes más, el HFC-32 y el HFO-1234yf, en la lista de refrigerantes ya disponible en el IICFM. Se estudiaron varios refrigerantes alternativos, pero aún no están normalizados mediante un número de refrigerante emitido por la American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), o no están relacionados con las aplicaciones cubiertas por el IICFM. En el futuro se pueden agregar nuevos refrigerantes alternativos al IICFM, según las necesidades; e

d) Inclusión de las propiedades de transporte de los refrigerantes para tener en cuenta las diferencias entre tecnologías en materia de intercambio térmico y de pérdida de presión, para lo cual se usó un coeficiente de corrección de la transferencia térmica basado en las características específicas de cada refrigerante[11].

8. En respuesta a la decisión 69/23(c), en la que se pidió a la Secretaría que realizara un examen independiente del IICFM, la Secretaría envió el IICFM totalmente desarrollado a tres expertos cualificados. El informe de los expertos se incluirá en la adenda al presente documento antes de la 73ª reunión. Las sugerencias formuladas por los expertos para seguir mejorando el IICFM también se pueden agregar a la recomendación de la Secretaría.

El IICFM y su propósito

9. El IICFM es una herramienta que:

a) Ofrece una indicación del efecto sobre el clima asociado con la conversión de empresas que fabrican equipos de aire acondicionado, equipos de enfriamiento comercial y equipos de congelación comercial desde el HCFC-22 (nivel básico) a refrigerantes[12] alternativos;

b) Funciona sobre la base de los siguientes parámetros que se incluyen en las propuestas de proyectos de conversión de empresas de fabricación de equipos presentadas al Comité Ejecutivo: país[13], tipo de equipo[14], cantidad de unidades producidas al año, carga de

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[11] Todos los factores de corrección se han establecido en 1 excepto para el HFC-407C, que se ha establecido en 0,9. En la sección 2.3.1 del manual adjunto en el Anexo I del presente documento se presenta una explicación detallada.

[12] Estos refrigerantes son: R-290 (propano), R-600a (isobutano), HFC-134a, HFC-32, HFC-404A, HFC-407C, HFC-410A, y HFO-1234yf.

[13] El IICFM contiene datos sobre el clima y la intensidad de carbono de una gran cantidad de países. La intensidad de carbono se refiere al volumen de CO_2 emitido por cada kWh de electricidad producido. El cálculo del impacto climático de una unidad de refrigeración que funciona en un determinado país tendrá en cuenta la frecuencia específica con que se producen temperaturas con arreglo a las cuales el equipo funcionará y la intensidad de carbono de la energía producida en el país en el que funcionará el equipo.

[14] El IICFM se ocupa de equipos de aire acondicionado, equipos de enfriamiento comercial y equipos de congelación comercial. Se hace una diferencia entre los sistemas ensamblados en fábrica y los sistemas ensamblados en el lugar (por ej., una unidad de condensación colocada sobre un almacén frigorífico y un conjunto de evaporador/ventilador colocado dentro de un almacén frigorífico).

refrigerante por unidad, vida del producto (años); capacidad de enfriamiento por unidad (W); y cantidad de unidades exportadas[15];

c)   Tiene en cuenta las emisiones de refrigerantes durante la fabricación, funcionamiento y al final de la vida (emisiones directas) y las emisiones de gases de efecto invernadero que se producen como resultado del consumo energético de los equipos de refrigeración y aire acondicionado basados en HCFC-22 (nivel básico) y en refrigerantes alternativos (emisiones indirectas);

d)   Calcula, para una unidad de refrigeración o de aire acondicionado, las emisiones equivalentes de CO_2 a lo largo de su vida, como la suma de las emisiones directas e indirectas, y multiplica el resultado por el número de unidades fabricadas en un año. Este resultado provisional representa el impacto climático de la producción anual de una tecnología dada a lo largo de la vida del equipo producido; y

e)   Compara el impacto climático de la unidad de refrigeración o de aire acondicionado seleccionada usando diversas tecnologías, incluso la tecnología básica (HCFC). El coeficiente de la tecnología de base (HCFC) y la alternativa se usa para obtener una comparación cualitativa de las distintas alternativas (valores porcentuales). Los valores negativos para el IICFM denotan una reducción en el impacto climático con respecto al nivel básico; los valores positivos denotan un aumento. La descripción del modelo de emisiones ofrece más detalles.

10.   Sin embargo, el objetivo del IICFM es únicamente ofrecer una indicación del impacto climático antes de cualquier actividad de conversión sobre la base de datos limitados y no hacer las veces de los análisis que se podrían realizar a partir de información más detallada sobre el desempeño de un determinado equipo de refrigeración o de aire acondicionado, tales como un análisis del desempeño climático del ciclo de vida o del análisis del ciclo de vida.

11.   Para realizar sus cálculos, el IICFM posee un modelo interno que calcula el consumo energético del sistema a partir de principios fundamentales del circuito termodinámico. Efectivamente, calcula los ciclos sobre la base de las características promedio del sistema, tales como las eficiencias del compresor previstas y el desempeño del intercambiador de calor. A continuación, se estima el desempeño de los refrigerantes alternativos a partir de principios fundamentales basado en las diferencias termodinámicas con los HCFC-22. El modelo supone que los refrigerantes alternativos no tienen repercusión en la eficacia del compresor ni en el desempeño del intercambiador de calor, lo cual, en realidad, puede que no sea el caso, ya que estos componentes deben ser optimizados —o puede que lo necesiten— para el refrigerante alternativo seleccionado.

Ejemplo de uso del IICFM totalmente desarrollado

12.   Para ilustrar el tipo de información que daría el IICFM totalmente desarrollado, la herramienta se aplicó para 100 000 unidades de equipos de aire acondicionado, con una capacidad de enfriamiento de 2 000 W y una vida del producto de 10 años, fabricadas en tres países con distintas fuentes de energía, a saber: país A (emite 0,057 kg de CO_2 por cada kWh de electricidad producido), país B (emite 0,827 kg de CO_2 por cada kWh de electricidad producido) y país C (emite 0,478 kg de CO_2 por cada kWh de electricidad producido). Los resultados se presentan en la figura 1.

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[15] En el IICFM el impacto climático del equipo exportado se calcula sobre la base de los valores promedio de los datos sobre el clima y la intensidad de carbono de todos los países.

Figura 1. Resultados del IICFM aplicado a tres países con intensidades de carbono distintas
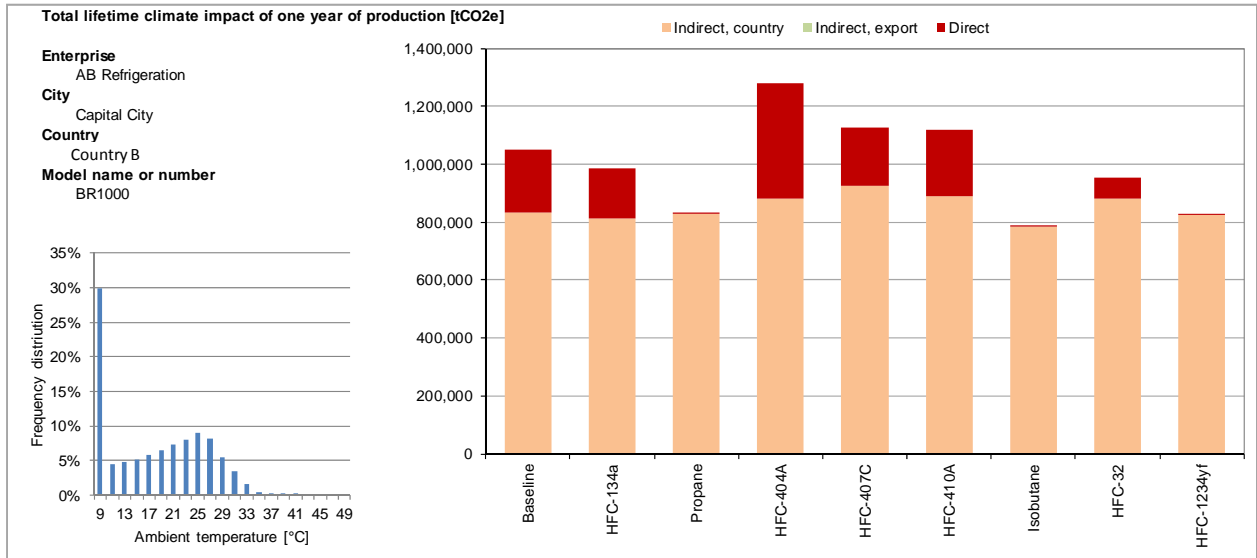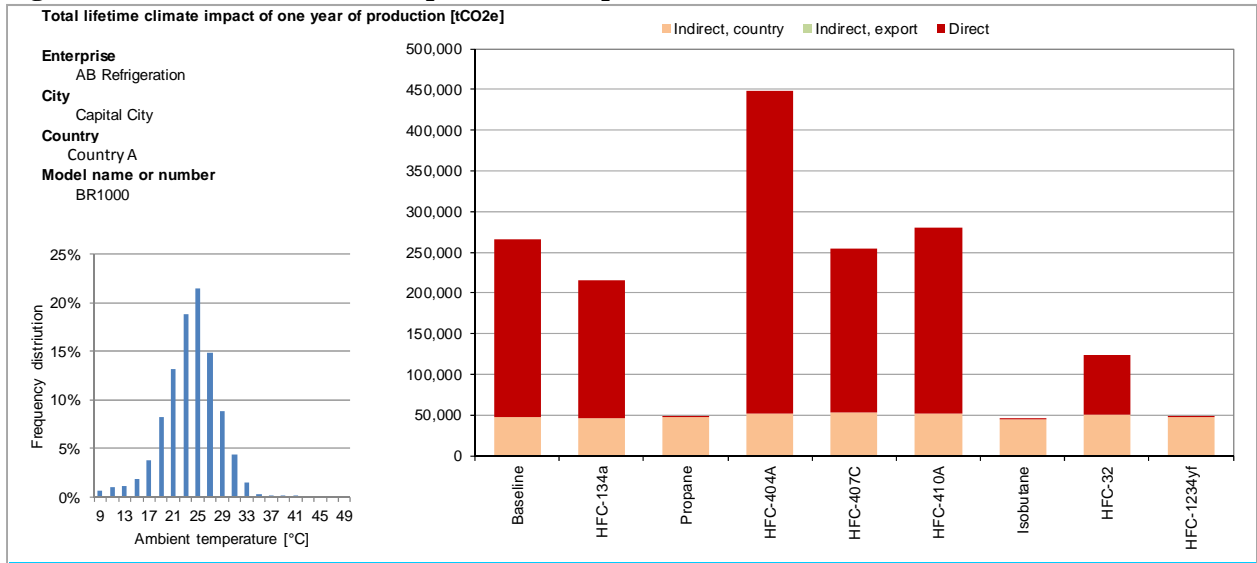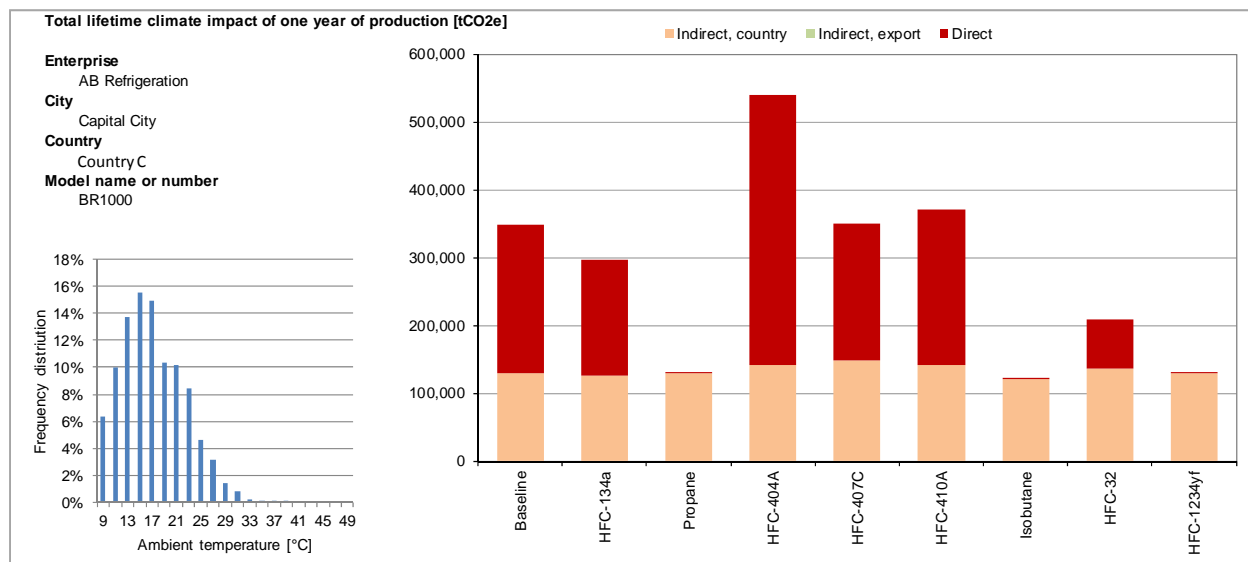
13. Las observaciones siguientes son importantes:

a) En el país A, el peso sobre el impacto climático potencial de las emisiones directas es mayor que el de las emisiones indirectas debido a su intensidad de carbono (0,057 kg de CO_2 emitido por cada kWh de electricidad producido). Las emisiones básicas de 266 145 toneladas de equivalente de CO_2 son en un 83 por ciento directas y en un 17 por ciento indirectas. Al seleccionar un refrigerante con bajo potencial de calentamiento de la atmósfera (propano, isobutano o HFC-1234yf), las emisiones totales de equivalente de CO_2 podrían reducirse en más del 80 por ciento;

b) En el país B, al haber una intensidad de carbono mayor asociada con la fuente de energía (0,827 kg de CO_2 emitido por cada kWh de electricidad producida), las emisiones indirectas constituyen una parte mayor del total de emisiones. Las emisiones básicas de 1 051 341 toneladas de equivalente de CO_2 son cuatro veces superiores a las del país A. En este caso, las emisiones directas representan sólo el 21 por ciento de las emisiones totales. Al seleccionar una tecnología con bajo potencial de calentamiento de la atmósfera, las emisiones totales de equivalente de CO_2 podrían reducirse en un 25 por ciento en el mejor de los casos; y

c) En el país C, con una intensidad de carbono asociada a la fuente de energía mayor que en el país A pero inferior a la del país B (0,478 kg de CO_2 emitido por cada kWh de electricidad producido), se producen unas emisiones básicas de 347 934 toneladas de equivalente de CO_2, de las cuales el 63 por ciento son emisiones directas. Al introducir tecnologías con bajo potencial de calentamiento de la atmósfera, las emisiones totales de equivalente de CO_2 podrían reducirse en un 65 por ciento en el mejor de los casos.

Cálculo del impacto climático para otros sectores de fabricación

14. Tras una evaluación detallada de las metodologías usadas actualmente para calcular el impacto climático para otros sectores de fabricación y la viabilidad de integrarlas en el IICFM, la Secretaría llegó a la conclusión de que esos cálculos no deberían cambiar, tal y como se describe en el documento UNEP/OzL.Pro/ExCom/69/34:

a) Para el sector de las espumas, el enfoque más sencillo es seguir calculando el impacto climático sobre la base de una comparación del volumen de HCFC usado para la

espumación, los volúmenes de agentes espumantes alternativos usados tras la conversión y el potencial de calentamiento de la atmósfera de ambas sustancias. Este enfoque se ha usado así en todos los cálculos de impacto climático presentados al Comité Ejecutivo con motivo de las presentaciones de planes de gestión de la eliminación de HCFC;

b) Para los sectores de aerosoles y disolventes y, cuando sea aplicable, en el sector de agentes de procesos, se supuso que todas las sustancias se emitían en la atmósfera el mismo año en el que se había producido o importado el disolvente o el agente de procesos. Los cálculos del impacto climático seguirán haciéndose sobre la base de los volúmenes de disolventes usados antes (HCFC) y después de la conversión, y el potencial de calentamiento de la atmósfera de ambas sustancias; y

c) Para el sector de lucha contra incendios, la liberación de agentes de lucha contra incendios se demora en comparación con la instalación inicial. Actualmente, no existe una pauta clara para clasificar claramente las aplicaciones y poder realizar un análisis de las emisiones. Por consiguiente, el cálculo del impacto climático se realizará sobre la base de una evaluación de los volúmenes usados, de las tecnologías alternativas seleccionadas y de la pauta de uso.

Conclusiones

15. La Secretaría seguirá aplicando el modelo del IICFM para calcular el impacto que tienen en el clima los proyectos de inversión en los sectores de fabricación de aparatos de refrigeración y de aire acondicionado. El cálculo del impacto climático de proyectos de inversión en otros sectores de fabricación seguirá realizándose a partir de las metodologías descritas en el párrafo 14.

16. No es posible integrar el IICFM en la base de datos de acuerdos plurianuales, dado que las propiedades operativas de las dos aplicaciones son diferentes.

17. La hoja de cálculo de Microsoft Excel para el IICFM está disponible en el sitio web de la Secretaría.

Recomendación

18. El Comité Ejecutivo podría:

a) Tomar nota del informe de la Secretaría sobre el Indicador de Impacto Climático del Fondo Multilateral (IICFM) totalmente desarrollado (decisión 69/23) que se recoge en el documento UNEP/OzL.Pro/ExCom/73/54; y

b) Tomar nota de que la Secretaría seguirá calculando el impacto que tienen en el clima los proyectos de inversión en los sectores de fabricación de aparatos de refrigeración y de aire acondicionado con el modelo del IICFM, y de que para proyectos de inversión en otros sectores de fabricación seguirá aplicando las metodologías descritas en el párrafo 14 del documento UNEP/OzL.Pro/ExCom/73/54.

MCII Model, Refrigeration and AC systems

MCII Model, Refrigeration and AC systems

Printed: October 2014

Table of Contents

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:

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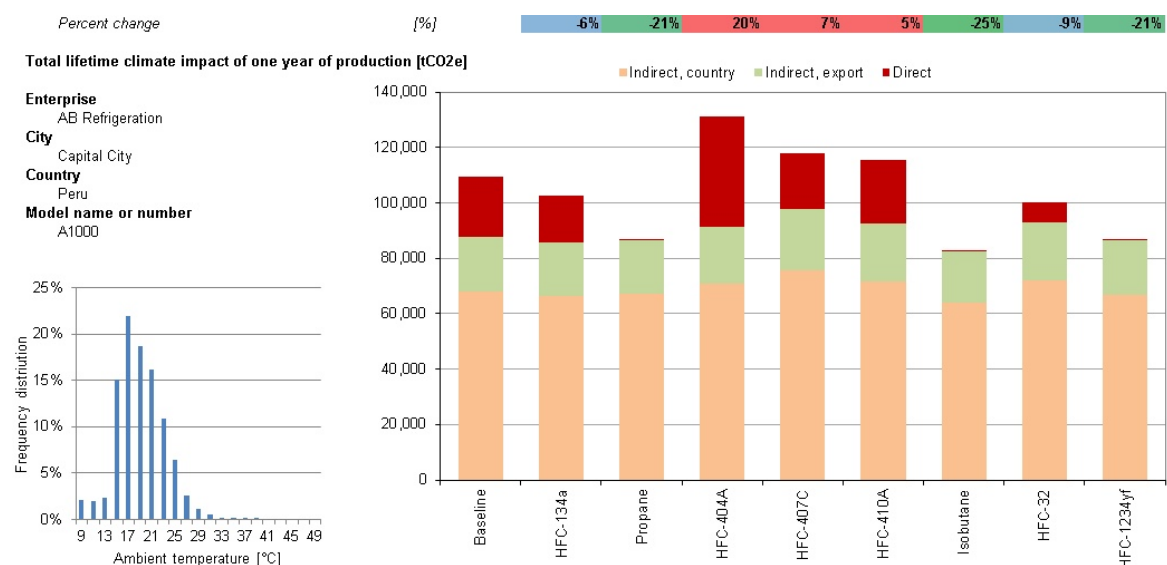
1.2 Nomenclature

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 [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/m3]

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.

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 CO_2 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 CO_2 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 CO_2 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 CO2e/ 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:

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 lead to a higher heat transfer coefficient (HTC).

b) 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.

c) 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.

d) 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:
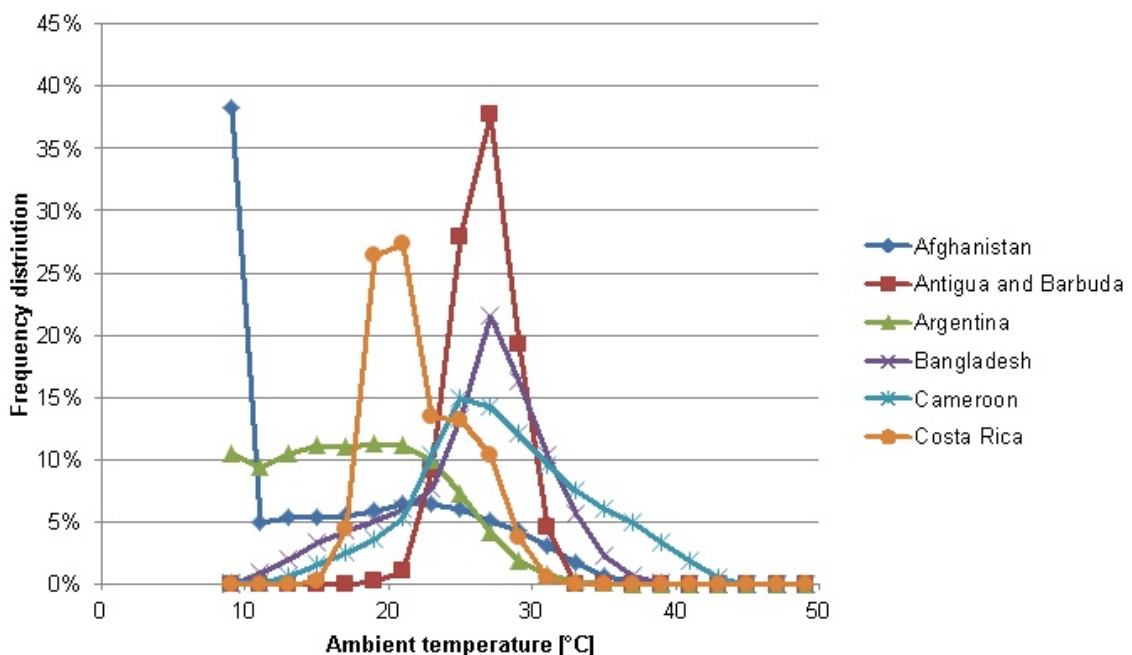
The temperature has been listed in increments of 2 K, this between 9 and 49 °C ambient. In case a large fraction remains below 10 °C, as is the case of Afghanistan, this indicates that during a large part of the year the ambient temperature is below this value. The calculations of the systems do not 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 CO_2 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 CO_2 whereas coal driven plants generally emit a large amount of CO_2.

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.
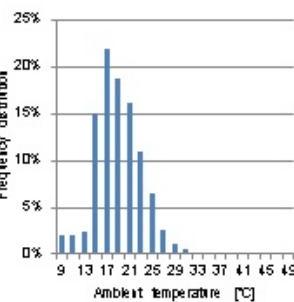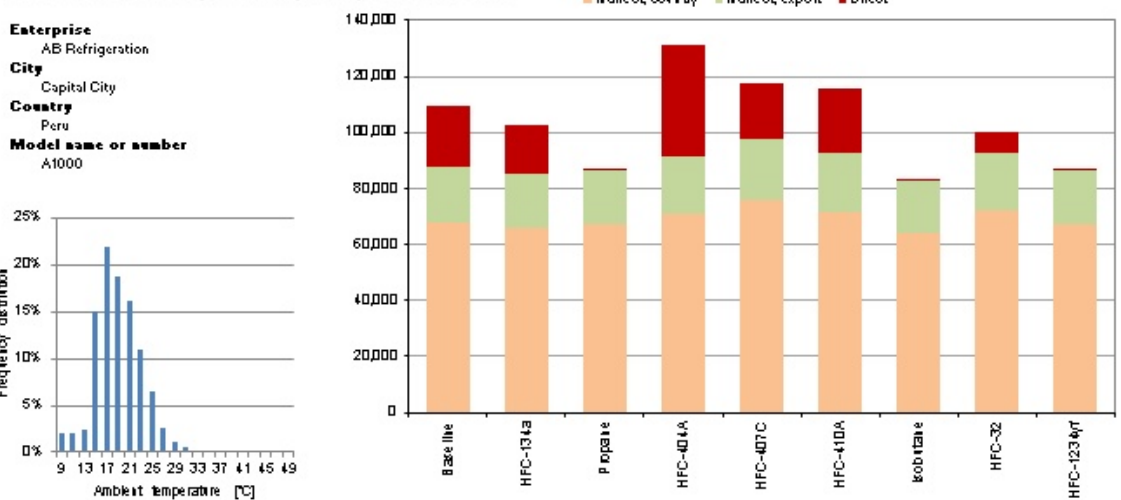
2.6 Example cases

A typical example is included below:

Multilateral Fund Climate Impact Indicator (MCII)

Version 3.1

		Calculate		Calculating	Done					
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]	21,310	16,387	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 breakdown										
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	-889	2,861	7,757	3,710	-4,020	3,959	-998
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	131,169	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]

Indirect, country ■ Indirect, export ■ Direct

Enterprise
AB Refrigeration
City
Capital City
Country
Peru
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 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:

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

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 (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:

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

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:

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

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 temperature efficiency is supplied by the system selected, it is possible to calculate the air exit temperature:

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

Knowing all temperatures the logarithmic temperature difference can be calculated:

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

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:

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

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 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:

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

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:
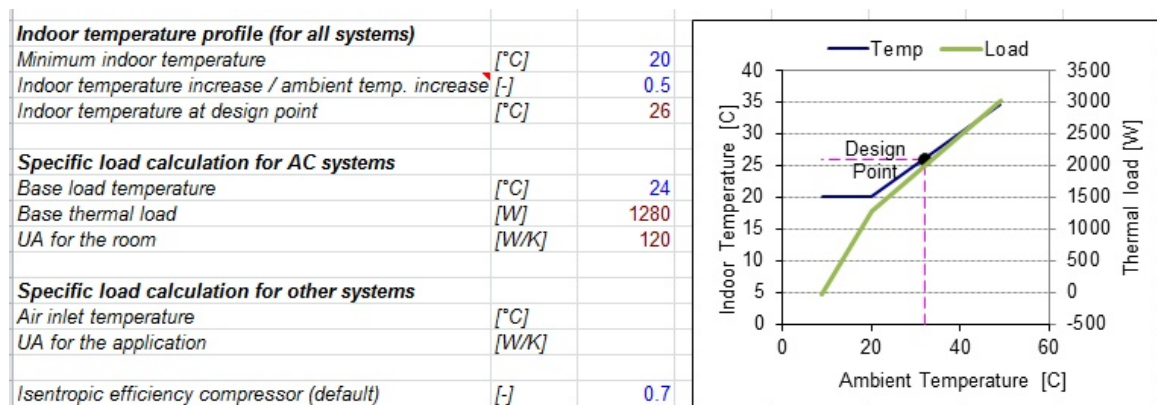
3.3   System calculations

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*)

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

and the compressor displacement volume is typically found as the product of the compressor swept volume and the operating frequency. 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:

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

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

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

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:

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

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

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

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

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:

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

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

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

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

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

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:

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

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:

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

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

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

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

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:

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

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:

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

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

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

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

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

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:

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

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:

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

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:

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

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

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

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 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:

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

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:

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

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:

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

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

$$else$$

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

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.
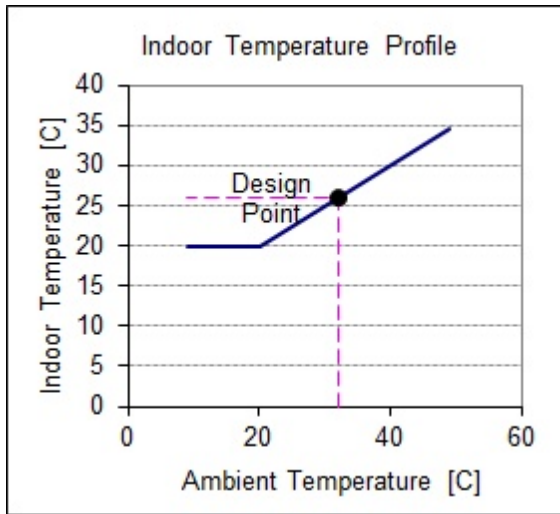
3.3.6 Thermal load

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

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

for an AC unit the thermal load is expressed as:

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

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):

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

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 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:

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

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
	280.4	588882	210352	274.2	1209	412899	422684	432

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
```