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Sustainable cooling supply for building air conditioning and industry in Germany

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Sustainable cooling supply for building air conditioning and industry in Germany

by

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Kurzbeschreibung

Die Kältetechnik in Deutschland weist insgesamt einen Anteil am Elektroenergiebedarf von ca. 14 % auf und ist für etwa 5 % der nationalen Treibhausgasemissionen verantwortlich. Angesichts der nationalen Klimaschutzziele verdeutlichen diese Zahlen den Handlungsbedarf zur Erhöhung der Klimafreundlichkeit in der Klima- und Kältetechnik.

Die Studie untersucht Möglichkeiten zur Steigerung der Klimafreundlichkeit der Kältebereitstellung in den Anwendungsgebieten Gebäudeklimatisierung und Industriekälte. Ausgangspunkt ist eine detaillierte Analyse des Kältebedarfs in den einzelnen Anwendungsbranchen sowie eine Charakterisierung der eingesetzten Kühltechniken. Hierauf aufbauend werden für eine Auswahl an Systemen für die Gebäudeklimatisierung sowie die Industriekälte mittels ganzjähriger Betriebssimulationen Energiebedarf, Treibhausgas-Emissionen und Kosten berechnet und bewertet. Weiter werden die Potenziale zur Substitution von Kompressionskältesystemen durch wärmegetriebene Kältesysteme anhand verschiedener Randbedingungen ermittelt. Zum Abschluss wird das Marktpotenzial klimafreundlicher Kühltechniken untersucht und Empfehlungen zur Steigerung des Marktpotenzials gegeben.

Abstract

Refrigeration technology exhibits a share of approximately 14 % of the total electrical energy demand in Germany and it is responsible for approximately 5 % of the national greenhouse gas emissions. Regarding the national climate protection objectives, these numbers illustrate the requirement for action in order to increase the climate-friendliness of air handling and refrigeration technologies.

The study determines possibilities to maximize the climate-friendliness of refrigeration supply in the field of building air conditioning and industrial refrigeration systems. It is based on a detailed analysis of the refrigeration needs of different industrial sectors and a characterization of the cooling techniques in use. Energy demand, greenhouse gas emissions and costs are calculated and evaluated for a selection of building air conditioning and industrial refrigeration systems. A year-long simulation of the operating refrigeration systems provides the results needed for the calculation. With the help of various boundary conditions the study identifies the replacement potential of compression type refrigeration systems with thermally driven refrigeration systems. Conclusively, the market potential of climate-friendly refrigeration systems is evaluated and recommendations to increase that potential are given.

Content

List of figures

List of tables

List of abbreviations

ACh	Ad- or absorption type refrigeration system
AbCh	Absorption type refrigeration system
AdCh	Adsorption type refrigeration system
AH	Absolute humidity
Ch	Chiller
CHP	Combined heat and power plant
CHPC	Combined heat, power, and cooling plant
COP	Coefficient of Performance (energy efficiency ratio)
CRP	Compression type refrigeration system
DEC	Desiccative and Evaporative Cooling
EEG	”Erneuerbare Energien Gesetz“
EER	Energy Efficiency Ratio
FKT	”Forschungsrat Kältetechnik“
GWP	Global Warming Potential
HFC	Hydro-fluorocarbons
HVAC	Humidification, ventilation, and air conditioning system
KWKG	”Kraft-Wärme-Kopplungs-Gesetz“
LEL	Lower Explosion Limit
Mm^3	million cubic metres (at standard reference conditions)
SEER	Seasonal Energy Efficiency Ratio
TEWI	Total Equivalent Warming Impact (sum of direct and indirect greenhouse gas effect)
VRF	Variable Refrigerant Flow (multi-split system with variable refrigerant flow rate)

1 Zusammenfassung

Die Kältetechnik in Deutschland weist insgesamt einen Anteil am Elektroenergiebedarf von ca. 14 % auf und ist für etwa 5 % der Treibhausgasemissionen (direkte und indirekte Emissionen der Anlagen) verantwortlich. Angesichts der nationalen Klimaschutzziele verdeutlichen diese Zahlen den Handlungsbedarf zur Erhöhung der Klimafreundlichkeit in der Klima- und Kältetechnik. Die vorliegende Studie dient als Grundlage für Aktivitäten zur Verbesserung der Klimafreundlichkeit in diesem Bereich. Sie umfasst Untersuchungen auf dem Gebiet der Industriekälte und Gebäudeklimatisierung.

Ausgangspunkt stellt eine Aufschlüsselung des Kältebedarfs nach Branchen und dessen bisher verwendete Techniken und dem damit verbundenen Endenergiebedarf dar. Diese wird durch charakterisierende Merkmale, welche für spätere Analysen und Bewertungen von Bedeutung sind, ergänzt. Im Anschluss erfolgt ein Überblick über klimafreundliche Kältesysteme und eine Recherche zur Verfügbarkeit der Systeme und benötigter Einzelkomponenten am Markt.

Nachfolgend werden für die Einsatzfälle (Nichtwohn-)Gebäudeklimatisierung und Industriekälte unterschiedliche Systeme und Systemvarianten anhand von dynamischen Anlagensimulationen über einen Jahreszeitraum untersucht. Ausgehend von den Berechnungsergebnissen werden Schlussfolgerungen hinsichtlich der Klimafreundlichkeit der untersuchten Systeme gezogen und der Einfluss unterschiedlicher Faktoren diskutiert.

Bei vorhandenen (Ab)wärmequellen bieten thermisch getriebene Kälte- und Klimasysteme die Möglichkeit zur Primärenergieeinsparung und zur Reduzierung der Treibhausgasemissionen. Die Ausweitung der Nutzung thermisch angetriebener Kältesysteme wird anhand unterschiedlicher Annahmen für die ausgewählten Bereiche untersucht und das mögliche Potenzial dargestellt.

Im Abschluss erfolgt eine Analyse von Hemmnissen und Werkzeugen zur Etablierung nachhaltiger Kältesysteme am Markt.

1.1 Nutz-, Endenergiebedarf und Klimarelevanz der Kältetechnik

Die stationäre Kälteerzeugung in Deutschland ist insgesamt für einen Strombedarf von ca. 70 TWh im Jahr 2008 verantwortlich (Guntram Preuß 2011). Dies entspricht über 13 % des Elektroendenergiebedarfs (Umweltbundesamt 2012a). Mit den spezifischen CO_2-Emissionen der Stromproduktion des Jahres 2008 von 568 g CO_2/kWh_{el} (Umweltbundesamt 2012a) verursachte die Kältetechnik etwa 40 Mio. t CO_2–Emissionen (indirekte Emissionen). Weitere 5,6 Mio. t CO_2-äquivalente Emissionen werden durch die direkte Treibhausgaswirkung von Kältemittelemissionen verursacht (Becken & Plehn 2010). Durch hohe Zuwachsraten bei Neuanlagen im Bereich der Gebäudeklimatisierung steigen die Emissionen in Zukunft weiter an.

Klimarelevanten Daten zur stationären Kältetechnik sind in Tabelle 1 zusammengefasst.

Tabelle 1: Klimarelevante Daten zur stationären Kältetechnik in Deutschland

Kennzahl	Wert	Anmerkung
Elektroenergiebedarf der stationären Kältetechnik in Deutschland	70.695 GWh/a	Datenbasis 2008, (Guntram Preuß 2011),
Anteil am gesamten Elektroenergiebedarf	13,5 %	Datenbasis 2008, (Guntram Preuß 2011; Umweltbundesamt

in Deutschland		2012a)
Treibhausgasemissionen der stationären Kältetechnik in Deutschland	45,8 Mio. t	Datenbasis 2008 für Elektroenergiebedarf (Umweltbundesamt 2012a) , Datenbasis 2007 für direkte Kältemittelbedingte-Emissionen (Becken & Plehn 2010)
Anteil an den gesamten Treibhausgas-emissionen in Deutschland	4,7 %	Datenbasis 2008
indirekter Anteil (Elektroenergiebedarf)	40,2 Mio. t (81,6 %)	
direkter Anteil (Treibhausgaswirkung der Kältemittelemissionen)	5,6 Mio. t (18,4 %)	
Elektroenergiebedarf der Kältetechnik der Bereiche Industriekälte und Gebäudeklimatisierung	30.244 GWh/a (42,8%)	prozentualer Anteil bezogen auf den Elektroenergiebedarf der gesamten Kältetechnik in Deutschland

Abbildung 1 zeigt den Elektroendenergiebedarf der Kältetechnik aufgeteilt nach Anwendungsbranchen (Guntram Preuß 2011). Für die vorliegende Studie sind hierbei die Anwendungen in der Industriekälte, Nahrungsmittelindustrie sowie der stationäre Anteil der Klimakälte relevant. Für diese Anwendungsbereiche ergibt sich ein Gesamtkältebedarf von 63.173 GWh/a (Abbildung 2), für dessen Deckung ein Endenergiebedarf von etwa 30 TWh/a Elektroenergie (5,7% des gesamten Elektroenergiebedarfs in Deutschland) und 3 TWh/a Wärme benötigt werden. Die Datengrundlage wird in Kapitel 5 ab Seite 40 erläutert. Eine detaillierte Unterteilung der Ergebnisse nach Bereichen ist auf Seite 55 enthalten.

Die stationäre Kälte- und Klimatechnik, die Gewerbekälte ausgenommen, verursachte 2007 etwa 800 t direkte HFKW-Emissionen (Becken & Plehn 2010). Dies entspricht ca. 1,6 Mio. t CO_2-Äquivalent. Betrachtet man die in der zitierten Studie dargestellten Emissionen der Vorjahre, so ist ein jährlicher Anstieg zwischen 15 und 20 % zwischen den Jahren 2002 und 2007 zu beobachten.

Der Bereich Klimakälte weist aufgrund der klimatischen Bedingungen in Deutschland eine geringe Vollbenutzungsstundenzahl auf. Der Leitfaden für Energiebedarfsausweise in Nichtwohngebäuden (BMVBS 2007) gibt für Bürogebäude 500 h, für Bildungsgebäude 350 h an. Die durchgeführten Simulationen ergeben Volllaststunden im Bereich von 400 bis 520 h. Im Rahmen des EvaSolK-Projekts wurden unterschiedliche Klimakälteerzeugungssysteme (Kaltwassersätze, Mono- und Multi-Split-Systeme) im praktischen Einsatz vermessen. Hierbei ergaben sich zwischen 170 und 430 Volllaststunden (Wiemken, Safarik, et al. 2012; Wittig et al. 2012). Neben den klimatischen Bedingungen liegt ein weiterer Grund für die geringe Auslastung von Klimaanlagen in der Anlagenauslegung, die neben Sicherheitsfaktoren auch zukünftige Pläne der Betreiber, wie z.B. Gebäudeerweiterungen, berücksichtigt. Dadurch werden Gebäudeklimaanlagen häufig überdimensioniert.

Durch die geringe Vollbenutzungsstundenzahl ist der direkte TEWI-Anteil für Klimakälteanlagen bei Leckraten zwischen 4 und 7 % (Strogies & Gniffke 2013) ein wesentlicher Faktor. Dies zeigen auch die Simulationsergebnisse. Durch den Einsatz des Kältemittels R410A bei Kaltwassersätzen liegt der direkte TEWI-Anteil – je nach betrachteter Klimaregion – bei ca. 40 %. D.h., ca. 40 % der treibhauswirksamen Emission werden allein durch das Entweichen von Kältemittel verursacht und die verbleibenden 60 % durch den Energiebedarf während des Betriebes. Für R134a ergibt sich ein direkter Anteil von ca. 32 % und für R32 von ca. 18 %. Bei dem HFKW-Kältemittel R1234yf und natürlichen Kältemitteln wie R290, R717 und R718 hat der direkte TEWI-Anteil einen sehr geringen bis keinen Einfluss auf den Gesamtwert. Bei direktverdampfenden Systemen ergeben sich aufgrund größerer Kältemittelfüllmengen noch höhere direkte Anteile am Gesamt-TEWI. Unter den oben genannten Annahmen liefern die Simulationsergebnisse einen Anteil von ca. 70 % für das Kältemittel R410A.

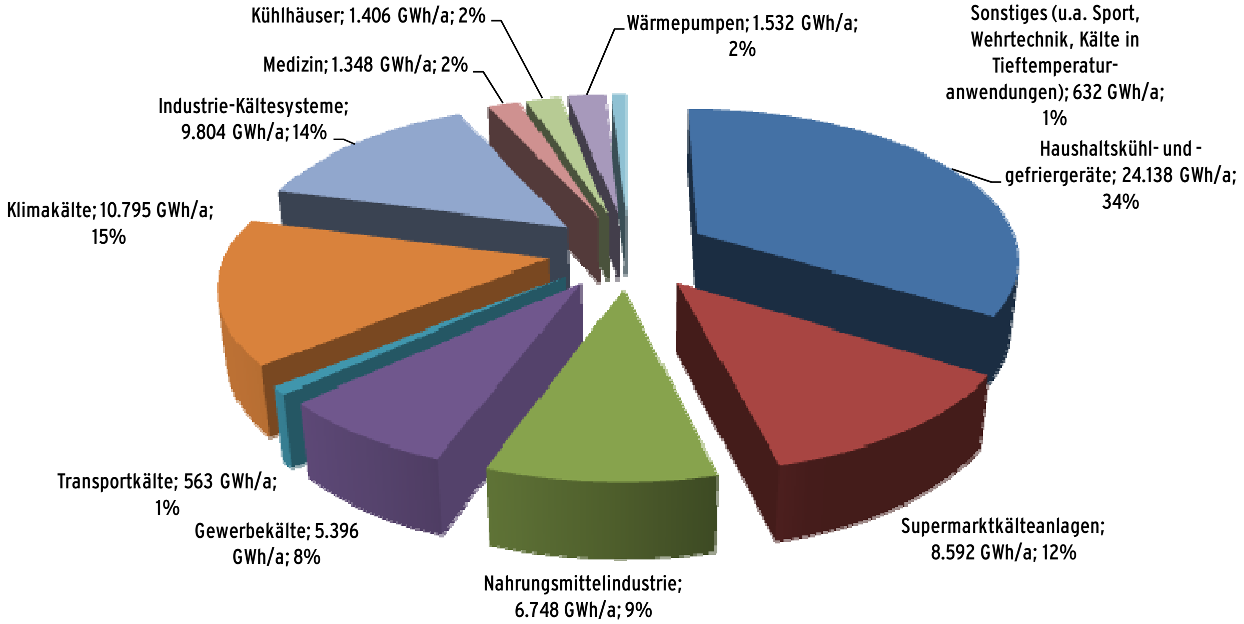


Abbildung 1: Anteil am Elektroendenergiebedarf der Kältetechnik nach Branchen (Guntram Preuß 2011)

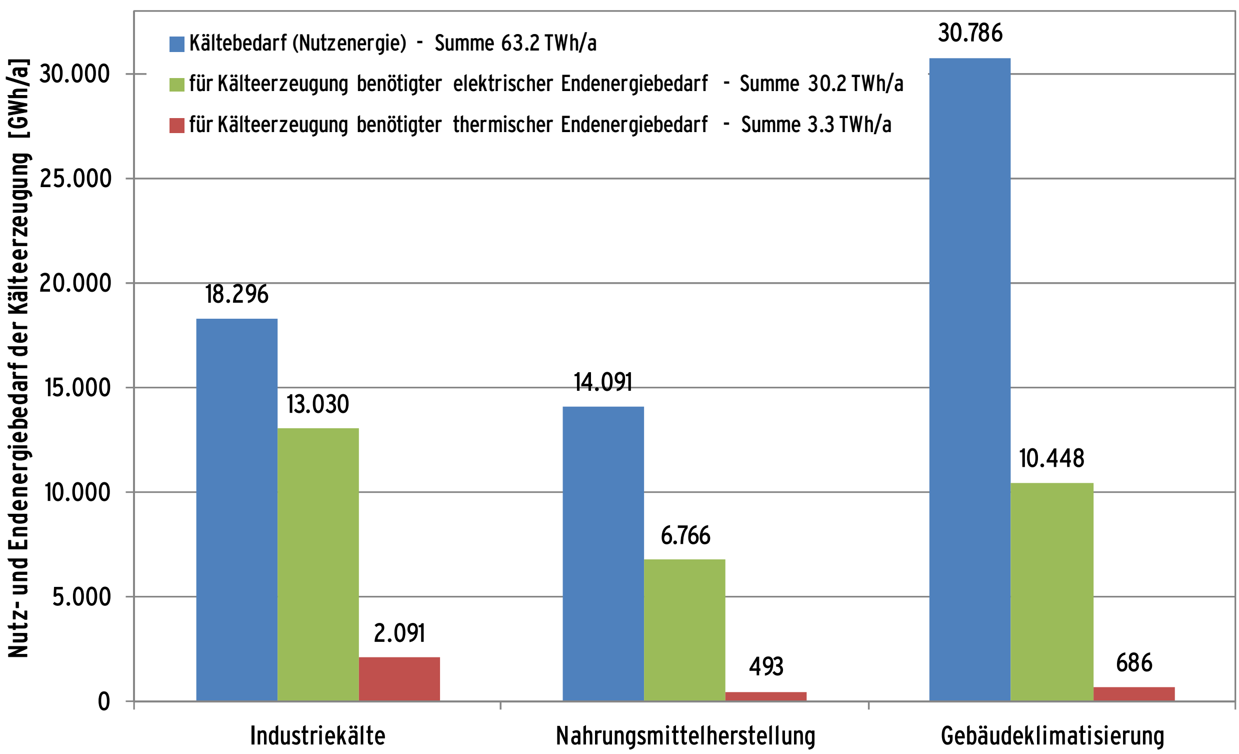


Abbildung 2: Nutz- und Endenergiebedarf der Kälteerzeugung der betrachteten Anwendungen

Bei einer besseren Auslastung der Kälteanlage reduzieren sich die relativen direkten Anteile. So ergibt sich aus den Simulationsergebnissen für einen ganzjährig betriebenen Kaltwassersatz in der Industriekälte ein direkter Anteil von unter 10 %. Trotz dieses relativ geringen Anteils sind die absoluten Emissionswerte durch das Entweichen von Kältemitteln mit hohem GWP-Wert nicht zu unterschätzen.

1.2 Bewertung verschiedener Kälteerzeugungstechniken

Ziel ist die Bewertung verschiedener Kälteerzeugungstechniken in der Gebäudeklimatisierung nach ökologischen und ökonomischen Gesichtspunkten, wobei die Auswirkungen auf das Klima im Vordergrund stehen. Hierzu werden Verbrauchs- und Emissionsdaten verschiedener Kältesysteme ermittelt und miteinander verglichen, welche zur Deckung einer mittels Simulationsprogramm bestimmten Kühllast in der Gebäudeklimatisierung und in der Industriekälteerzeugung herangezogen werden.

Die Basis der Bewertung sind dynamische Simulationen eines Nichtwohngebäudes und der dazugehörigen Anlagentechnik. Die Simulationen erfassen den Kältebedarf über den gesamten Jahreszeitraum in zwei unterschiedlichen Klimaregionen (Frankfurt und Hamburg) bei unterschiedlichen Klimaszenarien (mittlerer Sommer und extremer Sommer). Für die verschiedenen Kältesysteme und Klimaszenarien erfolgen Berechnungen zum Endenergiebedarf und den verursachten Treibhausgasemissionen nach dem TEWI-Ansatz.

Dem Vergleich von Systemen zur Gebäudeklimatisierung wird ein Ein-Zonen-Großraumbüro mit quadratischer Grundfläche von 400 m^2 zugrunde gelegt. Das Lastszenario entspricht einer typischen Büronutzung mit maximalen inneren Lasten von 30 W/m^2. Die Anlagentechnik entspricht dem Stand der Technik (z.B. Kondensationsdruckbegrenzung entsprechend einer Kondensationstemperatur von 15 °C, Kaltwasservorlauftemperatur minimal 10 °C). Die vergleichende Betrachtung wurde mit den folgenden Kältesystemen durchgeführt:

- System mit solar- und fernwärmegetriebener Absorptionskältemaschine und Deckenkühlsystem zur Wärmeübergabe (AKM)
- Multi-Split-System mit variablen Kältemittelmassenstrom unter Verwendung von Gebläsekonvektoren zur Wärmeübergabe (VRF)
- Kompressionskältemaschine (Kaltwassersatz, Kondensationsdruckbegrenzung minimal $t_C =$ 15 °C) mit Deckenkühlsystem oder Gebläsekonvektoren zur Wärmeübergabe sowie mit und ohne Kaltwasserspeicher (KWS)
- Raumlufttechnische Anlage mit offener Flüssigsorption unter Nutzung von BHKW-Abwärme (DEC: desiccative and evaporative cooling)

Für die Untersuchungen im Bereich der Industriekälte wird für alle Systeme ein ganzjährig konstanter Kältebedarf von 500 kW bei einer Kälteträger-Temperatur an der Übergabestelle von +2°C angenommen. Für diesen Bereich erstreckt sich die Betrachtung über folgende Systeme:

- Kaltwassersatz mit Kompressionskältemaschine (KWS)
- Direktverdampfendes Kompressionskältesystem
- BHKW-abwärmegetriebene Absorptionskältemaschine (AbKM)

Ein energetischer Vergleich der Systeme erfolgt anhand des Endenergiebedarfs. Unter Berücksichtigung des Energiegehaltes der einbezogenen Energiequellen und Energieträger wird daraus der Primärenergiebedarf abgeleitet und ebenfalls verglichen. Verwendete

Primärenergiefaktoren sind dem Entwurf der EnEV 2014 bzw. der DIN V 18599–1:2011-12 entnommen.

Für Betrachtungen zur Klimarelevanz ist der TEWI (*Total equivalent warming impact*) eine wichtige Vergleichsgröße. Dieser berücksichtigt zum einen CO_2-Emissioen, welche durch den Energiebedarf während des Anlagenbetriebes hervorgerufen werden (indirekter Anteil). Für die Berechnungen wird Elektroenergie mit 583 g CO_2/kWh (Strommix Deutschland 2010 inkl. Vorkette) bewertet. Die Bewertung der benötigten Wärme erfolgt nach drei verschiedenen Methoden. Mit resultierenden Werten von 0, 44 und 239 g CO_2/kWh lässt sich in Abhängigkeit verschiedener Energiequellen ein Ergebnisraum aufspannen. Für Wärme aus Umweltenergie wird ein Emissionswert von 0 g CO_2/kWh angesetzt. Der TEWI berücksichtigt zum anderen die Emission klimawirksamer Stoffe aufgrund von Leckagen im Kältekreislauf sowie Entsorgungsverluste (direkter Anteil). Es werden je nach System jährliche Leckraten von 4 bis 7 % und Entsorgungsverluste von 30 % zugrunde gelegt.

Die Betrachtungen beinhalten ebenfalls eine ökonomische Gegenüberstellung der verglichenen Systeme. Diese beinhaltet Investitions-, Betriebs-, Instandhaltungs- und Entsorgungskosten. Bei der Berechnung der Betriebskosten werden mehrere Kostenszenarien einbezogen, in denen verschiedene Energiepreiskonstellationen Berücksichtigung finden (Strompreise 6-25 ct/kWh, Wärmepreise 0-8 ct/kWh).

Für die Untersuchungen im Bereich Gebäudeklimatisierung wird auch der Aspekt der thermischen Behaglichkeit einbezogen.

Ergebnisse Gebäudeklimatisierung

Hinsichtlich der Behaglichkeit liefern Systeme mit Konvektoren die besten Ergebnisse. Kühldecken-Systeme sind besonders in feuchtwarmen Sommern nicht in der Lage, eine ausreichende Kühlleistung über den gesamten Klimatisierungszeitraum bei Unterbindung von Kondensation an den Kühlflächen zu erreichen. Eine Feuchtekontrolle ist mit Kühldecken nicht möglich. DEC-Systeme sind im Hinblick auf die Be- und Entfeuchtung positiv zu bewerten, durch die nach unten begrenzte Zulufttemperatur jedoch hinsichtlich der Kühlleistung beschränkt, was besonders an heißen Tagen zu Temperaturüberschreitungen führt. Durch ein solares Kältesystem mit AKM kann unter Berücksichtigung realistischer Größen für Kollektorfläche und Heißwasserspeicher die Kältelast nur durch Einbindung einer zusätzlichen Wärmequelle (z.B. Fernwärme) oder einer Kompressionskälteanlage als Back-up gedeckt werden. Für die betrachteten Klimaszenarien ergeben sich bei rein solarer Kühlung unbefriedigend häufig Überschreitungen der angestrebten maximalen Raumtemperatur. Eine energetische und wirtschaftliche Gegenüberstellung mit den Vergleichssystemen ist daher nur bedingt aussagekräftig.

Wird die rein solare Kühlung außer Betracht gelassen, zeigen sich bei einem primärenergetischen Vergleich für die Kompressionskälte-Systeme die niedrigsten Bedarfswerte. Dabei hat das VRF-System den höchsten Elektroenergiebedarf. Dies ist hauptsächlich bedingt durch die relativ niedrigen Verdampfungstemperaturen (6 °C) und eine damit einhergehende hohe Entfeuchtungsleistung. Die Erweiterung eines KWS-Systems durch einen Kaltwasserspeicher bewirkt je nach Klima eine Verringerung des elektrischen Energiebedarfes um 10 % bis 12 %. Durch Installation einer Kühldecke anstelle der Luftkonvektoren ergeben sich energetische Einsparpotenziale von bis zu 30 %, allerdings bei Einschränkungen hinsichtlich der Behaglichkeit. Raumlufttechnische Anlagen mit DEC-System erreichen durch die nach unten begrenzte Zulufttemperatur (16 °C sollten nicht unterschritten werden) nur geringe Temperaturdifferenzen zwischen Zu- und Abluft. Der Kältebedarf muss somit durch hohe Luftvolumenströme gedeckt werden. Gleichzeitig wird die Zuluft geregelt entfeuchtet. Dies führt zu großem Energieaufwand durch den Betrieb der Ventilatoren für die Luftförderung. Das DEC-System kann jedoch dann vorteilhaft sein, wenn eine Anwendung ohnehin hohe Luftwechselraten und eine geregelte Entfeuchtung erfordert.

Die geringe Zahl von Vollbenutzungsstunden (ca. 460 h für das Szenario FM, entspricht einem Vollbenutzungsgrad von 5%) für den Betrieb von Kältesystemen in den betrachteten Klimaregionen, führen zu einem unerwarteten Bild hinsichtlich des Verhältnisses von direkten und indirekten TEWI-Emissionen (Abbildung 3). Aufgrund geringer Betriebszeiten der Anlage und dem daraus resultierenden Strombedarf, ist der indirekte Anteil der Emissionen, verglichen mit anderen Kälteanwendungen (z.B. Gewerbekälte), sehr klein. Besonders stechen hier die Ergebnisse des VRF-Systems hervor. Durch den hohen direkten TEWI-Anteil (ca. 70 %) spielt der Einfluss des Energiebedarfes (ca. 30 %) nur noch eine untergeordnete Rolle. Daraus folgt für Anlagen zur Kälteerzeugung in der Gebäudeklimatisierung, dass ein geringes GWP des eingesetzten Kältemittels sowie eine gute Anlagendichtheit entscheidend für die Klimawirkung sind. Die hier durchgeführten Berechnungen und Recherchen zeigen, dass natürliche Kältemittel wie Ammoniak und Propan mit ihren niedrigen GWP-Werten neben Vorteilen durch den geringen direkten TEWI-Anteil auch energetische Vorteile beim Einsatz in Kompressionskälteanlagen im Vergleich zu HFKW-Kältemitteln aufweisen.

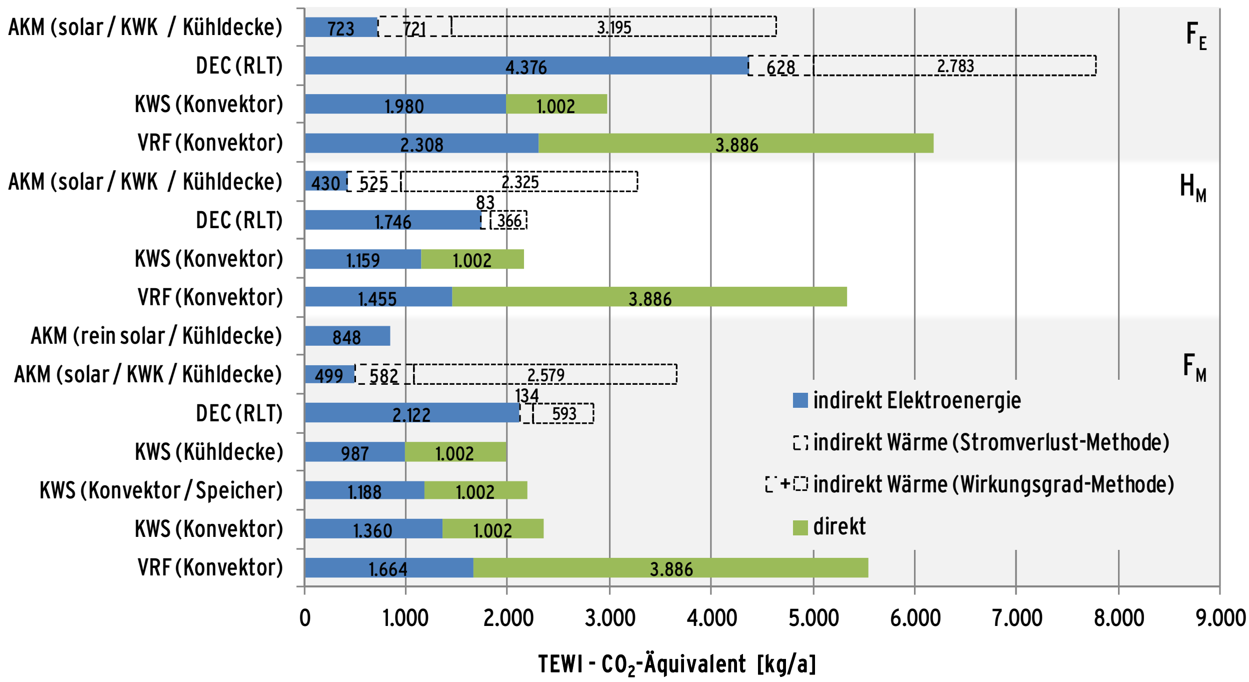


Abbildung 3: TEWI-Betrachtung für verschiedene Kälte-Systeme – Unterteilung in direkte und indirekte TEWI-Anteile (die Antriebswärme wird mit verschiedenen Allokationsmethoden berücksichtigt – siehe Abschnitt 4.2.10)

Thermische getriebene AKM-Systeme liefern je nach eingesetzter Wärmequelle sehr unterschiedliche Ergebnisse. Bei günstigen Bedingungen (Abwärme aus Erdgas-BHKW unter Berücksichtigung des Verdrängungsmixes) kann die Wärme mit einem TEWI von 0 g CO_2/kWh bewertet werden. Ein indirekter TEWI-Anteil entsteht in diesem Fall nur die notwendigen Hilfsenergien für Rückkühlung und Kälteverteilung, nicht jedoch durch die thermische

Antriebsenergie. Die Klimawirksamkeit dieses Systems ist dann im Vergleich zu den anderen Systemen minimal. Wird bei der Betrachtung die Nutzung von Fernwärme aus einem GuD-Kraftwerk vorausgesetzt (in Abbildung 3 „Stromverlust-Methode“) liefert die Wärme einen Beitrag zum indirekten TEWI. Die Summe aus direkten und indirekten Anteilen ist dann immer noch deutlich niedriger als bei den Vergleichssystemen. Wird die nötige Wärme allerdings aus einem Kohleheizkraftwerk bezogen (in Abbildung 3 „Wirkungsgrad-Methode“) ergeben sich sehr hohe TEWI-Werte, welche diejenigen der meisten Vergleichssysteme übertreffen.

Die Gegenüberstellung der Gesamtkosten untersuchter Kältesysteme für den Anwendungsbereich der Büroklimatisierung liefert ein breites Spektrum (Abbildung 4). In dieser Untersuchung schneiden die Systeme mit Kompressionskälte für den berücksichtigten Anwendungsfall wirtschaftlich deutlich günstiger ab als die übrigen Systeme. Das AKM-System verursacht über 50 % und das DEC-System über 130 % Mehrkosten im Vergleich zu den klassischen Kältesystemen. Den größten Einfluss haben hier die Investitionskosten. Diese übersteigen sowohl für das AKM-System, als auch für das DEC-System bereits die Gesamtkosten der Kompressionskältesysteme. Die Betriebskosten tragen nur zu einem relativ geringen Teil zu den Gesamtkosten bei. Selbst eine Betrachtung von Kostenvariationen aufgrund verschiedener Betriebskosten-Szenarien verändert dieses Verhältnis nicht wesentlich. Das Ergebnis liegt in der geringen Anzahl an Vollbenutzungsstunden begründet. Für alle Kältesysteme und betrachteten Klimaregionen liegt diese unter 800 h, was einem Nutzungsgrad[1] kleiner 10 % entspricht. Instandhaltungs- und Entsorgungskosten unterscheiden sich nicht wesentlich für die verschiedenen Kältesysteme und spielen daher bei einem Kostenvergleich keine ausschlaggebende Rolle.

Bei dem betrachteten Anwendungsbeispiel dominieren die Investitionskosten der Kältesysteme, da bei relativ geringer Vollbenutzungsstundenzahl die Betriebskosten entsprechend niedrig ausfallen. Alternative Kältesysteme wie AKM oder DEC-Systeme sind hier bei den derzeitigen Kostenstrukturen wirtschaftlich im Nachteil, da diese deutlich höhere Investitionskosten verursachen, die sich über geringere Betriebskosten nur über lange Zeiträume oder gar nicht amortisieren.

[1] Unter dem Nutzungsgrad ist hier der Quotient aus der Summe der erzeugten Kälteleistung pro Jahr zur maximal möglichen Kälteenergieerzeugung über das Gesamtjahr der Anlage zu verstehen.

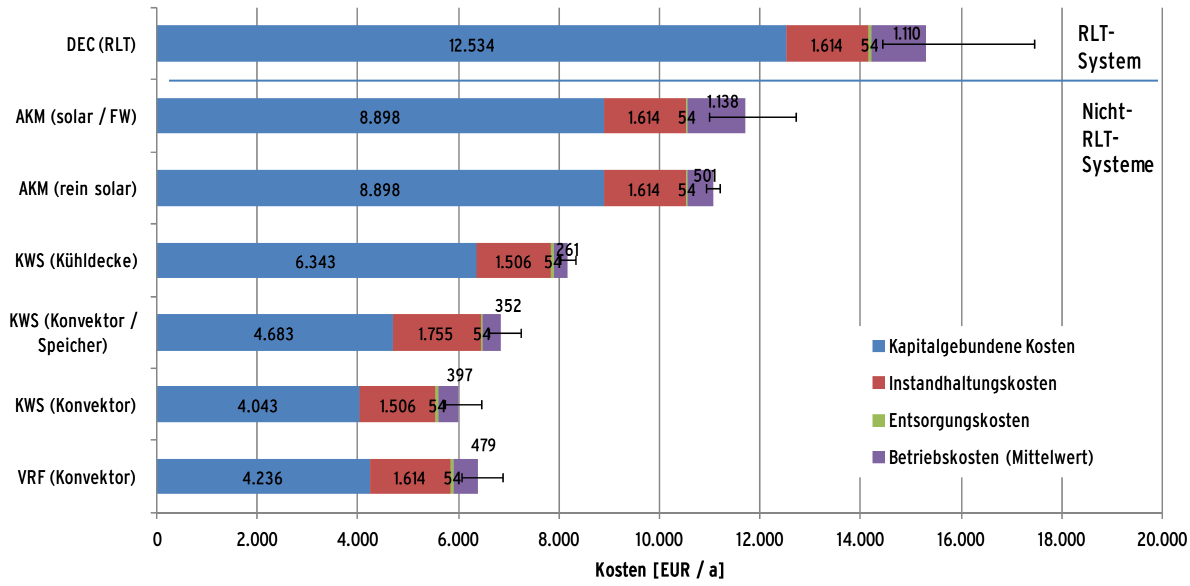


Abbildung 4: Jahresgesamtkosten untersuchter Kältesysteme für den Bereich Gebäudeklimatisierung – für die Betriebskosten ist jeweils ein Mittelwert für alle Energiekosten- und Klimaszenarien angegeben. Das Spektrum der Betriebskosten - verursacht durch die Betrachtung verschiedener Kostenszenarien – wird durch die Fehlerbalken ersichtlich.

Ergebnisse Industriekältebereitstellung

Wie schon bei der Gebäudeklimatisierung weisen die Kompressionskälte-Systeme auch bei der Industriekältebereitstellung den geringsten primärenergetischen Bedarf auf. Sowohl für direkt verdampfende als auch für Kaltwassersysteme bestehen feste Vorgaben hinsichtlich der Temperatur des Kälteträgers bei der Kälteübergabe. Hierdurch sind die Kaltwassersätze im Nachteil, da durch einen zusätzlichen Wärmeübergang an den Kälteträgerkreislauf niedrigere Verdampfungstemperaturen nötig sind.

Die Simulationsergebnisse zeigen, dass der Einsatz von natürlichen Kältemitteln energetische Vorteile bringen kann. Gegenüber R134a liefern die Kältemittel R290, R717 und R718 einen nahezu identischen Elektroenergiebedarf (Abweichungen kleiner 1 %). Vergleicht man die Werte mit dem Einsatz von R410A ergeben sich für die natürlichen Kältemittel Effizienzverbesserungen von etwa 3 %. Demgegenüber steht ein Mehrbedarf an Elektroenergie von über 10 % bei Einsatz von R1234yf. Datenblattangaben auf dem Markt verfügbarer Anlagen deuten darauf hin, dass Kältesystem, die mit natürlichen Kältemitteln betrieben werden, im energetischen Vergleich noch besser abschneiden als hier berechnet.

Die Kältebereitstellung mit der AKM ist aus elektroenergetischer Sicht im Vergleich zu den anderen Systemen herausragend effizient. Hinzu kommt jedoch bei der betrachteten Anwendung ein großer Bedarf thermischer Energie, welcher bis zu dem 30-fachen des elektrischen Energiebedarfs der AKM entspricht. Daher ergeben sich für Absorptionskälteanlagen erheblich höhere primärenergetische Bedarfswerte. Eine Ursache liegt in der nur noch marginalen Verbesserung des Wärmeverhältnisses von Absorptionskältemaschinen bei kleinen Temperaturhüben.

In Abbildung 5 sind die TEWI-Emissionen der Industriekältesysteme dargestellt. Die direkten Emissionen bei den Kompressionssystemen spielen eine weitaus geringere Rolle als bei der

Gebäudeklimatisierung. Ursache ist die hohe Zahl der Vollbenutzungsstunden. Unter den getroffenen Annahmen führt das direktverdampfende System mit dem Kältemittel R723 zu den geringsten TEWI-Emissionen. Auch bei Verwendung des fluorierten Kältemittels R134a lassen sich Vorteile gegenüber einem indirekt-verdampfenden KWS-System erreichen. Diese Ergebnisse sind jedoch stark vom Einsatzfall abhängig (z.B. Kältemittel-Leckraten).

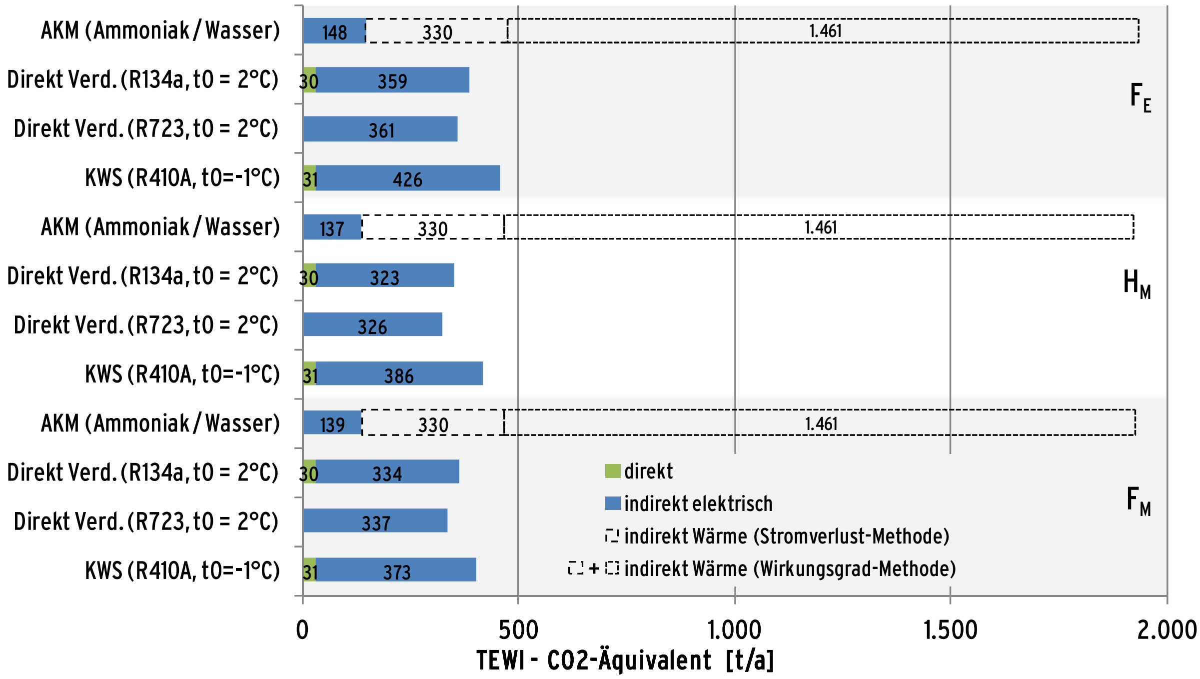


Abbildung 5: TEWI unterschiedlicher Industriekältesysteme unter unterschiedlichen Klimaszenarien

Der Wärmebedarf der Absorptionskältesysteme führt je nach Bewertungsmethode teilweise zu hohen TEWI-Werten. Die Nutzung von Fernwärme eines Kohle-Heizkraftwerkes stellt den ungünstigsten Fall der Betrachtungen dar, wobei diese mit der „Wirkungsgrad-Methode" bewertet wird. Dadurch ergeben sich Emissionswerte, die jene der Vergleichssysteme um mehr als das Fünffache übertreffen. Im günstigsten Fall stellt die Antriebswärme ein reines Abfallprodukt aus z.B. einem industriellen Fertigungsprozess dar oder es wird Abwärme eines BHKWs genutzt, dessen Strom den CO_2-intensiven Strom aus einem Kohlekraftwerk ersetzt (indirekter Anteil TEWI = 0 g CO_2/kWh). In diesem Fall zeigt sich das große Potenzial der Absorptionskälte zur Reduzierung der Treibhausgasemissionen. Hierbei liegen die Emissionen bei ca. 40 % im Vergleich mit den direktverdampfenden Systemen mit R723 bzw. bei ca. 33 % verglichen mit den Kaltwassersätzen. Hier zeigt sich, dass allgemeingültige Aussagen zur Klimawirkung der verschiedenen Kältesysteme kaum möglich sind. Für jeden individuellen Anwendungsfall sind bei der Wahl des Kältesystems detaillierte TEWI-Betrachtungen zu lokal verfügbaren Wärme- und Stromquellen durchzuführen, prinzipiell und auch differenziert nach Jahreszeiten.

Prozesskälte muss meist ganzjährig bereitgestellt werden. Hierdurch ergibt sich im Gegensatz zur Gebäudeklimatisierung ein hoher Anlagennutzungsgrad und damit ein umgekehrtes Verhältnis von Investitions- zu Betriebskosten. Die Dominanz der Betriebskosten wird in Abbildung 6 deutlich. Dadurch hat eine Variation der spezifischen Kosten für Strom, Wasser und Wärme einen erheblichen Einfluss auf die Gesamtkosten (angedeutet durch die Fehlerindikatoren in der Abbildung). Bei hohen Wärmebereitstellungskosten arbeitet ein AKM-

System wesentlich unwirtschaftlicher als vergleichbare Kompressionskältesysteme. Umgekehrt erlauben geringe bzw. keine Wärmebereitstellungskosten in Kombination mit einem wassersparenden Rückkühlsystem (z.B. hybrider Rückkühlturm) einen wirtschaftlicheren Betrieb des Absorbers. Instandhaltungs- und Entsorgungskosten haben bei dieser Betrachtung einen vernachlässigbar geringen Einfluss auf die Wirtschaftlichkeit der Systeme.

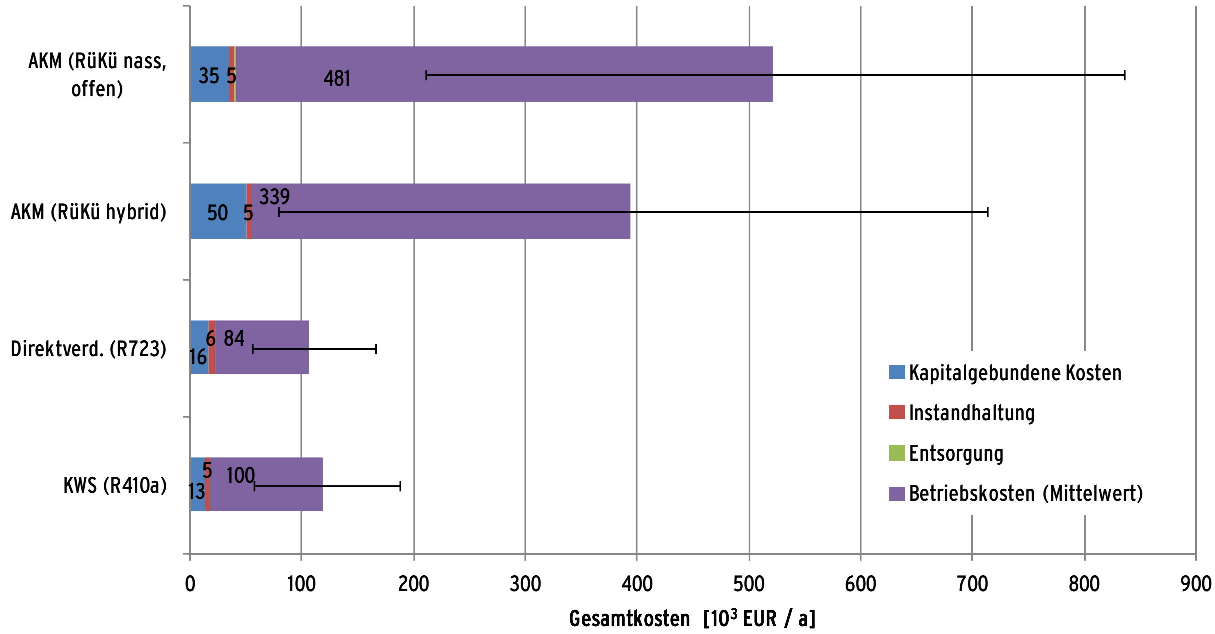


Abbildung 6: Jahresgesamtkosten untersuchter Kältesysteme für den Bereich Industriekälte – für die Betriebskosten ist jeweils ein Mittelwert zu allen Kosten- und Klimaszenarien angegeben. Das Spektrum der Betriebskosten - verursacht durch die Betrachtung verschiedener Kostenszenarien – wir durch die Fehlerbalken ersichtlich.

Schlussfolgerungen

Die Beispielrechnungen machen deutlich, dass generelle Aussagen zur Klimafreundlichkeit unterschiedlicher Kältesysteme schwierig sind. Ein Vergleich kann immer nur für einen speziellen Anwendungsfall mit den jeweiligen Systemparametern und Randbedingungen gelten. Allerdings erlauben die Ergebnisse folgende Schlussfolgerungen:

- Bei einer ungünstigen Auswahl bzw. Auslegung der Kältesysteme können sehr hohe TEWI-Werte zustande kommen. Die Vermeidung des Einsatzes von Klimakälte sollte bei der Projektierung von Gebäuden und Industrieanlagen oberste Priorität haben.
- Natürliche Kältemittel sind anderen Kältemitteln aus Klimaschutzgründen vorzuziehen.
- Die Berechnungen zeigen, dass von der Energieeffizienz nicht auf die Klimaverträglichkeit der einzelnen Systeme geschlossen werden kann. So gilt für Kompressions-Kältesysteme: die Leckrate im Kältekreislauf hat beim Einsatz fluorierter Kältemittel, mit hohem GWP, einen entscheidenden Einfluss auf den TEWI. Bei hohen Leckraten hat eine Effizienzsteigerung im Kältesystem (z.B. durch die Ergänzung mit einem Kaltwasserspeicher in der Gebäudeklimatisierung) nur einen geringen Einfluss. Deswegen ist den Leckraten, und den Verfahren zur Dichtheitskontrolle bzw. Leckagevermeidung, ein hoher Stellenwert beizumessen (Thema Leckagegrenzwerte, deren Überprüfung und Einhaltung in der Praxis).

- Die Nutzung ohnehin vorhandener und sonst ungenutzter Abwärme kann bei dem Einsatz einer AKM den TEWI deutlich reduzieren.
- Durch Systemkombinationen ließen sich Vorteile der verschiedenen Systeme gezielt besser nutzen: z.B. Verwendung der im Sommer nicht benötigten Abwärme durch eine Absorptionskälteanlage, Nutzung der hohen Effizienz einer Kompressionskälteanlage im Winterzeitraum (einschließlich freie Kühlung) oder auch die Nutzung von Absorptions-Kompressions-Kaskaden bei tiefen Kältenutztemperaturen.
- Einer weiteren Steigerung der Anlagendichtheit kommt eine hohe Priorität zu. Auch bei Einsatz der meisten natürlichen Kältemittel ist dies aufgrund der sicherheitstechnischen Aspekte relevant. Systeme zu automatischen Feststellung auftretender Lecks (über Prozessparameter) sind weiterzuentwickeln und in Systeme insbesondere mit größeren Füllmengen zu integrieren.

1.3 Substitution von Kompressionskältesystemen durch wärmegetriebene Verfahren

Die Ergebnisse aus dem Vergleich der Kälteerzeugungsverfahren zeigten die Möglichkeit der TEWI-Emissionsreduzierung beim Einsatz von Ab- und Adsorptionskälteanlagen in Verbindung mit bisher ungenutzten Abwärmeströmen.

Ausgehend von den möglichen TEWI-Emissionsminderungspotenzialen wurde untersucht, welche Anteile des Kältebedarfs durch wärmegetriebene Kältesysteme substituiert werden können. Hierbei wurden die Potenziale bei

- Nutzung vorhandener (Prozess-) Abwärme und
- Nutzung der Abwärme aus Erdgas-befeuerten KWK-Anlagen (vorzugsweise BHKWs)

untersucht.

Die Bewertung der Potenziale bezieht technische und wirtschaftliche Faktoren ein. Hierzu zählen:

- Temperaturniveau der Abwärmequelle
- Temperaturniveau der benötigten Kälte
- Saisonales und tageszeitliches Wärmeangebot, Kontinuität
- Saisonaler und tageszeitlicher Kältebedarf, Kontinuität
- Auslastung der Anlagen (Volllaststunden)
- weitere Hemmschwellen, wie z.B. zusätzlicher Platzbedarf
- Planungssicherheit (zukünftige Entwicklung des Kältebedarfs und der Wärmeverfügbarkeit)

Die Bewertung erfolgt für 25 Anwendungsbranchen der Industriekälte und für die Gebäudeklimatisierung. Aufgrund des hohen Anteils der Nahrungsmittelindustrie am Kältebedarf wird diese folgend separat dargestellt. Abbildung 7 stellt die ermittelten Potenziale zur Kältebedarfsdeckung mit wärmegetriebenen Kälteerzeugungsverfahren dar. Erkennbar sind die großen Potenziale insbesondere im Industriebereich. In diesem zeigt sich bereits heute eine vergleichsweise hohe Nutzung von wärmegetriebenen Kälteerzeugungsverfahren. Der hohe Anteil von Bestandsanlagen resultiert überwiegend aus der Chemieindustrie sowie aus

Betrieben, die bereits über eine Eigenstromerzeugung am Standort verfügen (z.B. Halbleiterindustrie).

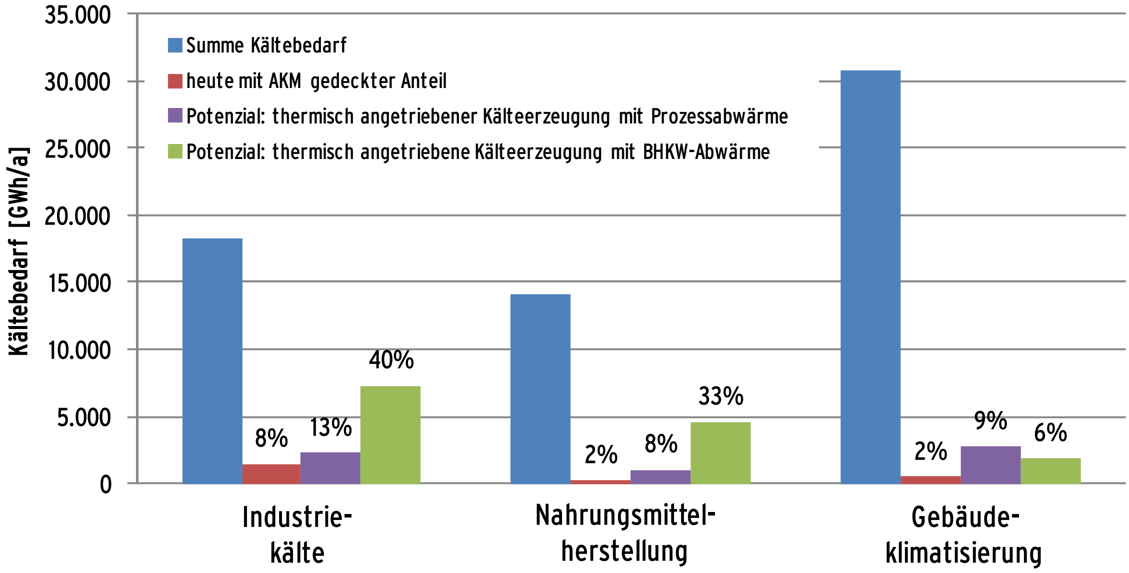


Abbildung 7: Potenziale zur Kältebedarfsdeckung mit wärmegetriebenen Kälteerzeugungsverfahren

Hinsichtlich der erreichbaren Elektroendenergieeinsparungen sind die Eigenverbräuche der wärmegetriebenen Systeme mit dem Mehraufwand für Pumpen und Rückkühlung zu berücksichtigen. Gegenüber guten Kompressionssystemen (Direktverdampfung mit R134a oder R717) im Industriebereich bzw. hocheffizienten Kaltwassersätzen in der Gebäudeklimatisierung werden Elektroendenergieeinsparungen von etwa 58 % im Industriekältebereich bzw. etwa 65 % in der Gebäudeklimatisierung erreicht (vgl. Kapitel 7.3.3.2 und 7.4.3.1).

Abbildung 8 fasst die erreichbaren Elektroendenergieeinsparpotenziale zusammen. Weitere Elektroendeinsparungspotenziale ergeben sich ausgehend von einer Reduzierung des Kältebedarfs, insbesondere im Bereich der Gebäudeklimatisierung. Die hohen ermittelten Potenziale hinsichtlich der Kältebedarfssubstitution in der Industriekälte finden sich im Elektroendenergiebedarf nicht wieder. Ursache ist der hohe Anteil der Gasverflüssigung am Elektroendenergiebedarf der Industriekälte. In diesem Prozess wird die Luft selbst zum Kältemittel, so dass dieser nicht durch wärmegetriebene Kältemaschinen ersetzt werden kann. Mit dem Einsatz von mehrstufigen Absorptionskältemaschinen sowie Neuentwicklungen auf dem Gebiet der Sorptionskälte (z.B. direktluftgekühlte Absorptionskältemaschinen) erschließen sich zusätzliche Potenziale zur Elektroendenergiesubstitution, welche hier nicht ermittelt wurden.

Die hieraus hervorgehenden TEWI-Minderungspotenziale – es wird nur der indirekte Anteil betrachtet – sind in Abbildung 9 dargestellt. Im Vergleich zu den indirekten Treibhausgasemissionen der gesamten stationären Kältetechnik entspricht die Summe des Minderungspotenzials von 1,6 Mio. t CO_2/a etwa 4 %, bezogen auf die betrachteten Anwendungen etwa 9 %.

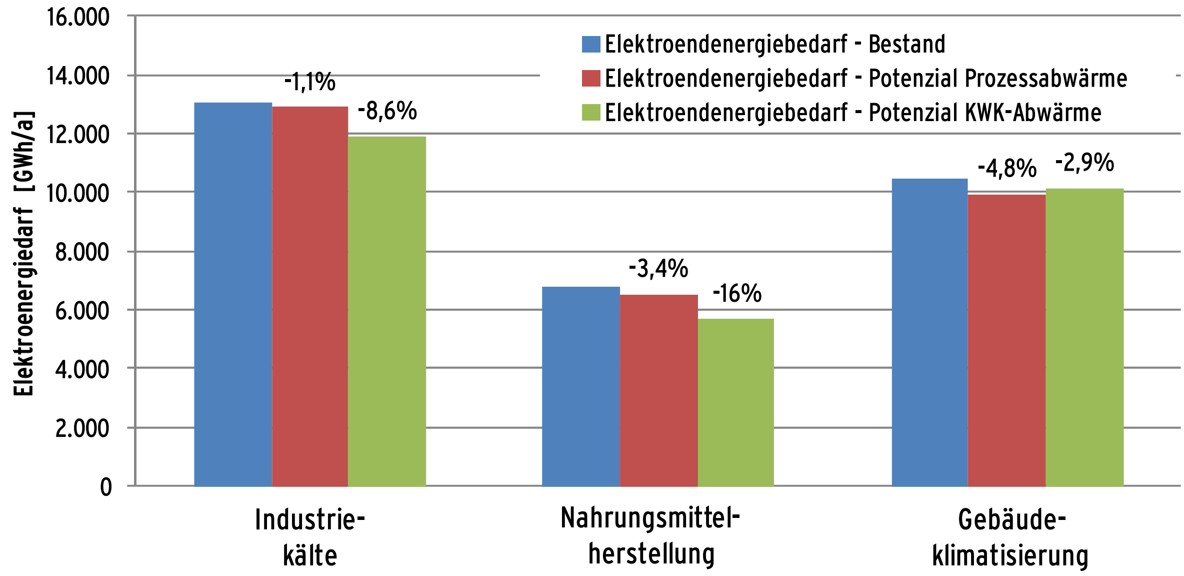


Abbildung 8: Elektroendenergieeinsparpotenziale durch den Einsatz von wärmegetriebenen Kälteerzeugungsverfahren (Stromeigenbedarf für Pumpen und Rückkühlung wurde berücksichtigt)

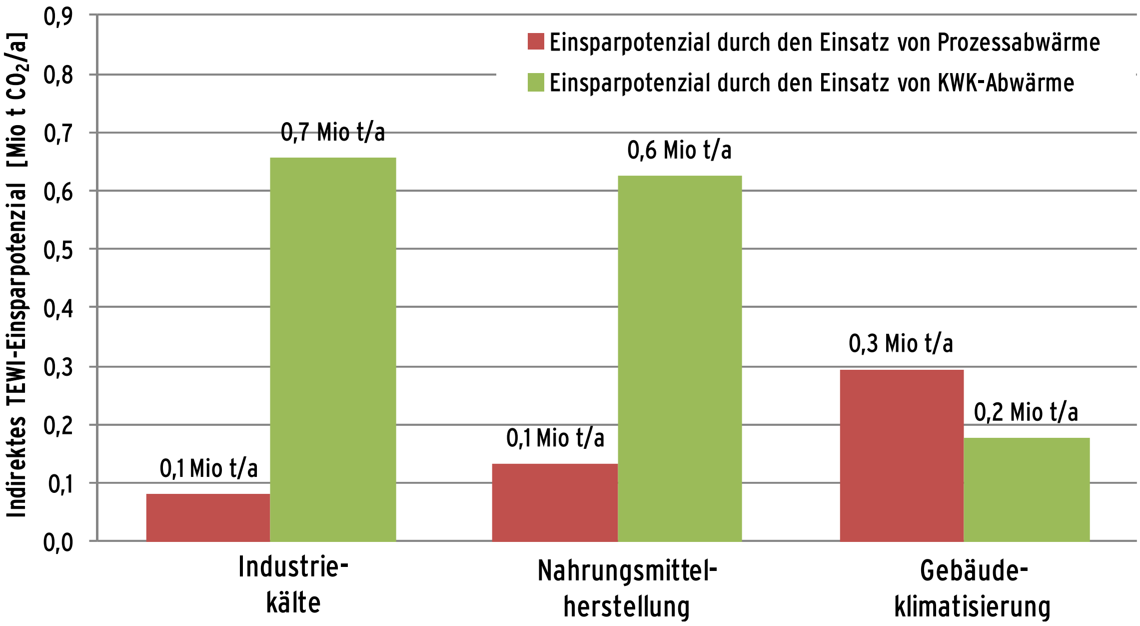


Abbildung 9: Indirekte TEWI-Minderungspotenziale durch den Einsatz von wärmegetriebenen Kälteerzeugungsverfahren (Stromeigenbedarf für Pumpen und Rückkühlung wurde berücksichtigt)

1.4 Hemmnisse und Handlungsempfehlungen

Die Vielfältigkeit nachhaltiger Kältesysteme lässt keine allgemein gültigen Aussagen zu Hemmnissen zu. Zu den einzelnen Bereichen lassen sich aber wichtige Kernaussagen treffen:

Für die Anwendung natürlicher Kältemittel gibt es nur wenige Anreize. Die entstehenden Mehrkosten durch notwendige Sicherheitstechnik bzw. aufgrund von Einzelanfertigungen werden nicht durch Kosteneinsparung durch eine mögliche höhere Energieeffizienz oder niedrigere Kältemittelkosten aufgewogen.

Die neue F-Gase-Verordnung (Verordnung (EU) Nr. 517/2014) enthält unter anderem eine zeitlich degressive Begrenzung der in den Verkehr zu bringenden Mengen an HFKWs (*phase down*). Die Begrenzung erfolgt hierbei auf Basis der CO_2-Äquivalente. Durch diese Maßnahmen ist mit einer erheblichen Verknappung und einer damit einhergehenden Verteuerung der HFKW-Kältemittel, insbesondere solcher mit hohem GWP-Wert, zu rechnen. Ergänzend wäre eine nationale GWP-gewichtete Besteuerung der Kältemittel nach den Vorbildern Skandinaviens oder Australiens denkbar.

Dem Einsatz wärmegetriebener Systeme stehen insbesondere die Hemmnisse hoher Investitionskosten, großes Bauvolumen, Ankopplung von Abwärmequellen sowie Informationsmangel zum Betriebsverhalten entgegen. Durch Förderung technischer Entwicklungen, wie z.B. kompakte Wärmeübertrager durch Oberflächenstrukturierung bzw. Oberflächenumwandlungsprozesse, Lösungen zu Absorptions-Kompressionskaskaden-Anlagen, alternativen Stoffpaaren und mehrstufigen Prozessen sind hier positive Ergebnisse hinsichtlich vermehrter Anwendung und wachsender Nachhaltigkeit im Sinne der energieeffizienten Nutzung von Wärme und Elektroenergie zu erwarten.

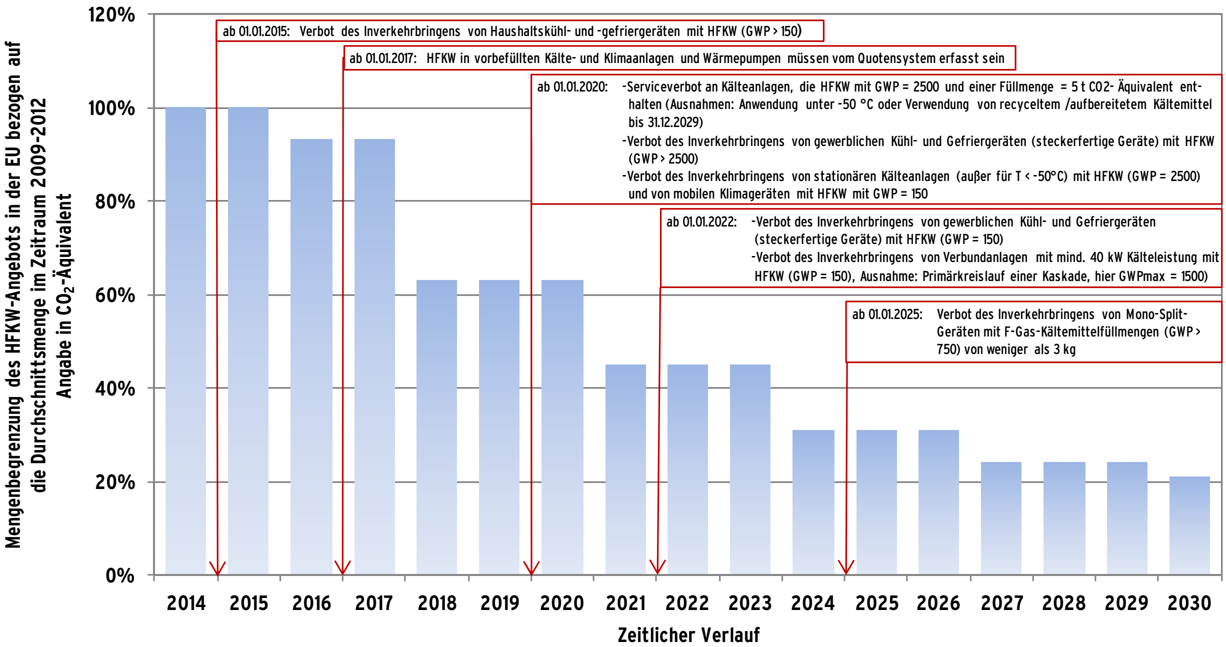


Abbildung 10: Zeitlicher Verlauf der Einschränkungen der HFKW-Mengen (quantifiziert über CO_2-Äquivalent) gemäß F-Gase-Verordnung (Verordnung (EU) Nr. 517/2014)

Im Bereich der Beauftragung der Kältesysteme wirkt zudem ein Informationsdefizit bei Planern bzw. Bauherren sowie bei Betreibern negativ auf die Marktdurchdringung alternativer Systeme. Hier sind Alternativen selten bekannt, zudem wird das Thema Kälte/Kühlung oft zu spät in die Planung einbezogen, so dass Systeme wie die Betonkernaktivierung nicht mehr als Möglichkeit in Frage kommen. Der planer- und betreibergerechten Information kommt hierbei besondere Bedeutung zu.

Die hohen Investitionskosten von effizienten und nachhaltigen Kältesystemen sind auch als Geschäftsmodell für den Finanz- und Energiesektor zu verstehen (Energie-Contracting, Energieeinspar-Contracting). Aufgrund steigender Energiepreise sind hier verschiedene Geschäftsmodelle denkbar, welche für beide Seiten, also sowohl den Betreiber als auch den Investor, von Vorteil sind.

Mit dem Impulsprogramm für gewerbliche Kälteanlagen liegt bereits ein Investitionsförderprogramm vor, welches mit bis zu 35% Förderquote bezogen auf die Gesamtinvestitionen des Kältesystems gezielt Anlagen mit natürlichen Kältemitteln, Wärmerückgewinnung sowie Komponenten auf dem aktuellen Stand der Technik fördert. In den Jahren 2009 bis 2011 wurden hierbei bereits 244 Anlagen mit einer installierten Kälteleistung von etwa 144 MW gefördert. Bei etwa 40% der geförderten Anlagen handelt es sich um Anlagenmodernisierungen. Bezogen auf den Fördermitteleinsatz lagen die CO_2-Vermeidungskosten bei 35 €/t CO_2-Äquivalent (Jörß & Klose 2012).

2 Summary

Refrigeration systems require in sum approx. 14 % of the total electrical energy demand in Germany and they are responsible for approx. 5 % of all greenhouse gas emissions. Regarding the national aims for climate protection, these numbers clearly show the necessity for more climate-friendliness in air handling and refrigeration technologies. The present study provides a base for activities in order to improve the climate-friendliness of these technologies and presents investigations in the fields of industrial refrigeration and building air conditioning.

The study is organised as follows: at first, the cooling demand itemised to industrial sectors is shown in terms of presently employed technologies. These are characterised by their required energy and some other properties, which are necessary for later analyses and evaluations. Second, an overview of climate friendly refrigeration systems is provided with a supplementary review of the availability of those systems and single components on the market. Third, by means of dynamic simulations for an entire year, two special cases of different systems and their variants are investigated. Based on the results it is possible to draw conclusions related to the climate friendliness of the systems under consideration and discuss the influence of certain factors on it. Fourth, given heat sources allow to employ thermally driven refrigeration and air conditioning systems yielding a reduction of primary energy and hence greenhouse gas emissions. The potential for an increased usage of thermally driven refrigeration systems is presented and discussed. Finally, barriers and tools are analysed which possibly prevent and promote the establishment of climate friendly refrigeration systems on the market, respectively.

2.1 Net and final energy demand, and refrigeration technology's relevance for the climate

Stationary refrigeration and air conditioning in Germany required in sum 70 TWh electrical energy in 2008 (Guntram Preuß 2011), corresponding to a share of 13.5 % of the final electrical energy demand (Umweltbundesamt 2012a). The relevance for the climate can be measured in terms of the total equivalent warming impact (TEWI), consisting of a direct share due to refrigerant emissions and an indirect one due to the energy consumption of the system. Given the specific CO_2 emissions for electrical energy of 568 g CO_2/kWh_{el} (Umweltbundesamt 2012a) in 2008, the refrigeration technology produced approximately 40 Mt (Mt: million metric tons) of indirect CO_2 emissions. Supplementary to those, additional 5.6 Mt CO_2 equivalent were released by direct refrigerant emissions (Becken & Plehn 2010). Details can be found in Table 2. Due to large growth rates of new plants in the sector air conditioning in buildings, rising emissions can be expected for the future.

Figure 11 shows the final electrical energy demand of the refrigeration technology itemised by industrial sectors (Guntram Preuß 2011). Of those sectors, industrial refrigeration, food industries, and stationary air conditioning are relevant for the present study. In Germany, these three sectors have a cooling demand of 63,173 GWh/a (Figure 11) which requires 30 TWh/a electrical energy (5.7 % of total electrical energy demand in Germany) and 3 TWh/a heat (Figure 12). The basis for these numbers are provided in Chapter 5 from page 40 on. A detailed itemisation with respect to industrial sectors will be presented on page 55.

Table 2: Climate-relevant data of stationary refrigeration technology in Germany

Characteristic number	Value	Comment
Electrical energy demand of stationary refrigeration technology in Germany	70,695 GWh/a	Data base 2008, (Guntram Preuß 2011),
Share of final electrical energy demand in Germany	13.5 %	Data base 2008, (Guntram Preuß 2011; Umweltbundesamt 2012a)
Greenhouse gas emissions of stationary refrigeration technology in Germany	45.8 Mt	Data base 2008 for electrical energy (Umweltbundesamt 2012a), Data base 2007 for direct refrigerant emissions (Becken & Plehn 2010)
Share of final greenhouse gas emissions in Germany	4.7 %	Data base 2008
Indirect emissions (electrical energy demand)	40.2 Mt (81.6 %)	
Direct emissions (greenhouse effect due to refrigerant)	5.6 Mt (18.4 %)	
Electrical energy demand of industrial cooling and building air conditioning	30,244 GWh/a (42.8%)	Share value based on final electrical energy demand of refrigeration technology in Germany

Stationary refrigeration and air conditioning technologies, except commercial refrigeration, are responsible for approximately 800 t direct HFC emissions (Becken & Plehn 2010), corresponding to 1.6 Mt CO_2 equivalent. Considering the emissions of previous years, one can observe an increase of emissions by 15 to 20 % per year between 2002 and 2007.

Due to the climatic conditions in Germany, the air conditioning sector has only few full-load hours. The guideline for energy performance certificates for non-residential buildings (BMVBS 2007) declares 500 h for office and 350 h for educational buildings. The simulations carried out in this study yielded full-load hours of approximately 400 to 520 h. Within the framework of the EvaSolK project, different air conditioning cooling devices, such as chillers, mono-, and multi-split compression- type refrigeration plants (CRP), have been investigated in practice and yielded full-load hours between 170 and 430 h (Wiemken, Safarik, et al. 2012; Wittig et al. 2012). Beside the climatic conditions, there is another reason for the low utilised capacity: security factors and – more importantly – prospective extensions of buildings lead to oversized air conditioning systems.

Given the small number of full-load hours, the share of direct emissions on the TEWI is significant, considering refrigeration plants with leakage rates of 4 to 7 % (Schwarz et al. n.d.). This observation can be confirmed by the present simulation results. For instance, the application of R410A in chillers leads to a TEWI direct emission share of up to 40 %, depending on the climate zone. For R134a it is 32 %, and for R32 it is 18 %. In other words, up to 40 % of the greenhouse effect-inducing emissions are caused by the leakage of the refrigerant, whilst the rest comes from the energy being required. In direct evaporating systems, the share is even greater due to the larger refrigerant charge. With the assumptions stated above, the simulations with R410A exhibit a direct share of up to 70 %. Opposing to the afore mentioned refrigerants, the HFC R1234yf and the natural refrigerants R290, R717, and R718 exhibit an almost negligible share of direct emissions, compared to the indirect ones.

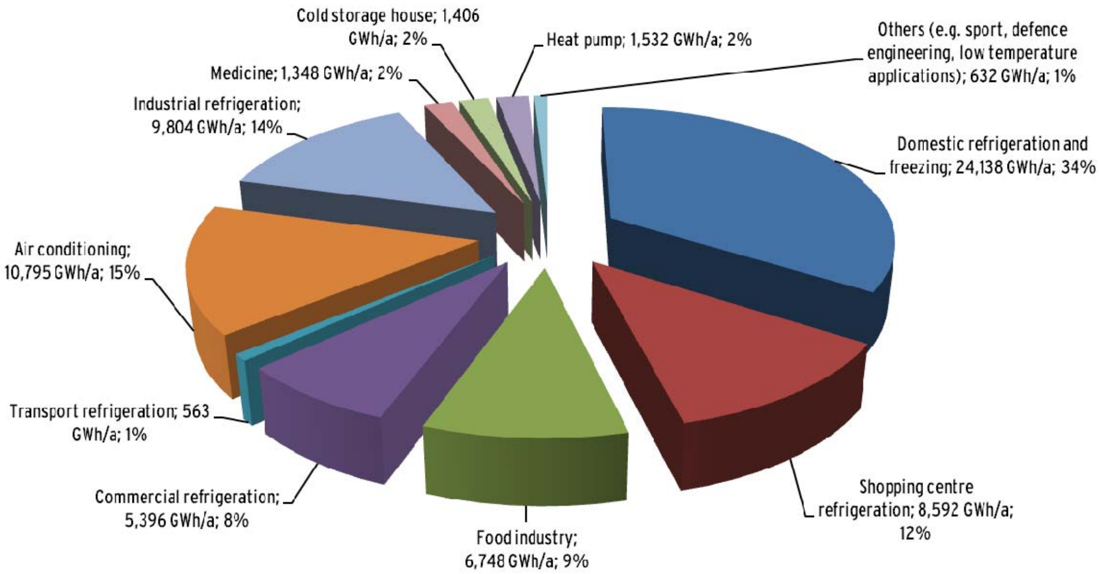


Figure 11: Share of refrigeration electrical energy demand of various industrial sectors (Guntram Preuß 2011)

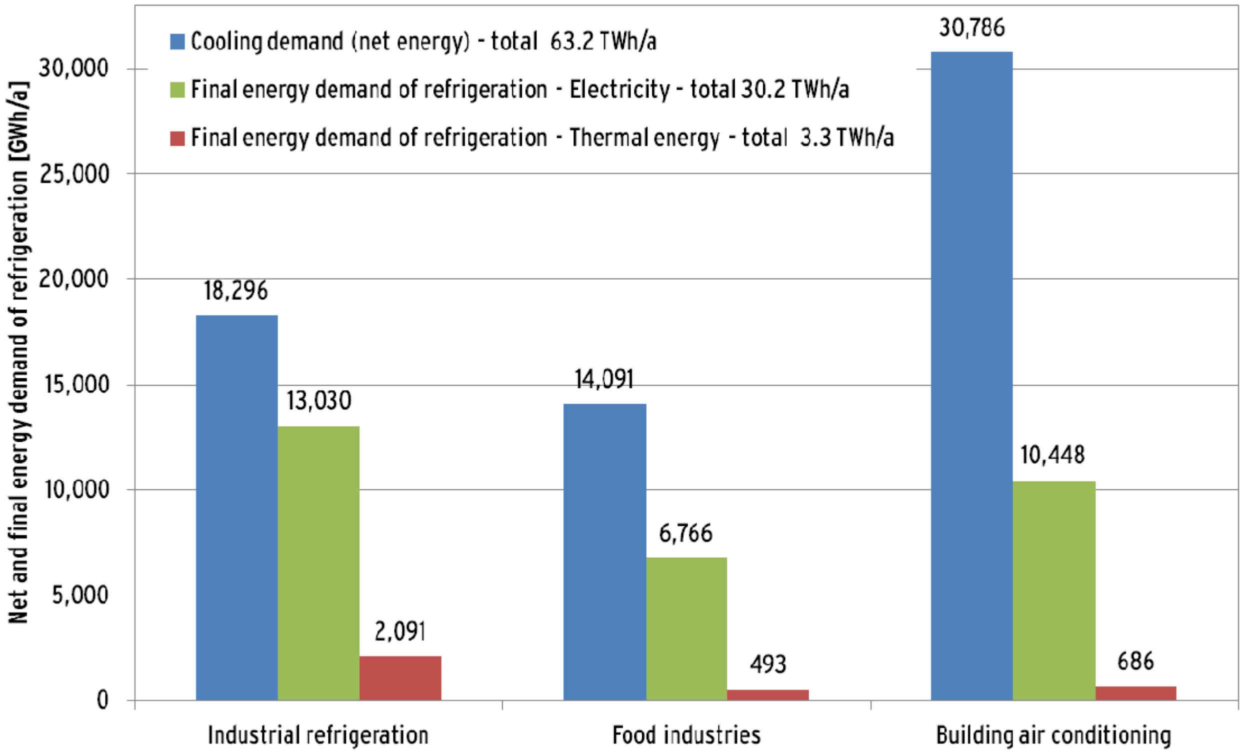


Figure 12: Cooling demand and final energy demand of refrigeration of particular sectors

With a better degree of capacity utilisation, it is possible to reduce the direct shares. For instance, simulations of industrial chillers being in operation the entire year showed a direct share below 10 %. However, despite the relatively small percentage, the absolute values can be significant, especially in large-scale plants and if refrigerants with a large GWP are applied.

2.2 Evaluation of various refrigeration and air conditioning techniques

Aim of this section is to evaluate different techniques for refrigeration and building air conditioning with respect to ecological and economical aspects, whereby the impact on the climate is of superior importance. Therefore, energy consumption and emissions of different refrigeration systems are determined, compared with each other, and employed for the simulation of building air conditioning and industrial cooling later on.

The basis for the evaluation are dynamical simulations of a non-residential building with its air conditioning system. Simulations cover an entire year in two climate regions (Frankfurt and Hamburg) utilising different climate scenarios (average and extreme summer). For each case, the final energy demand and the greenhouse gas emissions are calculated by using the TEWI approach. The building is modelled as an open-plan office with a squarish floor space of 400 m^2 and a thermal load of 30 W/m^2, which is a typical value for an office. The systems engineering conforms the state of the art (e.g., condensation pressure limited corresponding to saturation temperature of 15 °C using electronic expansion valves, minimal chilled water supply temperature of 10 °C). The following systems are considered:

- System with a solar- or district-heat-driven absorption chiller (AbCh) and a chilled ceiling for heat transfer
- Variable refrigerant flow system (VRF) with fan coils for heat transfer
- Compression-type refrigeration plants (CRP), (chillers, condensation temperature > 15 °C) with chilled ceiling or fan coils for heat transfer and with/without cold water storage
- Heating ventilation and air conditioning (HVAC) system with liquid sorption (DEC: desiccative and evaporative cooling) employing heat from a combined heat and power plant (CHP)

For investigations of industrial cooling, a constant cooling demand of 500 kW at a refrigerant supply temperature of +2 °C is assumed for an entire year. The following systems are considered:

- Chiller (Ch) with a compression-type refrigeration plant (CRP)
- Direct evaporation (also called direct expansion (DX)) CRP system
- Absorption chiller (ACh) driven by heat from a combined heat and power plant (CHP)

An energetic comparison of the systems is carried out with the final energy demand. Considering the energy content of the energy sources and carriers being employed, one can calculate the demand of primary energy sources. The factors for the primary energy sources are taken from the draft of EnEV 2014 and DIN V 18500-1:2011-12.

In order to evaluate the relevance for the climate of the above-mentioned systems, the TEWI is calculated by using 583 g CO_2/kWh (electricity mixture in Germany of 2010, including all losses) and three different values for thermal energy: 0, 44, and 239 g CO_2/kWh. Heat from the environment is assumed to produce no CO_2. In addition to the indirect contributions to TEWI, direct ones due to leakage of the refrigerant are considered depending on the system with 4 to 7 % and disposal losses of 30 %.

The examinations provide also economical comparisons of the variants above, containing investment, working, maintenance, and disposal costs. In order to calculate the cost of operation, different scenarios are included with prices for electricity of 6 to 25 ct/kWh and for heat of 0 to 8 ct/kWh.

For the case of building air conditioning, the aspect of thermal comfort is also considered.

Results for building air conditioning

Regarding comfort, systems with fan coils yield the best results, whereas chilled ceiling systems are not able to provide sufficient cooling power without condensation at the cold areas on humid and hot days. The humidity control is not possible with chilled ceiling systems. DEC systems proved to be able to control humidity, but leading to exceeding temperatures on hot days, due to a lower limit of supply air temperature restricting the cooling power. Solar refrigeration systems with ACh can be used only with realistically sized collector areas and hot water storages, with either including an additional heat source (e.g., district heating) or with a compression-type refrigeration system as backup. For cases of pure solar cooling, one can find exceeding temperatures often. An energetic and economic comparison with other systems is therefore of limited relevance only.

Neglecting pure solar cooling, CRP systems require the lowest amount of primary energy and the VRF system the largest one, which is due to the relatively low evaporation temperatures (6 $^\circ$C) and the resulting large desiccative power. An extension of the chiller by a cold water storage reduces the electrical energy demand by 10 to 12 %, depending on the climate. Implementing a chilled ceiling instead of fan coils reduces the energy demand by up to 30 % but with limitations regarding comfort. HVAC systems with DEC offer only small temperature differences between supply and outlet due to the limit of the supply temperature (>16 $^\circ$C) with a controlled desiccation. Hence, it is necessary to provide large volumetric flow rates, leading to a large energy demand for the fans. However, such a system might be of advantage in applications which already require large flow rates and controlled desiccation.

The small number of full-load hours (approx. 460 h for the scenario Frankfurt corresponds to a full-load of 5 %) for the operation of refrigeration systems in the observed climate regions, lead to an unexpected picture in terms of the ratio of direct and indirect TEWI emissions (Figure 13). Due to the small number of operating hours of the facility and the resulting power consumption, the share of indirect emissions, compared with other cooling applications (e.g. commercial refrigeration), is very small. Particularly remarkable are the results of the VRF system. Due to the high direct TEWI share (approx. 70 %), the influence of the energy demand (about 30 %) plays a minor role only. As a conclusion, low GWP of the refrigerant as well as a good plant tightness are crucial for the assessment of climate impacts of refrigeration systems for building air conditioning. The performed calculations and research show that natural refrigerants such as ammonia and propane with their low GWP values have advantages by their low direct TEWI share and may also have energetic advantages when used in compression-type refrigeration systems compared to HFC refrigerants.

Thermally driven ACh systems produce very different results depending on the heat source. In good conditions (waste heat from CHP, considering the mixture of substituted fuel) it is possible to obtain heat with a TEWI of 0 g CO_2/kWh. An indirect share exists only for auxiliary energy such as re-cooling and cooling distribution, but not for the heat source itself. The climate impact of such systems is minimal compared to other ones. Employing district heat from a gas-steam power plant ("electricity loss method" in Figure 13) leads to an indirect TEWI share. However, the sum of direct and indirect shares is still lower than the sum shares of comparable systems. The opposite is the case when the heat is provided from a coal-fired power station, where extremely large indirect TEWI shares can be observed, exceeding most of the comparable systems ("efficiency factor method" in Figure 13).

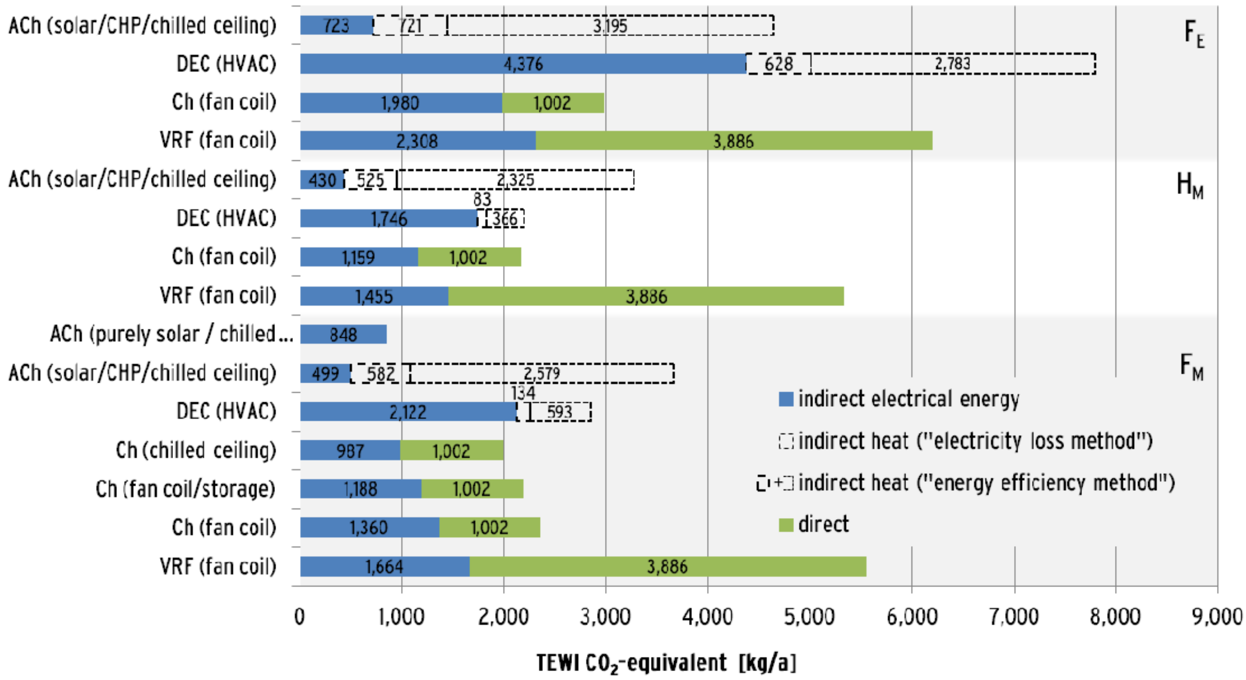


Figure 13: TEWI considerations for various refrigeration systems; itemised by direct and indirect TEWI shares (driving heat given by different allocation methods; see section 4.2.10)

A comparison of the total costs of the considered air conditioning systems reveals that they differ over a wide range (Figure 14). CRP systems are economically better than the other systems for the given case. ACh require over 50 % and DEC systems even 130 % more of the costs of classical refrigeration systems. The major part of the costs are the investment cost, which exceed for the case of ACh and DEC systems the total costs of compression-type refrigeration systems. Operation costs have a small share of the total costs only, even with different cost scenarios. The reason for this is, again, the small number of full-load hours. All refrigeration systems in the climate region under consideration exhibit operation times below 800 h, representing a degree of utilisation below 10 %. Maintenance and disposal cost do not differ significantly and are of minor importance in the comparison.

In this particular case of study, the investment costs of the refrigeration systems dominate all other costs due to low operation costs. Alternative systems, such as ACh and DEC systems are at a disadvantage due to their larger investment costs, which amortise only over long time periods or even never due to lower operation costs.

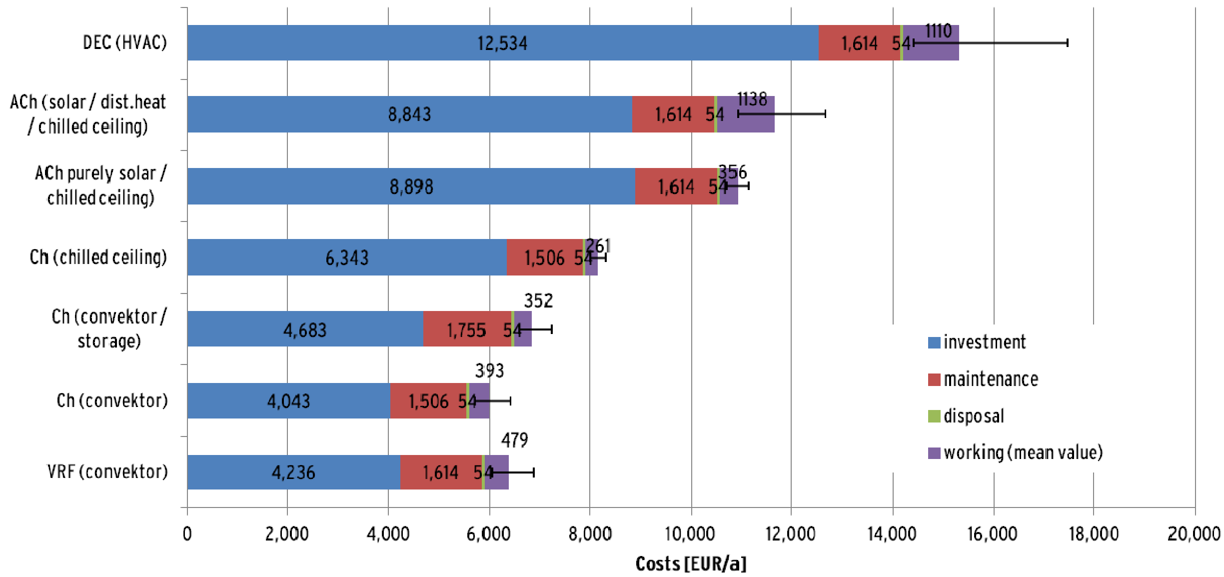


Figure 14: Final annual costs for air conditioning systems; operation costs are given by the mean value for different scenarios. Their variability is indicated by error bars.

Results for industrial cooling

As already observed for the building air conditions, compression-type refrigeration systems have the lowest primary energy demand in industrial cooling sector. Both, direct evaporation and chiller systems, have to provide the same temperatures level at the point where heat energy is absorbed. Thus, chillers have an energetic disadvantage since the additional heat transfer to the refrigerant cycle requires lower evaporation temperatures.

Simulation results showed that natural refrigerants exhibit energetic advantages. Compared to R134a, the refrigerants R290, R717, and R718 require almost the same amount of electrical energy (differences below 1 %). Compared to R410A, natural refrigerants have an increased efficiency by 3 %. In contrast to this, R1234yf requires 10 % more electrical energy. Data sheets of systems on the market indicate that refrigeration systems with natural refrigerant are even better in an energetic comparison than calculated here.

Cooling with ACh is extraordinarily efficient compared to other systems from an electro-energetic point of view. However, for the given application, ACh require up to 30 times more thermal energy than their electric energy demand. Hence, ACh have a much larger primary energy demand. The reason for this is that the heat ratio can be improved only in a limited range in the case of small temperature differences.

Figure 15 illustrates the TEWI emissions for industrial refrigeration systems. The direct emissions of the CRP are of less importance than it was for building air conditioning. The reason is the large number of full-load hours. With the assumptions been made, the direct evaporation system with R723 has the lowest TEWI emissions and the application of the fluorinated R134a has advantages over indirectly evaporating chillers. These results, however, depend strongly on the particular case (e.g., refrigerant leakage rates).

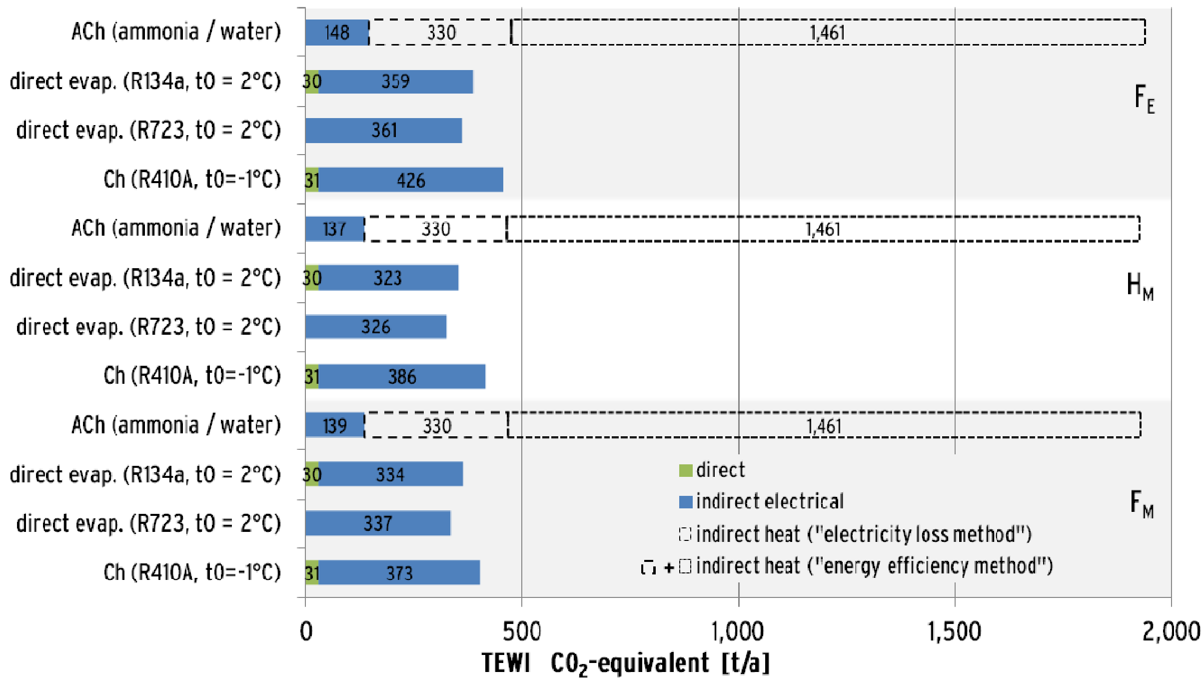


Figure 15: TEWI of industrial refrigeration systems and various climate scenarios

The heat demand of ACh can lead to large TEWI values, depending on the heat source. Hereby, district heat from a coal-fired power station is the worst scenario leading to five times larger TEWI than of comparable systems, evaluated with the "efficiency factor method". In the best case, the heat source is, e.g., some waste heat from industry or a CHP, whose electrical power is employed to substitute CO_2-intensive power from a coal-fired power station (indirect TEWI: 0 g CO_2/kWh). In the latter case, ACh exhibit their full potential to reduce greenhouse gas emissions. The emissions are approximately 40 % of the emissions from direct evaporation R723 systems and 33 % of those from chillers. However, this comparison also shows that a general statement regarding the climate impact of different systems is hardly possible. Moreover, for each individual application it is necessary to conduct detailed TEWI investigations considering refrigerants, heat and electrical power sources, and also, in principle, the time of the year.

Process cooling is usually required during the entire year, which leads to a large degree of utilisation and to a reciprocal ratio of investment to operation costs – in contrast to building air conditioning. The dominating effect of the operation costs is visible in Figure 16, which leads to an increased influence of the variability of the costs for electricity, water, and heat on the total costs (indicated by error bars). Large costs for heat make ACh less economical than comparable CRP, but otherwise at low or negligible heat costs in combination with water-saving re-cooling systems (e.g., hybrid cooling tower), ACh are at an advantage. The costs for maintenance and disposal have a negligible effect on the cost effectiveness of the systems.

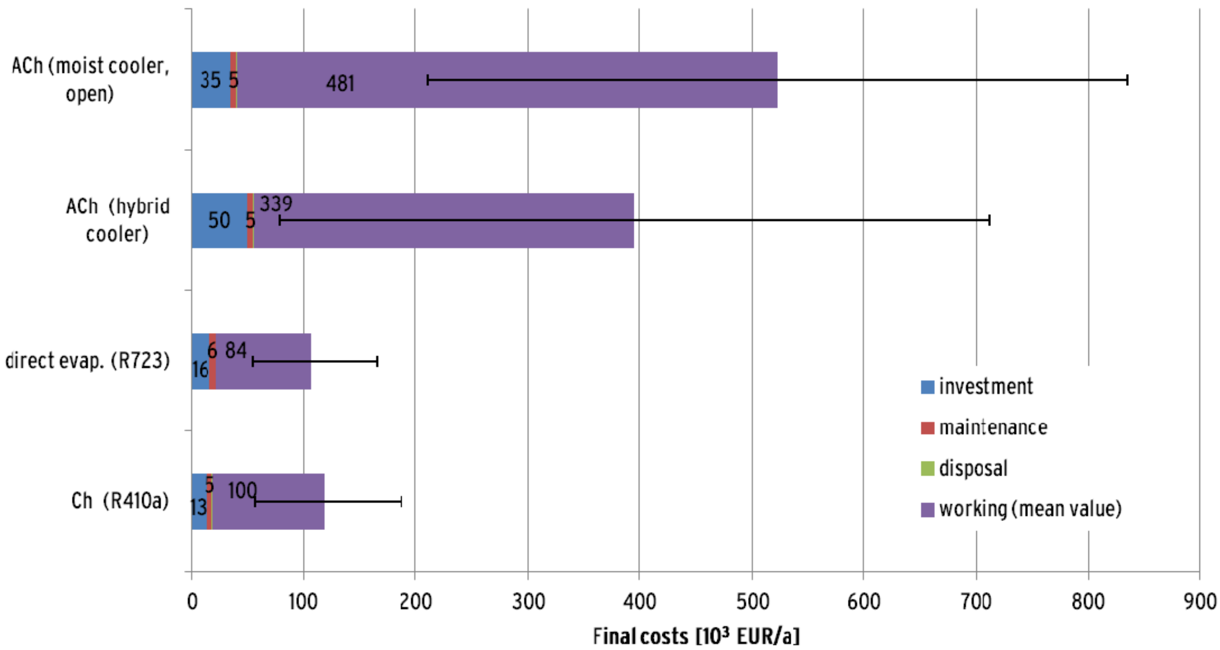


Figure 16: Final annual costs of industrial refrigeration operation costs are provided by the mean value for different scenarios. Their variability is indicated by error bars.

Conclusions

The simulations demonstrated that a general statement regarding the climate friendliness of different refrigeration systems is difficult. A comparison is possible only for a particular case with its parameters and boundary conditions. However, the results allow the following conclusions:

- With an adverse choice/design of the refrigeration system, extremely large TEWI values can occur. Avoiding air conditioning cooling should have the highest priority when designing new buildings or industrial plants.
- Natural refrigerants shall be favoured over others due to climate protection reasons.
- Good energy efficiency does not necessarily lead to good climate protection. E.g., CRP with fluorinated refrigerant of large GWP: refrigerant leakage increases TEWI; at large leakage rates other improvements of the system efficiency (e.g., cold-water storage) have only small effect. Hence, tightness, leakage rate, and avoiding leakages are very important.
- Utilisation of presently unused waste heat can decrease the TEWI significantly with ACh.
- Combinations of systems to create advantages: e.g., utilisation of waste heat with an ACh in the summer, and employing large efficiency of CRP in the winter (incl. free cooling) or utilisation cascades of absorption-compression at low temperatures.
- An increased tightness of the system has large priority; also for natural refrigerants for security purposes. Development of automatic devices asserting leakages (via process parameters) and implementation into large systems

2.3 Substitution of compression-type refrigeration by heat-driven systems

The results of the comparison of different refrigeration techniques showed the potential to reduce the TEWI of ACh in conjunction with presently unused waste heat. Based on the potential to reduce the TEWI, the share of systems which can be substituted by ACh is studied by means of the utilisation of

- Present (process) waste heat, and
- Waste heat of natural gas fired CHP.

The evaluation of the potential considers technical and economical factors. These are:

- Temperature level of waste heat
- Temperature level of required cooling
- Seasonal and temporal heat supply, continuity
- Seasonal and temporal cooling demand, continuity
- Utilisation demand of the system (full-load hours)
- Other inhibitions, e.g., additional floor area demand
- Planning reliability (prospective cooling demand and heat supply)

The evaluation is carried out considering 25 sectors of industrial refrigeration and building air conditioning. Owing to the large food industry share of the cooling demand, it is demonstrated separately. Figure 17 presents the potentials to cover the cooling demand by heat-driven techniques. One can see that there are significant potentials in the industrial sector, where a relatively large share of heat-driven systems is installed already today. A large share of these systems can be found in chemical industries and factories having their own electrical power supply (e.g., semiconductor industries).

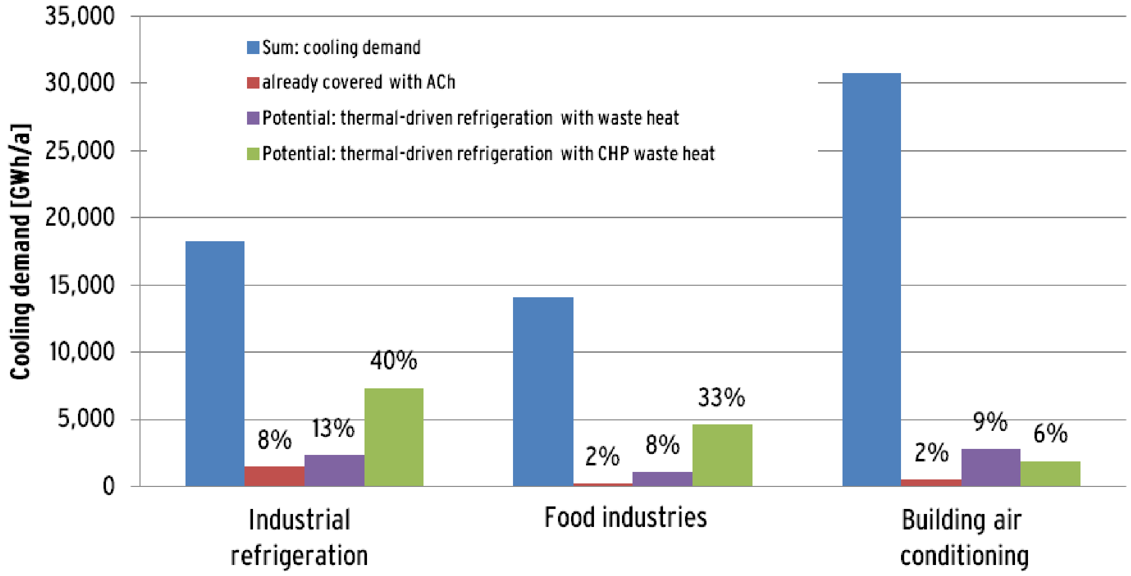


Figure 17: Potentials for covering cooling demand by means of heat-driven refrigeration techniques

Regarding the savings of electrical energy, the consumption of auxiliaries in heat-driven devices (pumps, coolers) has to be considered. Compared to good CRP (direct evaporation R134a or R717 systems) in industry or extremely efficient chillers in building air conditioning, electrical energy savings of up to 58 % in industry and 65 % in building air conditioning can be obtained (see sections 7.3.3.2 and 7.4.3.1).

Figure 18 summarises the potentials of saving electrical energy. The large potentials regarding the substitution of refrigeration demand in industry do not reflect in the demand for electrical energy. The reason for this is the large share of gas liquefaction of the total demand for electrical energy in industrial refrigeration. Hereby, air itself is the refrigerant, which is the reason why this process cannot be substituted by a heat-driven system. The resulting indirect TEWI reduction potentials are displayed in Figure 19.

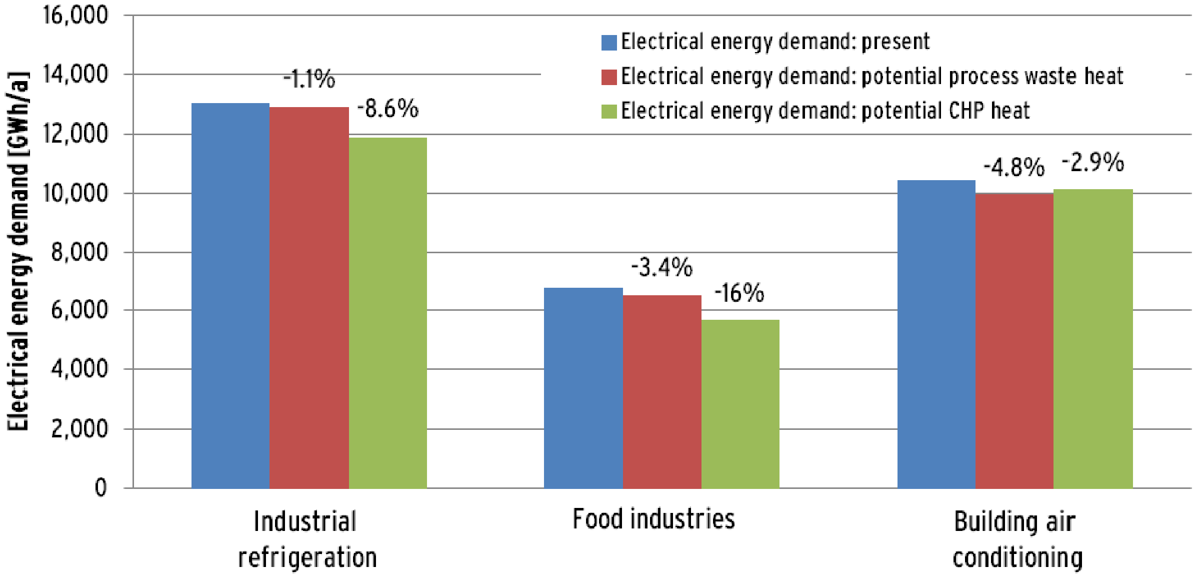


Figure 18: Potentials to save electrical energy by means of heat-driven refrigeration systems (auxiliary energy of pumps and cooler are taken into account)

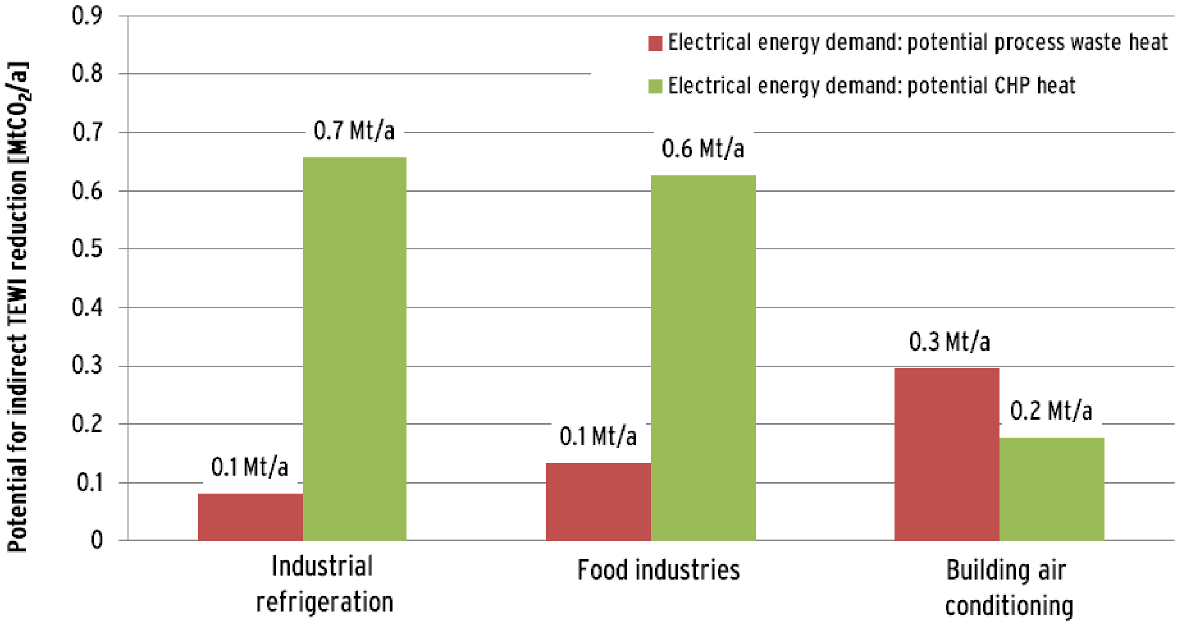


Figure 19: Indirect TEWI reduction potentials by means of head-driven refrigeration systems (auxiliary energy of pumps and cooler are taken into account)

2.4 Barriers and recommendations

The variability of climate friendly refrigeration systems does not allow a general statement regarding barriers. However, it is possible to highlight some important points:

For the application of natural refrigerants, only few stimuli exist. The increased costs for the required security installations and for custom-made items are not compensated by possibly larger energy efficiency or lower costs for refrigerants. The new Regulation (EU) No 517/2014 on fluorinated greenhouse gases comprises, amongst others, the gradual reduction of the quantity of HFCs (see Figure 20). The reduction's bases are the calculated CO_2 equivalents. These measures will lead to a significant reduction and thus to an increase of the price of HFC refrigerants – especially for those with a high GWP. In addition, it would be possible to establish national GWP-weighted taxes on refrigerants following Scandinavia and Australia.

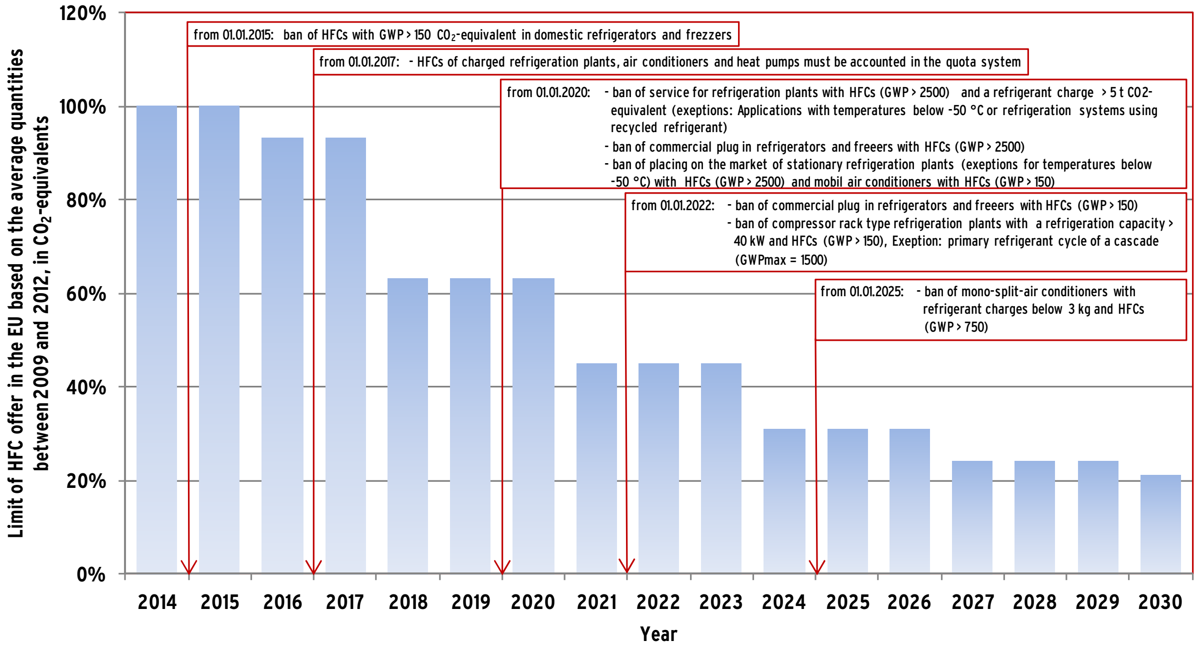


Figure 20: HFC phase down scheme (measured in CO_2 equivalents) according to Regulation (EU) No 517/2014 on fluorinated greenhouse gases

Barriers for the application of heat-driven system are large investment cost, large volume systems, coupling of waste heat, and lack of information of the operating behaviour. Due to grant-supported development of, e.g., compact heat exchangers with structured surfaces, solutions for absorption-compression systems, alternative fluid combination, and multiple-staged processes, positive results are expected regarding increased application and better sustainability in terms of efficiency of heat and electrical energy.

There is an information deficiency with planners, house-builders, and operators leading to a retarded establishment of alternative systems. These alternatives are often unknown and cooling itself is often considered too late in building projects, which is the reason why systems, such as concrete core activation, cannot be implemented any more. Planner- and operator-optimised information distribution has an especial relevance.

The large investment costs of efficient and climate friendly refrigeration systems shall also be understood as business model for the financial and energy sector (energy contracting, energy-saving contracting). Due to expectable rising energy prices, there are several business models cogitable, leading to advantages for both operator and investor.

With the so-called “Impulsprogramm,” a national funding programme is available which supports systems with natural refrigerants, heat recovery, and state of the art components with up to 35 % of the total investment. In 2009 to 2011, 244 plants with an installed cooling power of approx. 144 MW have been supported. Approximately 40 % of those plants were refurbishments. Based on the grants being issued, there have been costs for the avoidance of CO_2 of 35 €/t CO_2 equivalent (Jörß & Klose 2012).

3 Introduction

Refrigeration systems require approx. 14 % of the German electrical energy demand and emit approx. 5 % of all greenhouse gas emissions based on the CO_2 equivalent. Considering this, there is a need for action to increase the climate friendliness of air conditioning and refrigeration systems in Germany in order to comply with the national aims for saving the climate. The present study is a basis for activities and measures to increase the climate friendliness of the refrigeration technology. It covers detailed analyses for the application areas industrial refrigeration (e.g., process cooling) and building air conditioning. The study is organised as follows:

Application areas of this study are distinguished from each other in Chapter 4. Furthermore, the definitions of the relevant terms, parameters, and characteristic numbers are provided there. Those of the parameters and characteristic numbers being important for the evaluation for the climate friendliness are emphasised.

The itemisation of the cooling demand and its coverage of different industrial sectors are provided in Chapter 5 (pp. 40ff). It is supported by supplementary information, such as demand profiles and required refrigerant temperatures. The latter information is important for further analyses and evaluation. The data for this are obtained from different sources, compared, and checked for their plausibility.

Afterwards, an overview of refrigeration and air conditioning systems is provided in Chapter 6 (pp. 57ff), demonstrating the assets and drawbacks of them. The systems under consideration are of compression and absorption type. For the former, the influence of the refrigerant on the process, accessible energy efficiency ratios, and the global warming potential is discussed. Furthermore, a market overview of components and chillers as well as a compilation of realised systems for natural refrigerants is provided. For absorption type chillers, a market overview of components as well as already existing systems is given.

Based on dynamical system simulations for the period of one year, different system alternatives are investigated for building air conditioning and industrial refrigeration in Chapter 7 (pp. 84ff). From those results, influencing factors on the energy demand and the climate friendliness are discussed and conclusions are drawn to provide reasonable measures for climate-friendly system engineering.

Given waste heat sources allow to reduce the primary energy demand and greenhouse gas emission by employing thermally driven refrigeration systems. The increase of utilisation of such systems is investigated for selected sectors by means of suitable assumptions in Chapter 8 (pp. 135ff). The results are employed for deriving the potential of such systems, which will be illustrated.

Chapter 9 (pp. 151ff) finalises the study with an analysis of barriers and measures for the establishment of climate friendly refrigeration systems on the market. The barriers are investigated from legislative, safety-related, and economical points of view.

4 Definition of parameters, characteristic numbers, and classification of the application areas of the study

4.1 Definition and classification of the application areas of the study

The present study is concerned with climate friendly cooling generation in Germany by means of the application areas *industrial refrigeration* and *building air conditioning*. There is an inconsistency in the literature for the term *industrial refrigeration*. Sometimes, air conditioning of industrial buildings (e.g., offices, production halls, etc.) is considered separately, but most often both pure industrial refrigeration (e.g., for cooling purposes in processes) and air conditioning are considered in a combined manner with the term *industrial refrigeration*. Here, air conditioning of industrial buildings is considered building *air conditioning*. Similar uncertainties can be observed in the services sector – some authors consider air conditioning to be commercial cooling, whilst others treat it as building air conditioning.

As far as possible, the application areas are distinguished from each other, but sometimes it was not possible to exclude ambiguity due to missing definitions in the reviewed literature.

4.1.1 Industrial refrigeration

Industrial refrigeration comprises the cooling demand of the industry for the following applications:

- Process cooling
- Storage of basic, semi-finished, and final products
- Climate control of high-performance computers for the industrial sector and web servers
- Climate control of control cabinet

The following applications are *not* considered industrial refrigeration:

- Climate control of clean rooms, production halls (assigned to building air conditioning, industry), if not specified other or separation is impossible due to a lack of data
- Climate control of office buildings in the industry and commercial sectors (assigned to building air conditioning, industry as well as building air conditioning, commercial, trade and services (abbreviated CTS)
- Climate control of sales areas (assigned to commercial refrigeration), if separable from CTS
- Cooling of final product close to the sales areas (assigned to commercial refrigeration)
- Climate control of server rooms in office buildings (assigned to building air conditioning), if not assigned to climate control of control cabinets

4.1.2 Building air conditioning

Building air conditioning covers the climate control of:

- Office buildings,
- Server rooms and data centres, and
- Climate control of clean rooms and production places.

The following items are mentioned within this study but are not considered building air conditioning:

- Climate control of restaurants and hotels (assigned to commercial cooling)
- Climate control of sales areas (assigned to commercial cooling)

4.2 Definition of parameters and characteristic numbers

In this section, definitions of the parameters and characteristic numbers being utilised in the present study are provided. For the case of different definitions in the literature, one is selected upon discussion.

4.2.1 Primary energy demand

The primary energy demand comprises the energy content of the energy carrier (e.g., natural gas, heat, electricity) and the additional energy effort for energy extraction and conversion. The calculation is carried out on the basis of final energy and corresponding primary energy factors, which is in accordance with the planned amendment of the German Energy Saving Ordinance ("Energieeinsparverordnung") 2009[2] (EnEV 2009; Robbi & Sander 2012).

4.2.2 Primary energy factor

The primary energy factor is the ratio of primary energy and final energy, which depends on the energy carrier. The values being presented in Table 3 are taken from DIN V 18599-1:2011-12 and are applicable as soon as EnEV 2014 has entered into force. The evaluation of district heating from industrial waste heat was carried out according to AGFW-Sheet FW 309 Part 1 (AGFW 2010). For illustrating the primary energy factors, two different shares are considered: total and non-renewable share. For fossil energy carriers, both values are the same, but for renewable carriers, they differ depending on the evaluation of the energy carrier. An illustrative example for this is wood, which requires some preparation effort comprising the energies for harvest, desiccation, comminution, and transport. This effort is considered in both shares with 0.2. For calculating the efficiency of the chain of transformation, it is required to evaluate the energy content of the combustible. Due to the complete transformation into heat during combustion (factor 1) the total share having a value of 1.2 (=1+0.2) in this example.

For a national economical evaluation of energy utilisation, employing the primary energy factor as the cost parameter, it is necessary to calculate the investment of non-renewable energy sources. The non-renewable share considers only the energy content of fossil or nuclear fuels (e.g., 0.2 for wood as stated above). Renewable sources do not contribute to it (assigned to be zero).

Aim of the present study is to evaluate refrigeration systems by means of their environmental friendliness (energy demand and total greenhouse gas emissions) and their costs. The influence of the increasing share of renewable energy sources in the power supply has to be considered. In order to comply with this requirement, primary energy factors and their non-renewable share are used in this study.

[2] Prospective entry into force as EnEV2014 according to www.enev-online.de (as of 15.08.2013)

Table 3: Primary energy factors according to DIN V 18599-1:2011-12 (Robbi & Sander 2012)

Final energy carrier		Primary energy factor total	Primary energy factor non-renewable share
Fuels	Extra light fuel oil	1.1	1.1
	Natural gas H, liquefied gas	1.1	1.1
	Stone coal	1.1	1.1
	Brown coal	1.2	1.2
Biogenous fuels	Wood	1.2	0.2
	Biogas	1.5	0.5
	Biopetroleum	1.5	0.5
District heat from CHP	Fossil fuels	0.7	0.7
	Renewable fuels	0.7	0.0
District heat from heating plant	Fossil fuels	1.3	1.3
	Renewable fuels	1.3	0.1
District heat from industrial waste heat	Supply demand, if not known in detail	0.4	0.4
Electrical energy	General mixture	2.8	2.4
	Displacement mixture	2.8	2.8
Environmental energy	Solar energy	1.0	0.0
	Geothermal energy	1.0	0.0
	Ambient heat, ambient cold	1.0	0.0
Waste heat within the building		1.0	0.0

4.2.3 Final energy demand

The final energy demand is the energy bought by the customer by means of a particular energy carrier. It comprises the net energy demand and the system losses (EnEV 2009).

4.2.4 Net energy demand

The net energy demand is the amount of energy being required for a particular cooling purpose. Within this study, it is equal to the cooling demand.

4.2.5 Cooling power demand

The cooling power demand is the total heat flow to be extracted from a process or building.

4.2.6 Cooling energy demand

The cooling energy demand is the accumulated heat to be extracted from a process or building during a given period of time.

4.2.7 Coefficient of performance (COP) and energy efficiency ratio (EER)

The *coefficient of performance* (COP) is defined to be the ratio of net thermal capacity to supply power. The *energy efficiency ratio* (EER) has the same definition and is utilised equivalently for the case of compression type chillers. However, for the case of heat pumps and split air conditioning systems as well as in application of DIN V 18599, the coefficients are employed depending on the kind of usage. The COP is utilised for the case of heating and the EER for the case of cooling. Manufacturers of compressors refer to the compressor unit alone and not the entire compression type chiller. The numerical values of COP and EER are only comparable if they are provided together with the boundary conditions, especially the temperatures of heat source and sink, at which they have been calculated.

4.2.8 Seasonal energy efficiency ratio (SEER)

The seasonal energy efficiency ratio (SEER) is defined to be the ratio of rejected heat versus the received electrical energy during one year. It should not be confused with the performance number, which can be obtained at standardised conditions in the laboratory. The SEER includes seasonal profiles of the temperature and load conditions.

4.2.9 Heat ratio

The thermal efficiency of sorption type chillers can be calculated by utilising the so-called heat ratio, which is defined to be the received heat (benefit) versus the driving heat (effort). Here it is also necessary to provide the temperature conditions of the heat supply (cold and hot-water side) and the heat rejection (cold-water side) with the numerical value of the heat ratio. An exception is the value for the maximum heat ratio, which depends on the refrigerant-absorbent combination and the system type only.

4.2.10 Emission factors for electricity and heat production

The emission factor for the generation of electrical energy in Germany comprises the supply chain from the initialisation, the transformation losses, and the losses in the grid up to the customer (GEMIS 2012). A special problem exists with the calculation of the specific CO_2 emissions of heat and electricity from combined heat and power plants (CHP) due to the simultaneous generation from the same fuel. According to VDI 4660 and VDI 4608-2, six different approaches exist for the calculation – the so-called allocation or evaluation methods (energetic method, labour value method, exergetic method, exergy loss method, residual value method, and substitution method). In addition to this, there are further ones, such as the Finnish method, which is a comparison against a separate generation of both heat and electricity. The application of different allocation methods leads to very different results. A good overview of the methods and their results has been provided by (Dittmann & Robbi 2008).

Two cases are important for the present study:

- CHP as an expansion for industry and agriculture for self-contained electricity generation and utilisation of waste heat for, e.g., absorption type chillers
- District heat from CHP

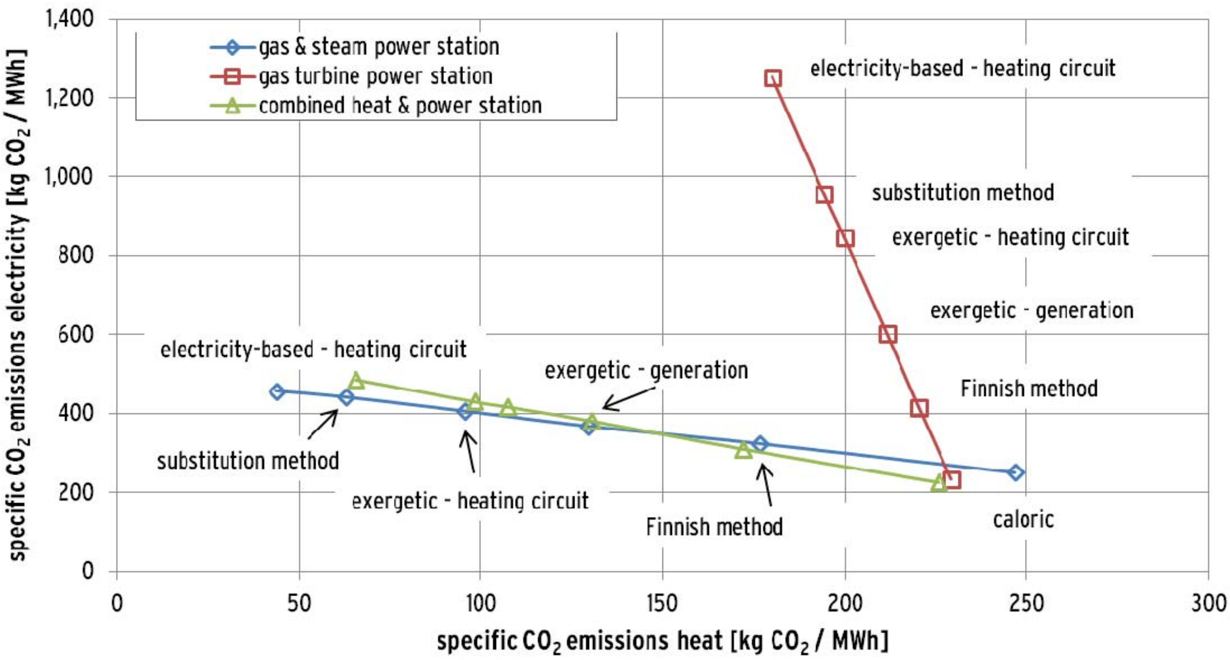


Figure 21: Specific CO_2 emission for heat and electricity from CHP calculated with different allocation methods. (Dittmann & Robbi 2008)

Heat generation in CHP

Electrical energy from CHP is an economic alternative to provide electricity for commercial and industrial companies. Feeding electrical energy into the grid is governed by the Renewable Energy Act ("Erneuerbare Energien Gestz", EEG) for renewable fuels and landfill gas and by the KWKG ("Kraft-Wärme-Kopplungs-Gesetz") for fossil fuel (EEG 2009; KWKG 2009). Hereby, the feed-in is especially interesting when the waste heat is also used, due to a CHP incentive (presently 5.11 ct/kWh for systems < 50 kW).

Different aspects influence the origin of the electricity being displaced by CHP electricity: CHP and renewable energies have a larger priority than other sources when fed-in, leading to a displacement of non-renewable sources. The displacement is subjected to the merit order effect at the electricity stock exchange (Pfeifroth & Beer 2009). By means of a simulation with hourly time steps, the study of (Pfeifroth & Beer 2009) indicates a displacement mixture for 10 TWh CHP electricity with the emission of 821 g CO_2/kWh_{el} as can be observed in Figure 22.

The electrical efficiency of CHP increases with the power class of the system. According to (ASUE 2011), the average efficiency of natural gas CHPs is 38 % based on the fuel. The average thermal efficiency is approx. 49 %.

Referring to the specific CO_2 emissions of natural gas of 198.5 g/kWh (GEMIS 2012) and an average electrical efficiency of 38 %, CHP electricity is responsible of 522 g CO_2/kWh_{el}. Compared to the displaced mixture, savings of approx. 300 g CO_2/kWh_{el} can be observed. Waste heat does not contribute to the CO_2 emissions, but it is imaginable to provide a credit for utilised waste heat. However, for comparison purposes with heat from other waste heat source, a credit is not included here.

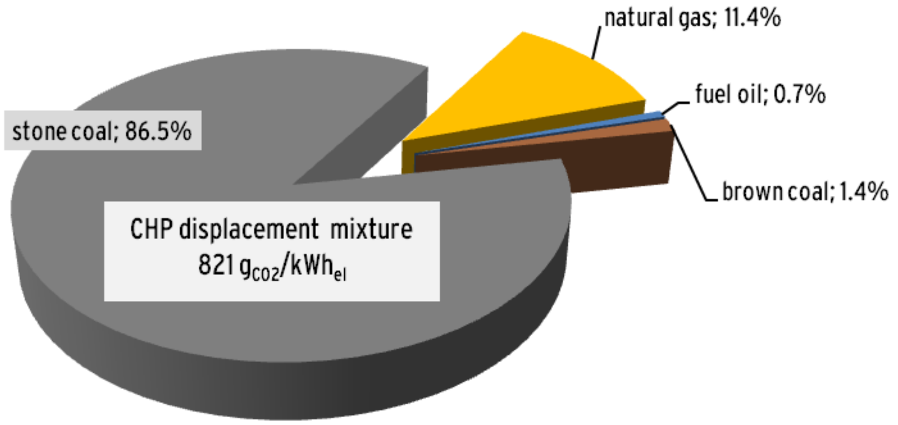


Figure 22: Displacement mixture for CHP electricity feed-in (Pfeifroth & Beer 2009)

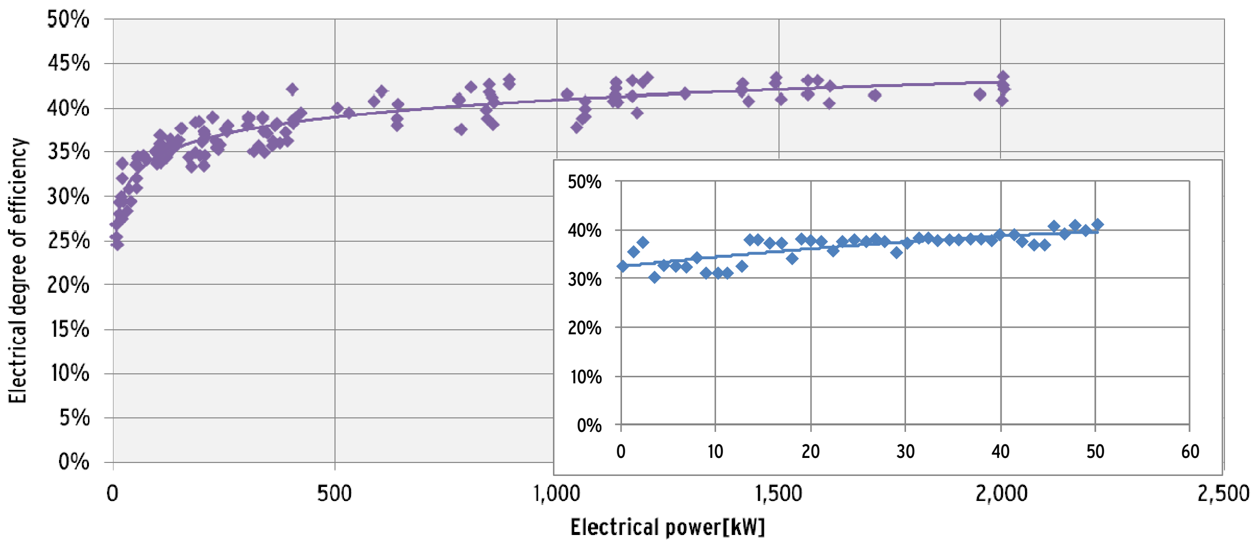


Figure 23: Electrical efficiency of natural gas CHP (ASUE 2011)

District heat

For the extraction of heat from a district heating grid it is necessary to consider the influence of the heat extraction on the process and the efficiency of electricity generation. The evaluation of the working ability of the extracted heat (electricity loss method, Dresden method) is considered a useful method by many experts (Dittmann & Robbi 2008; Dittmann et al. 2009; Zschernig & Sander 2007). With this method, it is possible to calculate the electrical energy which can be generated in a real process when expanding down to ambient temperature instead of heat extraction temperature. For this purpose, the Carnot efficiency is calculated and multiplied with the efficiency factor of the turbine. This method can be applied to any existing power station and does not depend on a fictitious standard cycle (Zschernig & Sander 2007).

However, the calculation of the CO_2 emissions produced by CHP district heat in Germany is not generally possible with this method. The reason is that the fuels at different locations and the extracted heat have to be considered. These problems have been illustrated by the authors by means of some examples (Zschernig & Sander 2007). Also the study of the "Öko-Institut" (Fritsche & Rausch 2008) provides emission values for single kinds of power plants only.

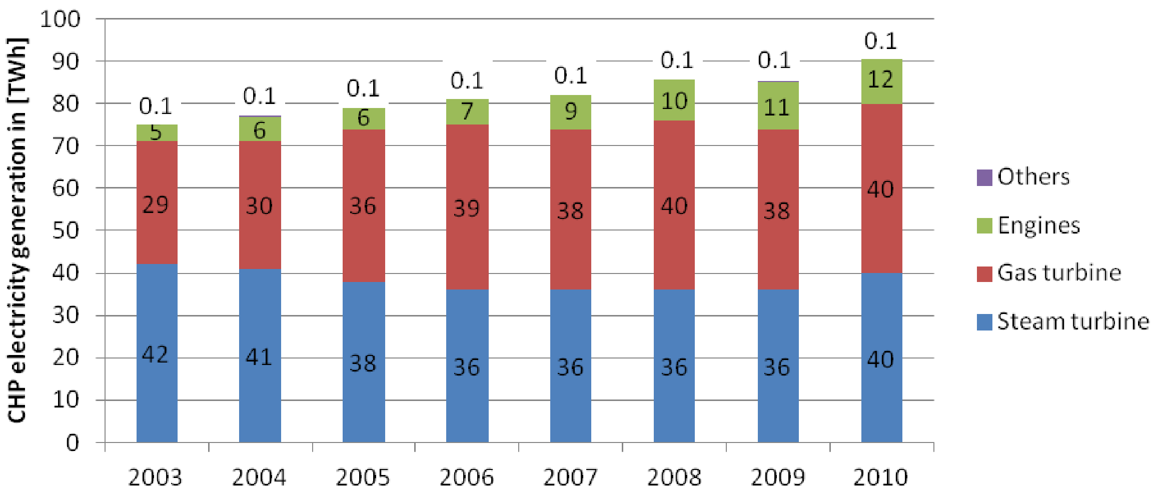


Figure 24: CHP electricity generation in different systems (source: Öko-Institut via (Umweltbundesamt 2012b))

Different allocation methods in conjunction with uncertain data lead to a wide range of possibilities for evaluation. Due to this, it is useful to create a sample space illustrating reasonable emission values for the utilisation of heat for cooling generation. The lower and upper limits for the evaluation within this study are:

- District heat from gas-steam power plant, assigned with electricity loss method → 44 g CO_2/kWh_{th} (Dittmann & Robbi 2008)
- District heat from coal power plant, assigned with efficiency method → 239 g CO_2/kWh_{th} (Fritsche & Rausch 2008)

The specific CO_2 emissions of different energy supplies/carriers utilised within this study are provided in Table 4.

Table 4: Specific CO_2 emissions of different energy supplies for refrigeration systems

Energy supply/energy carrier	specific CO_2 emissions [g/kWh]
Electricity (mixture of Germany in 2010, incl. supply chain, GEMIS 4.7)	583
Ambient heat (solar thermal, geothermal)	0
Waste heat from natural gas CHP considering displacement mixture	0
District heat from gas-steam power plant, assigned with electricity loss method	44
District heat from coal power plant, assigned with efficiency method	239

Following the Federal Environment Agency ("Umweltbundesamt"), CHP systems have generated 86 TWh_{el} in Germany in 2009. Considering the ratio of electricity and heat production of CHP in 2010 (46.2 %), one can observe a heat production of 100 TWh_{th}. A fuel utilisation level of

85 % (Fritzsche 2008, p.40) yields 219 TWh fuel input. The CO_2 emissions of CHP systems are 121.1 million tonnes (Mt) CO_2 in the same period of time (based on the savings compared to reference system) and the average specific CO_2 emissions of the fuel are 553 g $CO_2/kWh_{Fuel, CHP}$.

4.2.11 Global warming potential of refrigerants (GWP)

The global warming potential (GWP) is a characteristic factor estimating the greenhouse effect of a gas being released to the atmosphere compared to the effect of CO_2. It comprises the heat radiation absorption of a given gas and the resting time of molecules in the atmosphere. The GWP of CO_2 equals 1. The numbers being shown in Table 5 have been taken from Tables 2.14 and 2.15 of (Forster & Ramaswamy 2007). They are effective from 2013 on for greenhouse gas emission reports according to the Kyoto protocol. Figure 25 displays the GWP and the safety classification according to DIN EN 378, whereby the latter comprises the classification with respect to flammability and toxicity. An overview of the criteria in DIN EN 378 is provided in Table 6.

Table 5: GWP values for different refrigerants

Refrigerant	Type	Description/Composition	GWP value
R32	HFC	Diflourmethane	675
R134a	HFC	1,1,1,2-Tetrafluorethane	1,430
R404a	HFC Blend	Mixture of R125, R134a, and R143a	3,922
R407C	HFC Blend	Mixture of R32, R125, and R134a	1,774
R410A	HFC Blend	Mixture of R32 and R125	2,088
R1234yf	HFO	2,3,3,3,Tetrafluorprop-1-en	4
R1234ze	HFO	Trans-1,3,3,3-tetrafluoroprop-1-en	6
R290	Hydrocarbon, natural	Propane	3.3
R600a	Hydrocarbon, natural	Isobutene	4
R717	Natural	Ammonia	0
R718	Natural	Water	0
R723	Natural	Mixture of dimetylether and ammonia	1
R744	Natural	Carbon dioxide	1
R1270	Hydrocarbon, natural	Propene (propylene)	1.8

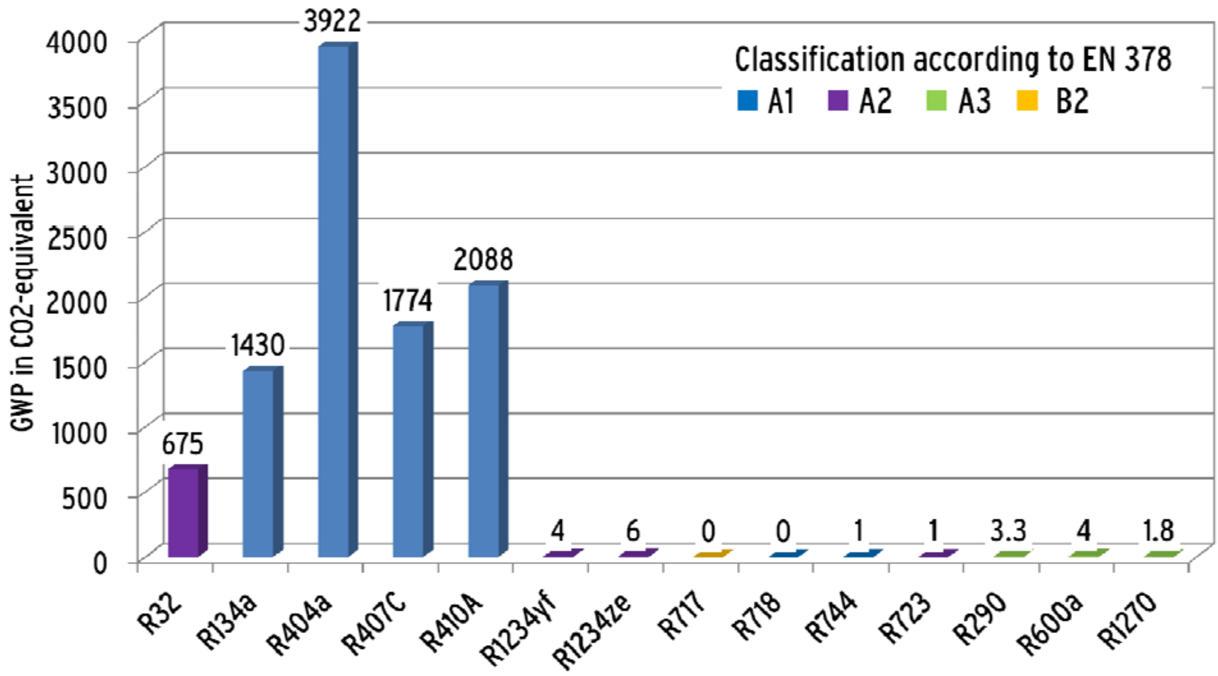


Figure 25: Comparison of the different refrigerants by means of GWP values and their safety classification according to DIN EN 378

Table 6: Safety classes for flammability and toxicity (DIN EN 378-1)

Increasing flammability ↑	Increased flammability (lower ignition limit ≤ 3.5 Vol.-%; combustion heat ≥ 19 MJ/kg)	A3	B3
	Low flammability (lower ignition limit ≥ 3.5 Vol.-%; combustion heat < 19 MJ/kg)	A2	B2
	No flammability	A1	B1
		Low toxicity (TLV ≥ 400 ppm)	Increased toxicity (TLV < 400 ppm)
		Increasing toxicity →	

4.2.12 Total equivalent warming impact (TEWI)

The total equivalent warming impact (TEWI) is a number for estimating the greenhouse gas emissions of a system comprising direct refrigerant emissions (e.g., through leakages, improper disposal) and indirect emissions (e.g., carbon dioxide emissions) originating in the conversion of energy being required for working the refrigeration system. All emissions are accumulated during life time of the system and measured in terms of CO_2 equivalents (DIN-EN-378-1 2011). A more detailed discussion can be found in Section 7.2.2 on p. 85.

5 Energy consumption of non-residential building air conditioning and process cooling demand of industry and manufacturing trade

5.1 Preliminary considerations

Compared to existing data for electricity generation, electricity, and fuel consumption for electricity generation, building heating, and traffic, there is only limited data available for the cooling demand and its final energy demand in Germany. Hence, the influence of refrigeration and air conditioning on the energy-based CO_2 emissions is difficult to evaluate.

Statistical coverage of the energy consumption of various cooling tasks would be useful for the assessment of refrigeration systems in different applications. This information could be also helpful for the user deciding for a refrigeration system. As soon as the energy demand, emissions, and costs for providing cold, and the opportunities for manipulating costs and emissions via system design are known, increasing appreciation for this problem will be observed.

Due to a lack of data of energy consumption of the refrigeration technology, a comparison of literature data is carried out. The sources are checked with theoretical calculations and tests for plausibility and cogency.

5.2 Methodology

This chapter is the base for a quantitative evaluation of the present state of the art. From this information, it is possible to assess measures and their effect on the consumption of electrical and primary energy as well as the CO_2 emissions originating from refrigeration systems. Therefore, a database system has been erected for different scenarios and industrial sectors. In order to judge the substitutability of electrically driven by thermally driven refrigeration systems, the energy demand for cooling generation, the temperature of the cold, the temporal characteristics of the cooling demand, and the availability of heat sources have to be determined. The resulting database is larger than the one of the study about the energy demand for refrigeration technology in Germany (Guntram Preuß 2011). However, for some information it was necessary to make some feasible assumptions, which are documented in following sections where the energy demand for refrigeration technologies (air conditioning and process cooling) in various industrial sectors are derived. Determination of possible climate protection effects includes the use of natural refrigerants in compression and absorption type refrigeration systems, as well as for thermal storages.

Beside the cooling demand itself, the characteristics of the demand (average cooling temperature, seasonal and daily oscillations) are covered within the database. The same is carried out for waste heat sources, in order to be able to study the effect of a change from compression to absorption type chillers. In addition to the waste heat sources being already employed, prospective ones can be found in the heat demand of the other sub-processes. This leads to an increased potential of CHP systems.

Figure 26 illustrates the structure of industrial sectors which has been adopted from a VDMA study (Guntram Preuß 2011) and further employed in the database. In order to estimate the cooling demand for food production, the scheme has been reduced due to lack of data in the VDMA study. The classification has been carried out by the size of the company divided into normal and low-temperature cooling. For the estimation of the waste heat potential, a structure is necessary which is provided in Figure 27. However, a transformation of the data from the VDMA study into the present one is only possible with some assumptions.

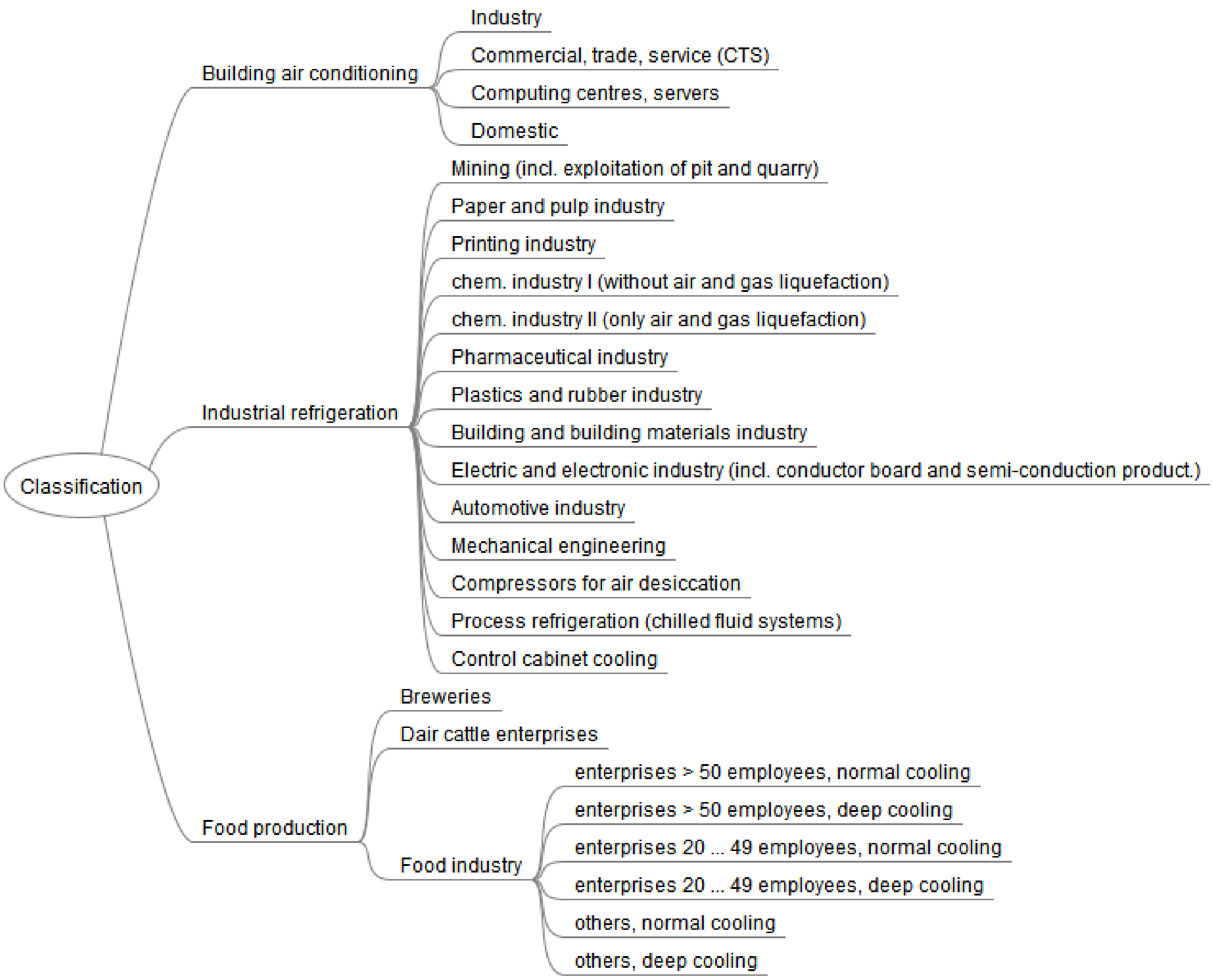


Figure 26: Classification of the cooling and energy demand – food industry according to VDMA study (Guntram Preuß 2011)

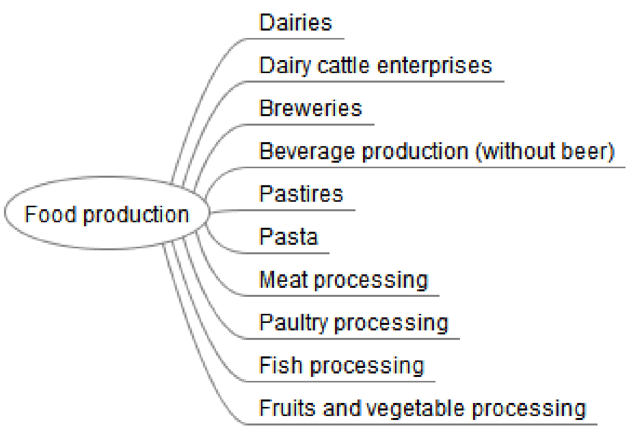


Figure 27: Sub-classification of the food industry according to industrial sectors

5.3 Data sources

Data for the cooling demand of industrial and building air conditioning are only limited available. Compared to the electrical energy, natural gas, and fuel oil consumption, the cooling energy is usually not measured. The same is true for the energy being necessary for driving the refrigeration machine, leading to the situation that quantitative statements have to be made on the basis of assumptions and simulations. An additional drawback of the lack of data is that users are not aware of the potential to reduce the energy for cooling generation.

The most important references of the energy demand assessment is presented in the following. Beside those references, industry-sector-specific literature is provided.

5.3.1 Preuß: Energy demand for refrigeration technology in Germany - 2011

An important reference is the VDMA study already mentioned above on the energy demand of refrigeration technology in Germany, which was available only in the draft versions of February and March 2011 (Guntram Preuß 2011). This study extends an earlier study on the utilisation of refrigeration technology in Germany written by the same author (Gruntram Preuß 2010) and follows up a DKV study on the state of the art as of 2002 (DKV 2002).

Basis for the VDMA study are calculations and analyses of a variety of statistics and information of refrigeration technology companies and users. Due to a lack of data, the author has carried out estimations and plausibility assumptions and considers the outcome of the study to be a basis for further discussion. Within the study, the author presented the final energy demand in terms of electrical energy, whereas the cooling demand is provided only rarely. Furthermore, the inclusion of sorption refrigeration systems appears to be incomplete, compared to other studies. Three out of the eleven sectors of the refrigeration technology are also employed here: food production, air conditioning, and industrial refrigeration.

5.3.2 DKV status report 22 — energy demand for cooling generation - 2002

The DKV study (DKV 2002) is a predecessor of the VDMA study which has been compiled by different institutions with various methods and approaches. Opposing to the VDMA study, it provides detailed information about required temperature levels, energy efficiency ratios, and the cooling demand itself. In absences of up-to-date data these information are also employed for the present study.

5.3.3 Data basis for the evaluation of measures for increasing the energy efficiency - 2008

Prognos, TU Munich, and Fraunhofer ISI (Seefeldt et al. 2010) provided detailed information about the electrical energy demand of the refrigeration technology, itemised by industrial sector and cooling application. Furthermore, the authors provide electrical energy and total heat demand information, and information about energy sources for heat generation. This information is employed in the present study to supplement the data of the VDMA study and to itemise the cooling demand of air conditioning and process refrigeration.

5.3.4 "AG Energiebilanzen e.V." - 2011

The statistics of the AG Energiebilanzen e.V. (AG-Energiebilanzen-e.V. 2011) provides some detailed insight into the utilisation of different energy carriers in their respective industrial sectors. Furthermore, one can find statements related to electricity and heat demand and their corresponding sources. This information can be employed to estimate the potential for thermally driven refrigeration systems. This reference is additionally to (Seefeldt et al. 2010) very useful for providing a database of electricity and total heat demand as well as of the energy carriers for the heat generation.

5.3.5 ASUE — Absorption chillers — basics and references — 1995

The above-mentioned studies provide only little information about thermally driven refrigeration systems. However, the working group "Arbeitsgemeinschaft für sparsamen und umweltfreundlichen Energieverbrauch e.V." provided a study on absorption type chillers (ASUE 1995). As it has been published in 1995, it cannot represent today's situation but demonstrates applications of absorption type chillers very well. The share of these chillers on the total cooling demand can be checked for plausibility with this study.

Due to the long lifetime of absorption type chillers (due to few components with movable parts up to 50 years), it can be expected that there are still a lot of chillers in service. Furthermore, (Simmert 2011) observed that absorption type chillers with a total cooling of 45 MW have been installed in 2010, which supersedes the annual average of the years 1970 until 1996 by factor four. Compared to the annual average between 1990 and 1994, there is still an installation rate being twice as high. From this information, it can be concluded that the data within the study represents the lower limit.

5.4 Industrial refrigeration

5.4.1 Mining industry, exploitation of pit and quarry (only stone coal)

The DKV status report 22 (DKV 2002) provides an analysis of the mining industry in Germany and states that the necessity for refrigeration technology is found for stone coal mining aeration and for salt-mines. The exploitation of salt is usually carried out with mobile machines having a driver's cabin and therefore is considered vehicle air conditioning (Guntram Preuß 2011). Preuß has corrected the data of the DKV status report by the reduction of stone coal exploitation to a third of the value of 1999, yielding a final energy demand of 160 GWh/a (Guntram Preuß 2011). Following (DKV 2002) and seasonal energy efficiency ratios provided within, the cooling energy demand is estimated to be 560 GWh/a at a cooling medium temperature of 3 °C.

There are two approaches available for employing thermally driven systems. Firstly, firedamp (mine gas) originating from coal mines, which can be burnt in CHP. The resulting heat (available year-round) can be utilised for cooling generation. Such systems are built in national mines and abroad (e.g., North Rhine-Westphalia, Poland, Russia, China, etc.) (Evonik_Industries 2010; IGV 2012). At the German locations, approx. 250 Mm^3 (million cubic metres, at standard reference conditions) of firedamp have been accumulated in 2010, whereof more than 85 % originate in passive mines (IGV 2012). The firedamp is burnt in 125 CHPs in Germany with a total installation power of 158 MW_{el}. Considering a heat to electricity ratio of 1.35 (see Figure 110 in section 8.2, page 142), this value corresponds to a heat flow of 215 MW_{th}. Considering furthermore a degree of capacity utilisation of 98 % during one year, 1,790 GWh_{th} can be provided, whereof 260 GWh_{th} originate from active mines. Heat from the active mines is currently utilised by the public services who couple the heat into the district heat grid (IGV 2012). It is very likely that there exists a heat excess during the summer. There are some places where beside firedamp also coal is burnt or converted to coke or raw gas. For instance, Ibbenbüren has a coal power station with an electrical power of 709 MW_{el} but the heat of the firedamp is not used yet. However, due to progressing mine-closings, this option will become less important.

5.4.2 Paper and pulp industry

The paper and pulp industry has only limited necessity of process cooling. A much larger demand is given with the air control of the production halls, which is assigned to building air conditioning (Guntram Preuß 2011). The accumulated electrical energy for process cooling accounts for 90 GWh/a (Seefeldt et al. 2010). With a seasonal energy efficiency ratio of 2.8 (LfU-Bayern 2003) one can find a cooling demand of 252 GWh/a.

The references list of the ASUE exhibits absorption type chillers with a total installed cooling power of 5.5 MW (ASUE 1995). Even with full-load during the entire year, cooling contribution of this fraction is negligible.

5.4.3 Printing industry

Refrigeration technologies are required in the printing industry for cooling printing machines and for cooling print products after their infrared or ultra-violet desiccation (Guntram Preuß 2011; Quint-sdi-GmbH 2010). Furthermore, it is often necessary to air control the production halls, especially when the waste heat of the machines is not directly lead to the ambient. The chilled water temperatures for print machines are usually between 20 and 28 °C which is quite high (Ltd / Co. KG 2010; Tebert & Schmid-Bauer 2006). Therefore, a refrigeration system is required only in a few summer months. However, the chilled water is usually much colder (e.g., 6 °C supply temperature (Henrichs 2009)).

The required electrical energy for refrigeration systems in the printing industry is estimated to be 55 GWh/a (Guntram Preuß 2011). For this purpose it has been assumed that 5 % of the specific electrical energy consumption (150 kWh/t_{paper} (Gloor 2010)) are required for refrigeration technology. It has been assumed furthermore that a third of the annual paper production of 22 Mt is printed (Guntram Preuß 2011).

Due to the large temperatures of the waste heat, it is difficult to estimate the cooling demand. For instance, paper rolls are cooled down after the desiccation process from 130 °C to 35 °C. (Cornehl 1998) provides information for Mohndruck Media GmbH in Gütersloh, which has an own gas-steam power plant with three gas turbines and an absorption type chiller downstream. The installed cooling and electrical powers are both 19 MW, and the annual cooling energy production is quoted to be 35 GWh/a. The produced cold is employed for process cooling and for air conditioning of storage and office buildings. The annual printed paper production of Mohndruck amounts to 380 kt (Henrichs 2009), which is approx. 5 % of the estimated annual production of 7.3 Mt in Germany (Guntram Preuß 2011). Extrapolating the cooling demand of Mohndruck to the annual printed paper production in Germany yields 700 GWh/a cooling energy demand. Considering the electrical energy demand for cooling of 55 GWh/a, and assuming compression type chillers, the average seasonal energy efficiency ratio is 12.7. This very large value is quite reasonable due to the large supply temperatures (Ltd / Co. KG 2010; Tebert & Schmid-Bauer 2006). The air conditioning share of the total cooling demand can be estimated by considering 2,000 employees of Mohndruck Media GmbH in Gütersloh (Hommel 2006) and the typical annual cooling demand per person of 640 kWh/(a·person). Assuming that two third of the employees work in offices, the air conditioning demand is 0.85 GWh/a. The share of 2.4 % of total cooling demand is almost negligible.

The application of combined heat, power, and cooling plants (with CHP or gas turbine systems) in the printing industry has been reported by various authors (Fischedick et al. 2002; Henrichs 2009). However, only Mohndruck Media GmbH provided information about a system in operation (Henrichs 2009).

5.4.4 Chemical industry (without air and gas liquefaction)

Detailed data for the application of refrigeration technology in the chemical industry are not available. The study of the VDMA (Guntram Preuß 2011) is based on the report of (Seefeldt et al. 2010) and takes their values for the electrical energy demand for process cooling in basic chemical industry and and other chemical industry. They account the electrical energy demand to 1950 GWh/a in 2007 (Seefeldt et al. 2010), whilst (DKV 2002) mention 404 GWh/a in 1999. However, the rise cannot be explained with expanded production. The data of (Seefeldt et al. 2010) is utilised within the present study.

Cooling generation is usually carried out with large systems of installed cooling capacities greater than 1 MW (DKV 2002). The weighted average temperature of the cooling medium is given to be -4.2 °C and the cooling demand is 1085 GWh/a in 1999 (DKV 2002). Assuming solely compression type refrigeration, one can calculate a COP of 2.7 in 1999. Applying this to the electrical energy consumption in 2007 (Seefeldt et al. 2010) yields a cooling demand of 5270 GWh/a.

The VDMA study does not provide any information about absorption type chillers for this industrial sector. However, (ASUE 1995) provides information of the installed cooling capacity for the chemical and pharmaceutical industries, which is approx. 100 MW and therefore half of the installed absorption type cooling capacity in Germany in 1995. It has to be stressed that systems are considered only which have been taken into service after 1970. Referred to the cooling capacity, 97 % of those plants are indirectly heated systems (waste heat, district heat, steam). The annually installed power was 4 MW/a (8.3 MW/a for all industrial sectors). Comparing these numbers with recent installation rates (e.g., 45 MW in 2010 (Simmert 2011)), it is evident that the inventory has to be corrected. An installed absorber power of 200 MW is assumed for the calculation of the energy demand.

Estimation of the full-load hours is difficult, since users often have installed both compression and absorption type chillers which are utilised depending on the costs for electricity and heat.

4,000 full-load hours have been assumed for the estimation, yielding a cooling demand of 800 GWh/a and a heat demand of 1,140 GWh/a assuming a heat ratio of 0.7.

5.4.5 Chemical industry: air and gas liquefaction

Due to the special refrigeration procedure and their large energy demand, air and gas liquefaction has a special part in chemical industry. The gas itself is the refrigerant in the most common processes (Linde and Claude-Heylandt processes) (Jungnickel et al. 1985). Hence, the cooling generation cannot be carried out by other machines.

The annual oxygen production in Germany has been in the range between 5,000 and 7,700 Mm^3 in the previous decade, whereby values at the lower end have been observed only in 1999 and 2009 (VCI 2011). For the further evaluation, an average value of 7,000 Mm^3 is applied (consider 2008: 7,210 Mm , 2010: 6934 Mm). The production of 7,697 Mm^3 oxygen consumed 5,400 GWh_{el} in 2007 (Seefeldt et al. 2010), which corresponds to a specific electricity consumption of 0.7 kWh/m^3.

The itemised shares of the generated cooling energy for different gases has been provided by (DKV 2002). Oxygen generation has a share of 65 % of the total gas amount and has a share of 68 % of the produced cooling energy. There has been an oxygen production of 5634 Mm^3 in 1999 (VCI 2011). The invested operating power for gas liquefaction has been 5,006 GWh/a in 1999 and the generated cooling energy 900 GWh/a (DKV 2002). Referring to the above-mentioned shares, oxygen liquefaction required 3,253 GWh/a. An average oxygen production of 7,000 Mm^3/a and a specific electricity demand of 0.7 kWh/m^3 yield an electrical energy demand of 4,900 GWh/a. Considering the share of oxygen production according to (DKV 2002), the total electrical energy demand has been approx. 7,540 GWh/a.

The VDMA study (Guntram Preuß 2011) found for a sub-average oxygen production in 2009 an electrical energy demand of 5,200 GWh/a. Extrapolating for years with an average production rate, an electrical energy demand of 7,540 GWh/a appears to be realistic. The generated cooling energy is 1,120 GWh/a employing an analogous approach of calculation.

A substitute for the cooling production for air separation is impossible with heat-driven processes. However, examining the Claude process in the T-s diagram (see Figure 28), an isothermal compression is realised between the points of state 10 and 1.

The isothermal compression is carried out with a multi-staged compression unit with intermediate cooling, having up to thirty compressor stages. However, the intermediate cooling is not carried out after every compressor stage in order to reduce the pressure drop and hence the efficiency loss. This leads to gas exit temperatures of 500 °C in industrial plants, whereby the waste heat is converted into mechanical work through a steam process.

A further sub-cooling by employing a heat-driven refrigeration system has to be checked from technical and economical point of view. Some possible approaches for a realisation have been published in patents (Riesch 1999).

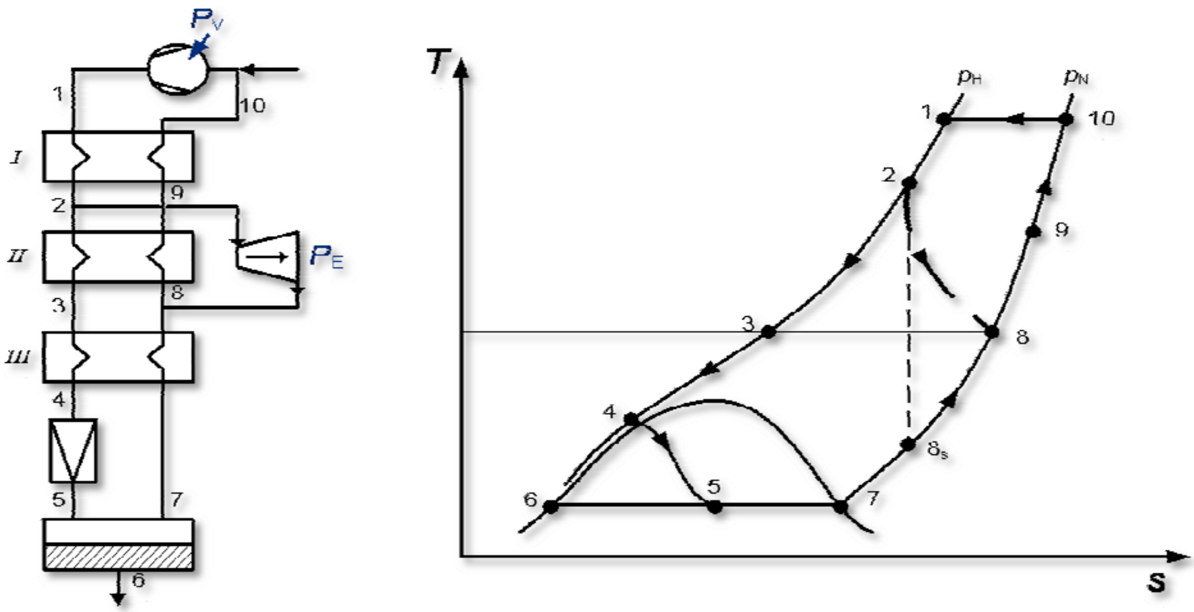


Figure 28: Schematic diagram (left-hand side) and process in T-s-diagram (right-hand side) of air decomposition by means of an expansion machine (Jungnickel et al. 1985)

5.4.6 Pharmaceutical industry

No statistical data exists for the cooling and its resulting final energy demand for the pharmaceutical industry (DKV 2002). (Guntram Preuß 2011) estimated a final energy demand of 328 GWh/a for 304 enterprises with more than 20 employees. The utilisation of absorption type chillers is mentioned within the study, but not in a quantitative manner. Hence, it is assumed that the absorption type chillers require a share of 35 % of the final energy demand, as it has been observed in the chemical industry. For the calculation of the cooling demand, a seasonal energy efficiency ratio of 4.0 for compression type chillers (large systems, cooling demand year-round), and a heat ratio of 0.7 for absorption type chillers have been assumed.

5.4.7 Plastics and rubber industries

There is a strong necessity for cooling in the plastics and rubber industries to cool components after the extrusion or the injection die casting processes. Depending on the product and machine, the temperatures are between 200 and 270 °C (LfU-Bayern 2002). The cooling speed is mainly influenced by the supply temperature of the cooling medium. In order to increase the degree of capacity utilisation, even processes with a cooling from 200 to 80 °C are cooled with a cooling medium temperature of 6 °C. Furthermore, there is a necessity for very dry air in plastics production.

The overall electricity demand of the plastics and rubber industries accumulates to 14,150 GWh/a (Seefeldt et al. 2010). VDMA and ILK Dresden estimated the share of refrigeration technology of the total electricity consumption to be 5 % (yielding 700 GWh/a (Guntram Preuß 2011)), whereas (LfU-Bayern 2002) obtained 10.3 % from measurements. The company where the measurements have been carried out has installed compression type systems only. The COP of the systems (including cooling towers), measured between February and April, was 5.8 (at 2 °C average ambient temperature) and 2.8 (at 11 °C average ambient temperature). Considering even warmer months, an average COP of 2 appears to be reasonable. Based on the results of (LfU-Bayern 2002), 10 % is considered to be the share of cooling electrical energy consumption. Therefore, the cooling electrical energy consumption is 1,410 GWh/a and the cooling demand is 2,820 GWh/a assuming an average COP of 2.0 and a supply temperature of 6 °C.

The above-mentioned temperatures above 200 °C in extrusion and injection die casting systems offer some potential for absorption type refrigeration systems.

5.4.8 Building and building material industry

The main applications of refrigeration technology in the building industry are soil freezing and concrete hardening (DKV 2002; Guntram Preuß 2011). For the latter, only liquid nitrogen is employed, which is accounted in the cooling demand of the chemical industry. Following (DKV 2002), soil freezing is carried out with a brine requiring 0.9 GWh/a cooling energy demand and a final energy demand of 0.6 GWh/a. (Guntram Preuß 2011) postulates an accumulated maximum final energy load in this particular industrial sector of 10 GWh/a. Since (DKV 2002) provided a comprehensible calculation, their data is utilised in the following.

5.4.9 Electric and electronics industry

In the electric and electronics industries, refrigeration technologies are employed for the cooling of process in the semiconductor and photovoltaic production, and the air conditioning of clean rooms. The VDMA study estimates the final energy demand for cooling (without clean room air conditioning) to be 80 GWh/a (Guntram Preuß 2011). (Brinckmann 2008) provides insight into the application of absorption type chillers in a large semiconductor factory in Europe. Four single and seven double-staged absorption type systems are in charge for providing the cooling base load at 5 °C and 11 °C supply temperatures. Referred to a supply temperature of 11 °C, a cooling power of 53.4 MW is installed. In addition, there are turbo chillers with a total power of 50 MW installed for covering peak demands. According to the base load coverage, it is very likely that there are a large number of full-load hours. Assuming 4,000 full-load hours of the absorption type chillers yields a cooling demand of approx. 210 GWh/a. For the turbo chillers it is assumed that they have 1,500 full-load hours and hence provide a cooling load of 75 GWh/a. From the data of (Brinckmann 2008) it is, however, not possible to determine the shares of air conditioning of clean rooms and process cooling. Hence, it is assumed that both have the same share. The heat ratio is appointed to be 0.9 (absorption type refrigeration) and the COP to be 4.0 (compression type refrigeration), yielding a final energy demand of 235 GWh/a of heat and 18.8 GWh/a of electrical energy. It is worth to notice that the electrical energy demand for process cooling is approx. a tenth of the estimate of the VDMA study (Guntram Preuß 2011).

The total electrical energy demand of the industrial sector is taken from (Guntram Preuß 2011) and the cooling demand calculated with a COP of 4.0. The share of absorption type refrigeration is estimated to be four times larger than the value for the Globalfoundries factory in Dresden.

The estimation of the total energy demand for the cooling generation, heat generation, and other processes is difficult, since they are not determined separately (AG-Energiebilanzen-e.V. 2011; Seefeldt et al. 2010). An assignment to the sector “other manufacturing industry” is possible. The electrical energy demand is 27.1 TWh/a here and the cooling generation for air conditioning is observed to be 770 GWh/a (Seefeldt et al. 2010).

5.4.10 Automotive industry

The VDMA study provides numerous applications of refrigeration technology in the automotive industry. Among those are mentioned: air control for the painting process, desiccation systems, test facilities, and chilled water for the production systems (Guntram Preuß 2011). By means of exemplary calculations, the study mentions a share of 2 % of the refrigeration technology of the total electric energy demand of this industrial sector. The share corresponds to an electrical energy demand of 370 GWh/a. (Seefeldt et al. 2010) observed an electrical energy demand of 520 GWh/a, however, including also building air conditioning.
(ASUE 1995) report an installed absorption cooling power of 17.8 MW between 1970 until 1995. Considering the rates of increase according to (Simmert 2011), it is reasonable to assume a presently installed absorption power of 35 MW in the automotive industry. The degree of capacity utilisation is assessed to be 2000 full-load hours.

From the refrigeration systems of the VW factory in Dresden, it is known that there are two compression type chillers with water of 1,000 kW each, one district-heat-driven H_2O-LiBr absorption type chiller of 1,400 kW, one system for free cooling with 1,212 kW, and a compression type chiller of 550 kW cooling power. The available data (i.e., refrigerant and refrigerant-absorbent combination) allows estimating the supply temperature to be approx. 6 °C. Furthermore, the COP is assumed to be 4.0 for compression type chillers and the heat ratio to be 0.7 for the heat-driven chillers.

5.4.11 Mechanical engineering

Regarding the utilisation of refrigeration technology, automotive industry and general mechanical engineering are comparable. With a similar scheme, the share of refrigeration technology of the electricity consumption is estimated to be 1 % yielding an electricity demand of 117 GWh/a (Guntram Preuß 2011). However, (Seefeldt et al. 2010) determined 4.5 % of the electricity consumption. For the present study, an average value of 250 GWh/a is assumed and the COP is considered to be 4.0, similarly to the automotive industry. There is no information available about the utilisation of absorption type chillers in mechanical engineering (ASUE 1995; DKV 2002; Guntram Preuß 2011; Seefeldt et al. 2010).

5.4.12 Compressors and compressed air systems

Due to separate data for compressed air systems (Radgen & Blaustein 2001) it is possible to investigate them in detail. Refrigeration technology is utilised for dehumidifying the air requiring 94 GWh/a of electrical energy (Guntram Preuß 2011). Compressed air dehumidifiers are employed down to 3 °C leading to a COP of 4.0 at an almost constant cooling demand during the year. The waste heat occurring during compression can be employed for driving an absorption type system or for the regeneration of saturated sorbents, whereby the latter option should be preferred.

5.4.13 Process cooling (water chillers) for industrial applications

The VDMA study provides detailed information about the application of water chillers in the industry. The majority of refrigeration systems is employed in metal-working industries for cooling of machine tools. However, a detailed itemisation with respect to industrial sectors is not available. Sectors that are considered elsewhere are not included here (see Figure 29). The remaining sectors are metal and woodworking. A final energy demand of 504 GWh/a is mentioned in the study (Guntram Preuß 2011). Assuming compression type systems only and a

COP of 3.5 (due to usually smaller systems than in the chemical industry), the cooling demand is 1,764 GWh/a.

Referred to the ASUE statistic in the years between 1970 and 1995, approx. 15 MW ad- and absorption cooling power has been installed in the metal and woodworking industries (ASUE 1995). Correcting these numbers with (Simmert 2011), one can estimate 30 MW of presently installed heat-driven cooling power. Assuming 2,000 full-load hours yields a cooling energy demand 60 GWh/a, and with a heat ratio of 0.7, a thermal energy demand of 85 GWh/a.

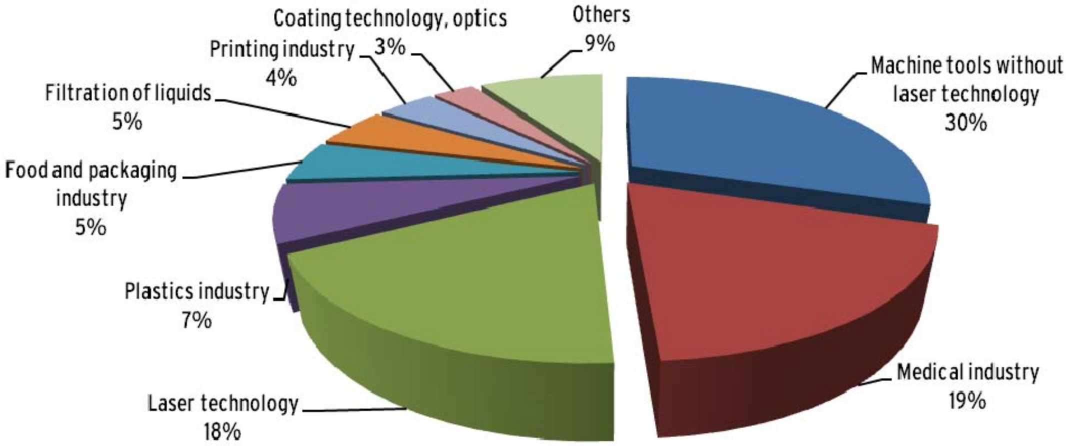


Figure 29: Process cooling with cooling medium systems – sales of the manufacturer according to client industrial sectors (Guntram Preuß 2011)

5.4.14 Cooling of control cabinets

The VDMA study (Guntram Preuß 2011) mentions 313,000 units for the cooling of control cabinets with active systems, power electronics, process measuring and control technology, smaller server cabinets, and server rooms. The delimitation to data centres, which are assigned to air conditioning, is difficult, since there are no criteria determined, such as the size or the power demand. The study observed an electrical energy demand of 313 GWh/a.

For most of the control cabinets, there is a cooling demand rather than a refrigeration demand. Most of the major components still work fine at temperatures of 60 °C. However, due to the large heat fluxes, refrigeration systems are in operation. There is no information available, if the 313,000 units with cooling are operated with systems including a refrigeration unit or not. However, there have been some exemplary analyses. The heat rejection has been sometimes carried out with a refrigeration unit and sometimes without. The COPs of both types have been between below 1.0 and 2.1 (new systems) (eCool 2011; Nelles et al. 2011; Scholl & Koch 2011). For the further evaluation of the cooling demand, the COP is assumed 1.2.

There exists a huge potential reducing the energy demand, e.g., by employing systems working similar to free cooling (just employing a pump for the fluid) (Marcinichen et al. 2010).

5.5 Food production

The VDMA study itemises this industrial sector into three major parts:

- Food industry
- Dairy farm
- Brewery

5.5.1 Food industry

All the sectors are summaries here, which are not breweries and dairy farms. Hence, food industry comprises:

- Butcheries
- Meat processing
- Fish processing
- Fruit and vegetables processing
- Companies for milk processing and ice cream production
- Companies for production of pastries (without long-life pastries)

The German Federal Ministry for Food, Agriculture, and Consumer Protection provides very detailed information about the final energy demand in terms of electricity, natural gas, heating oil, and coal for their branches in the statistical almanac (Ernährung 2009). However, there is no detailed information available about the cooling demand and its resulting final energy demand. The VDMA study subdivides these branches according to their number of employees and assigns some refrigeration system with an average power. Furthermore, there is a distinction between normal cooling and frrezing with a ratio of 60:40 based on the final energy demand (Guntram Preuß 2011). (DKV 2002) mentioned the specific final energy demand of butcheries (15 kWhel/pork, other animals correlated with factors), bakeries (2.06 % of total electrical energy demand), and for juice production (12 Wh/l cooling energy).

The numbers for the cooling demand and hence the final energy demand differ significantly between the VDMA study (Guntram Preuß 2011) and (DKV 2002). Whilst DKV estimates the cooling demand to be 52,487 GWh/a, VDMA mentions 12,322 GWh/a. Similarly, the final energy demands are 18,562 GWh/a and 6,027 GWh/a, respectively. The COPs differ also, and are mentioned to be 2.8 and 2.0, respectively (DKV 2002; Guntram Preuß 2011). With an electricity demand of the sector (without breweries and dairy cattle) of approx. 17,000 GWh/a (Ernährung 2009; Seefeldt et al. 2010), the results of (DKV 2002) appear to be too large. Therefore, those of the VDMA study are employed here.

The study of the ASUA lists more than 20 examples of installed absorption type systems with more 23 MW installed cooling power for the food industry (ASUE 1995). A detailed research revealed that the thermal energy has been provided by waste heat from CHP. However, there has not been any evidence for the utilisation of process waste heat as a heat source. Earlier investigations of ILK Dresden about the usage of absorption type systems in the bakery industry revealed that the temporal characteristics of heat supply and cooling demand do not correlate usually. Therefore, heat-driven cooling systems require an additional cold or heat storage.

Accumulating the demand of the energy carrier natural gas, heating oil, coal, and district heating, the industrial sector requires approx. 40,000 GWh/a (AG-Energiebilanzen-e.V. 2011; Ernährung 2009). From these numbers can be concluded that there are various opportunities for the application of combined heat, power, and cold plants. However, there are already existing waste heat sources in the food production process, which can be utilised for cooling generation:

- Flue gas of natural gas or heating oil-driven heating generation (however, cooling down to utilisation temperature is often only possible; e.g. in bakeries)
- Plume from bakeries

5.5.2 Dairy farms

Dairy farms have a cooling demand for rapid-cooling, when fresh milk is cooled down from 35 °C to approx. 2 °C in a very short time period. A part of the cooling is carried out with a preheating of tap water through a preceding heat exchanger. The VDMA study ascertained a cooling energy demand of 1,049 GWh/a, yielding a final electrical energy demand of 583 GWh/a for providing the cooling. These numbers correspond to a 7.5 % share of the cooling demand and 8.5 % of the final energy demand of the food industry (Gruntram Preuß 2010).

5.6 Building air conditioning

Beside air control for thermal comfort, other applications such as cooling demand of data centres, production halls, and clean rooms are also assigned to be building air conditioning (Guntram Preuß 2011). A delimitation of the building air conditioning and air control of other commercial cooling applications (shopping centres, etc.) is impossible with the data from the VDMA study. For this purpose it is better to employ other studies, such as the pre-study for the eco-design directive of the European Union (Ecodesign 2008).

For the calculation of the cooling demand for air control, it has been assumed that the COP is 2.9, which is in agreement with DIN V 18599-7:2007 for multi-split systems and an average part-load factor of 1.0. The calculation for absorption-type chillers is based on a heat ratio of 0.71, which is also in agreement with DIN V 18599-7:2007 with the following conditions: H_2O-LiBr absorption type chiller, chilled water 6/14 °C, cooling water 27/33 °C, heating medium 80/70 °C (DIN-V-18599 2007).

Building air conditioning is thought to exhibit strong rates of increase between 3.5 and 5 %/a based on the installed cooling power (Ecodesign 2008) (see Figure 30), leading to an increasing importance in the future. Due to the lifetime of air condition systems of approx. 15 years it is advised to take care of sustainable sizing of the systems to be built (see Figure 31). Both Figure 30 and Figure 31 have been taken from (Ecodesign 2008) and show the trend very well.

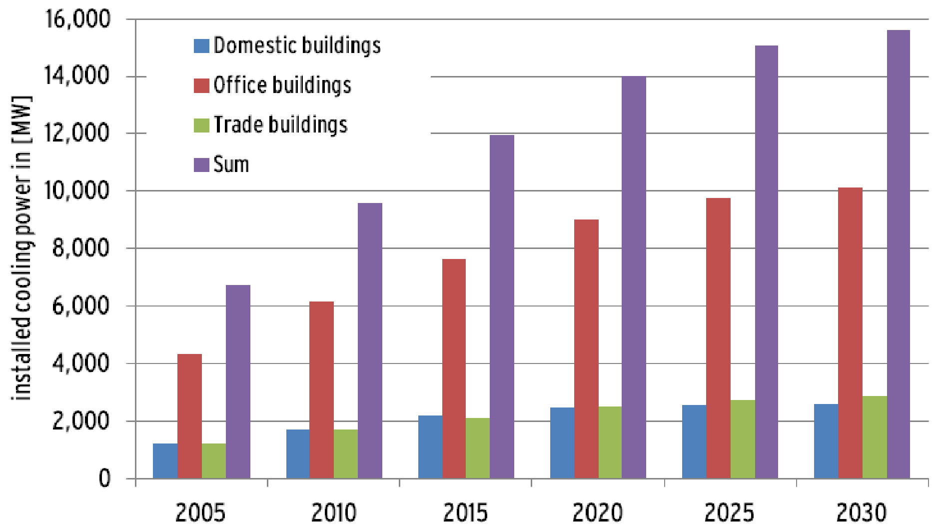


Figure 30: Development of the installed cooling power for building air conditioning (Ecodesign 2008)

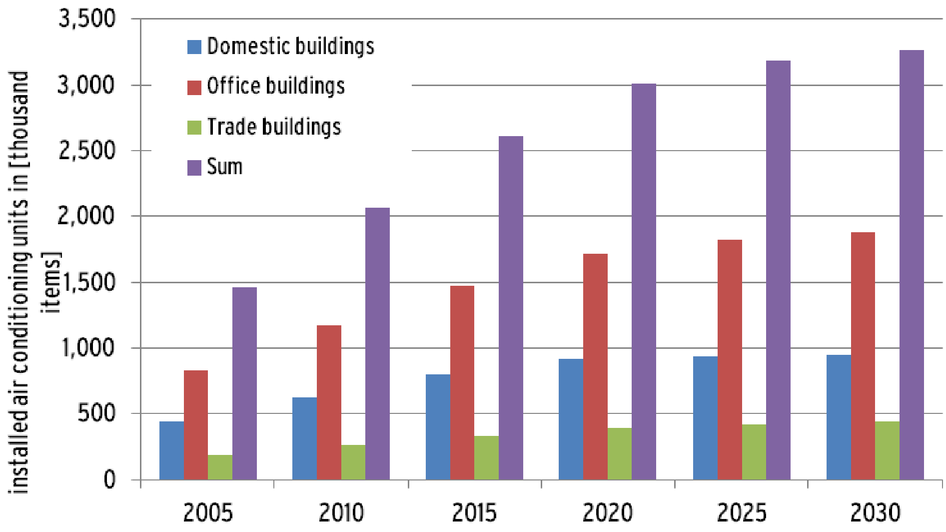


Figure 31: Development of the installed air conditioning systems (Ecodesign 2008)

5.7 Presentation of the results

Table 7: Cooling, final, and primary energy demand depending on application

Application	Cooling energy [GWh/a]			Final energy [GWh/a]			Primary energy [GWh/a]
	sum	thereof CRP	thereof ACh	sum	thereof CRP	thereof ACh	
Industrial refrigeration	18,296	16,831	1,465	15,070	13,030	2,091	35,376
Food production	14,091	13,845	246	7,258	6,766	493	17,935
Building air conditioning	30,786	30,299	487	11,134	10,448	686	27,645
Sum	**63,173**	**60,975**	**2,199**	**33,463**	**30,243**	**3,269**	**80,956**

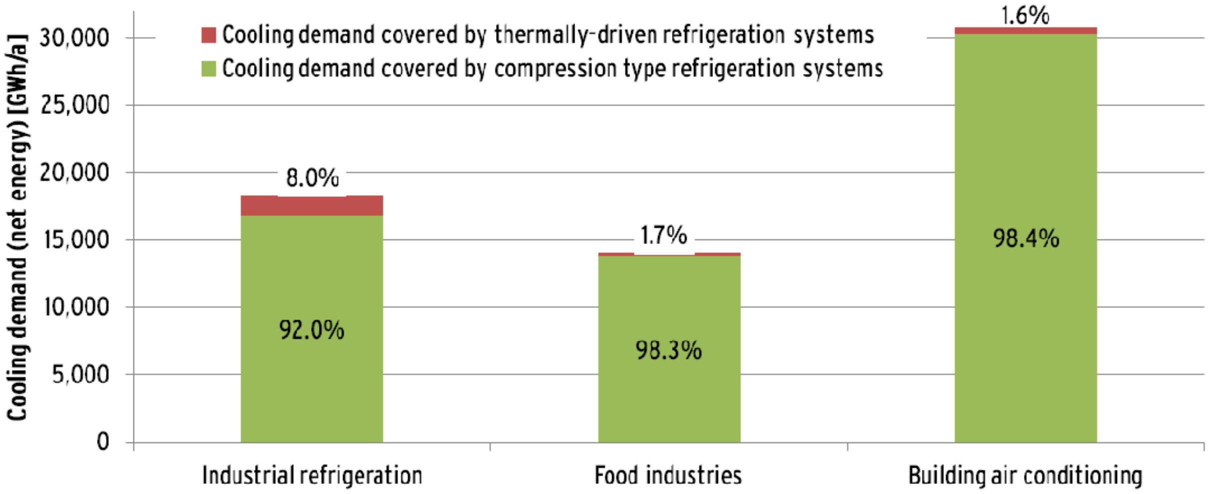


Figure 32: Cooling demand (net energy) of the industrial sectors

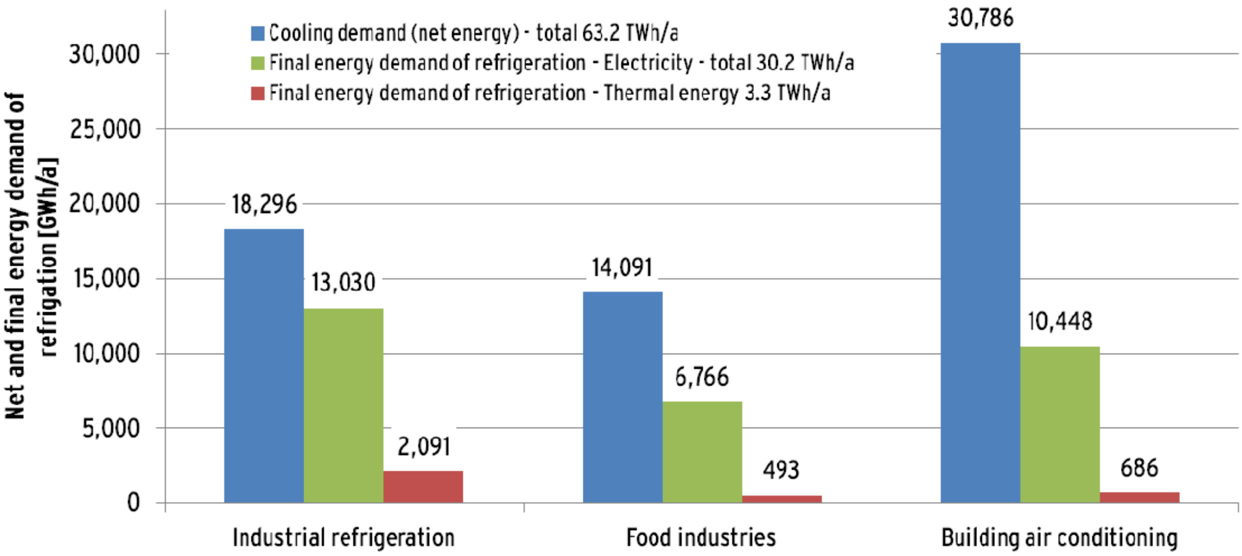


Figure 33: Cooling demand (used energy), and required electrical and thermal final energy demand of the industrial sectors

Table 8: Cooling, final, and primary energy demand depending on industrial sectors, incl. sub-groups

Temperatures are highlighted with green colour, which are suitable for single-staged H_2O-LiBr absorption type chillers.

Industrial sector			Cooling demand [GWh/a]				Fin. energy demand [GWh/a]			Prim. energy demand [GWh/a]
Major group	Minor group	Sub-group	Sum	There-of CRP	There -of ACh	At t_{demand} [°C]	Sum	Electricity	Heat	
Industrial refrigeration	Mining, exploitation of pit and quarry (only stone coal)		560	560	0	3	160	160	0	416
	Paper and pulp industry		252	252	0	6	90	90	0	269
	Printing industry		679	644	35	20	105	55	50	178
	Other chemicals industry without air and gas liquefaction		6,065	5,265	800	-4	3,090	1,950	1,140	5,868
	Other chemicals industry only air and gas liquefaction		1,120	1,120	0	-190	7,540	7,540	0	19,604
	Pharmaceutical industry		933	853	80	6	328	213	115	635
	Plastics and rubber industry		2,820	2,820	0	6	1,410	1,410	0	3,666
	Building and building materials industry		1	1	0	-25	1	1	0	2
	Electric and electronics industry (also conductor board production, soldering, and semiconductor production)		740	320	420	5	680	80	600	628
	Automotive industry		1,550	1,480	70	6	470	370	100	1,032
	Mechanical engineering		1,000	1,000	0	6	250	250	0	650
	Compressors, e.g., for compressed air generation, air desiccation (with refrigeration system)		376	376	0	3	94	94	0	244
	Process refrigeration (water chillers) for industrial applications		1,824	1,764	60	6	590	504	86	1,370
	Control cabinet cooling		376	376	0	20	313	313	0	814
Food production	Food industry	> 50 employees, normal cooling	5,314	5,208	106	0	1,805	1,627	177	4,355
		> 50 employees, freezing	3,543	3,472	71	-25	2,492	2,315	177	6,142
		20…49 employees, normal cooling	1,323	1,297	26	0	543	499	44	1,327
		20…49 employees, frrezinf	882	864	18	-25	764	720	44	1,904
		Other normal cooling	757	742	15	0	322	297	25	789
		Other freezing	504	494	10	-25	474	449	25	1,185
	Breweries		719	719	0	-7	276	276	0	717
	Dairy farms		1,049	1,049	0	4	583	583	0	1,516
Building air conditioning	Commercial, trade, service		9,052	8,677	375	6	3,520	2,992	528	8,149
	Industry		17,672	17,560	112	6	6,213	6,055	158	15,854
	Data centres, servers		3,335	3,335	0	6	1,150	1,150	0	2,990
	Domestic		728	728	0	6	251	251	0	653
Sum			**63,173**	**60,975**	**2,199**	**-2**[3]	**33,513**	**30,243**	**3,269**	**80,956**

[3] Average temperature of cooling generation, energy-content-based mean value

Table 9: Electricity and heat demand, and their coverage in different industrial sectors and sub-groups

Industrial sector			Electricity demand [GWh/a]	Heat demand [GWh/a]						
Major group	Minor group	Sub-group		Sum	Thereof district heat	thereof renew-able sources	Thereof stone and brown coal	Thereof heating oil	Thereof natural and other gases	Thereof others
Industrial refrigeration	Mining, exploitation of pit and quarry (only stone coal)		2,316	2,960	15	10	710	252	0	0
	Paper and pulp industry		22,350	45,050	4,230	8,660	3,910	1,420	2,6210	590
	Printing industry		1,100	0	0	0	0	0	0	0
	Other chemicals industry (without air and gas liquefaction)		36,290	100,650	24,650	7,010	2,710	3,620	4,695	2,250
	Other chemicals industry: only air and gas liquefaction		7,540	0	0	0	0	0	0	0
	Pharmaceutical industry									
	Plastics and rubber industry		14,150	8,500	1,310	70	80	1,340	5,590	40
	Building and building materials industry		8,740	52,870	100	7,260	18,300	7,900	14,900	5,220
	Electric and electronics industry (also conductor board production, soldering, and semiconductor production)									
	Automotive industry		19,640	16,780	4,040	50	370	1,080	1,1110	0
	Mechanical engineering		11,700	12,730	1,550	40	50	2,900	8,190	0
	Compressors, e.g., for compressed air generation, air desiccation (with refrigeration system)		14,000	0	0	0	0	0	0	0
	Process refrigeration (water chillers) for industrial applications		14,320	17,480	320	70	250	2,810	14,010	10
	Control cabinet cooling									
Food production	Food industry	> 50 employees, normal cooling	6,348	14,705	885	270	889	2,370	10,156	143
		> 50 employees, freezing	6,348	14,705	885	270	889	2,370	10,156	143
		20…49 employees, normal cooling	1,247	2,889	174	53	175	466	1,993	28
		20…49 employees, freezing	1,247	2,889	174	53	175	466	1,993	28
		Other normal cooling	400	926	56	17	56	149	639	9
		Other freezing	400	926	56	17	56	149	639	9
	Breweries		1,070	2,660	60	20	110	460	2,020	0
	Dairy farms									
Building air conditioning	Commercial, trade, service		140,260	240,980	32,840	3,770	3,250	93,070	108,050	0
	Industry		199,820	440,660	42,170	27,250	88,080	39,490	222,630	21,040
	Data centres, servers									
	Domestic		139,200	798,860	81,850	73,380	13,210	255,100	365,920	9,400
Sum			648,486	1,777,220	195,365	128,270	133,270	415,412	808,901	38,910

6 Market overview of climate-friendly cooling technologies

6.1 Classification of air conditioning systems and HVAC systems

A variety of technical concepts exist for the air conditioning and cooling (partial air control without humidity control) of buildings. The standard DIN V 18599-7 (DIN-V-18599 2007) provides the system of equations for the implementation of the EnEV 2009 (EnEV 2009). Herein, the VHAC systems are classified as illustrated in Figure 34.

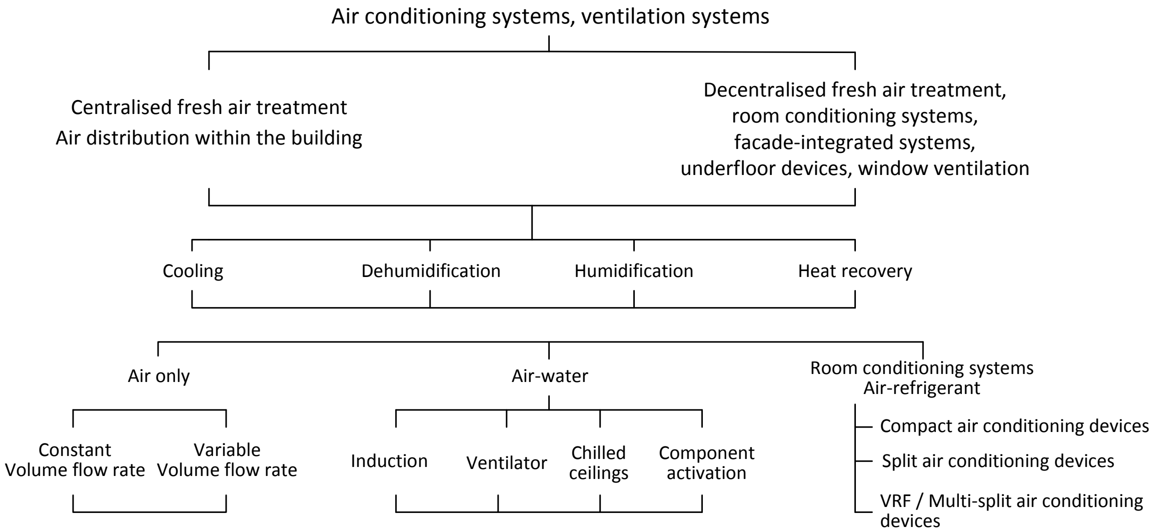


Figure 34: Classification of HVAC systems in non-residential buildings (DIN-V-18599 2007)

For the assessment of the climate-friendliness of various systems, there are multiple starting-points, which refer on system details (microscopic influences) or to the system concept (macroscopic influences). For instance, high impact system details are the refrigerant with its direct and indirect influence on the CO_2 emissions (TEWI), the refrigerant charge, and the transferring area of heat exchangers with their realisable temperatures. The positive and negative effects of various systems regarding their impact on the climate and energy efficiency are displayed in Table 10.

Table 10: Impact of various system concepts related to their climate friendliness

Negative effects	System concept	Positive effects
Large amount of refrigerant	**Direct evaporation**	Saving of an additional fluid circuit and a heat exchanger → minimal cooling generation temperature closer to temperature of utilisation; quicker controllability with adaption of the boundary conditions
Large amount of refrigerant	**Air-cooled**	Saving of an additional fluid circuit and a heat exchanger → maximum refrigerant temperature closer to ambient temperature
Large amount of refrigerant	**Direct evaporation and air-cooled**	Free cooling possible (with recirculating refrigerant)

Additional pumping effort; additional heat transfer → minimal cooling generation temperature is smaller than in direct evaporation system; no free-cooling possible	**Additional cooling medium**	Small amount of refrigerant
Requires moveable parts in compressor → reduced life-time	**Compression type chiller**	Large energy efficiency at small temperature differences; power easily adjustable
Compared to compression type chillers more waste heat; low energy efficiency at small temperature differences	**Thermally driven chiller**	Utilisation of waste heat possible; thermal compressor → no moveable parts required
Desiccation to be avoided → lower bound for temperature (dew point) and power; larger cooling demand due to radiative heat transfer with walls (yields lower wall temperatures → large heat flow from outside)	**Chilled ceiling for cold distribution**	no energy for fans necessary, larger air temperature at same effective temperature (sensible temperature, average of air and radiative temperature); quiet, low air velocity
Large energy demand for ventilator if solely applied as HVAC system	**DEC or open sorption process**	Waste or solar heat for driving desiccation

Beside the classification according to DIN V 18599, it is possible to re-classify the sub-category of air control (see Figure 35). In many points, it is possible to obtain a higher degree of detail. With thermally driven refrigeration systems, open and closed processes have to be distinguished. An overview is provided in Figure 36.

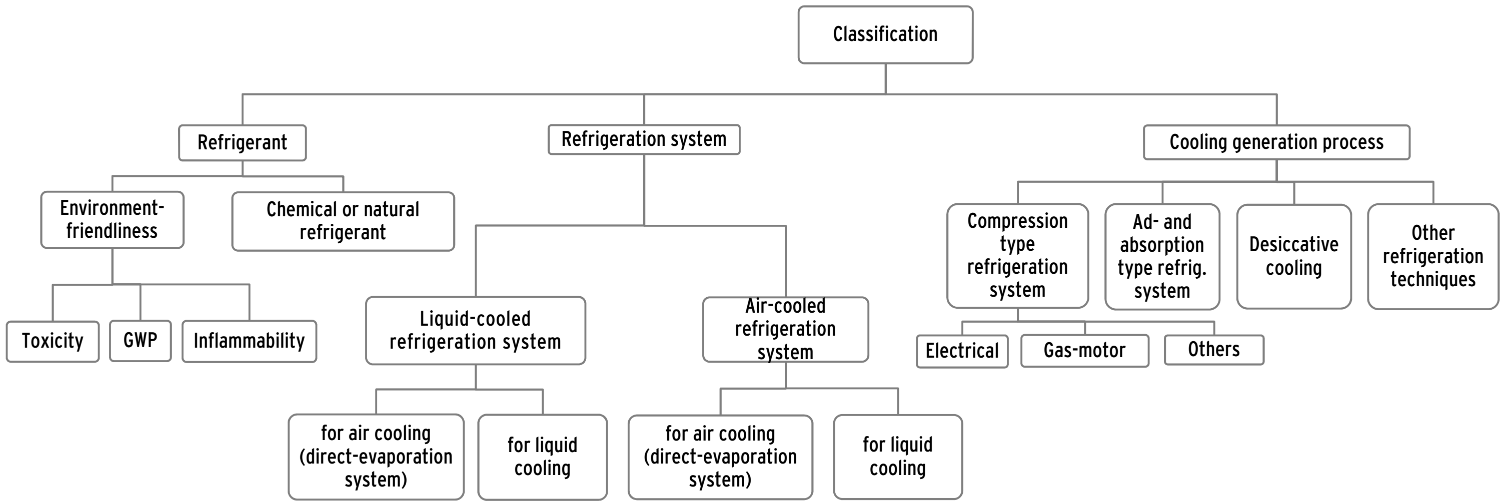


Figure 35: Classification of air control systems (cooling only)

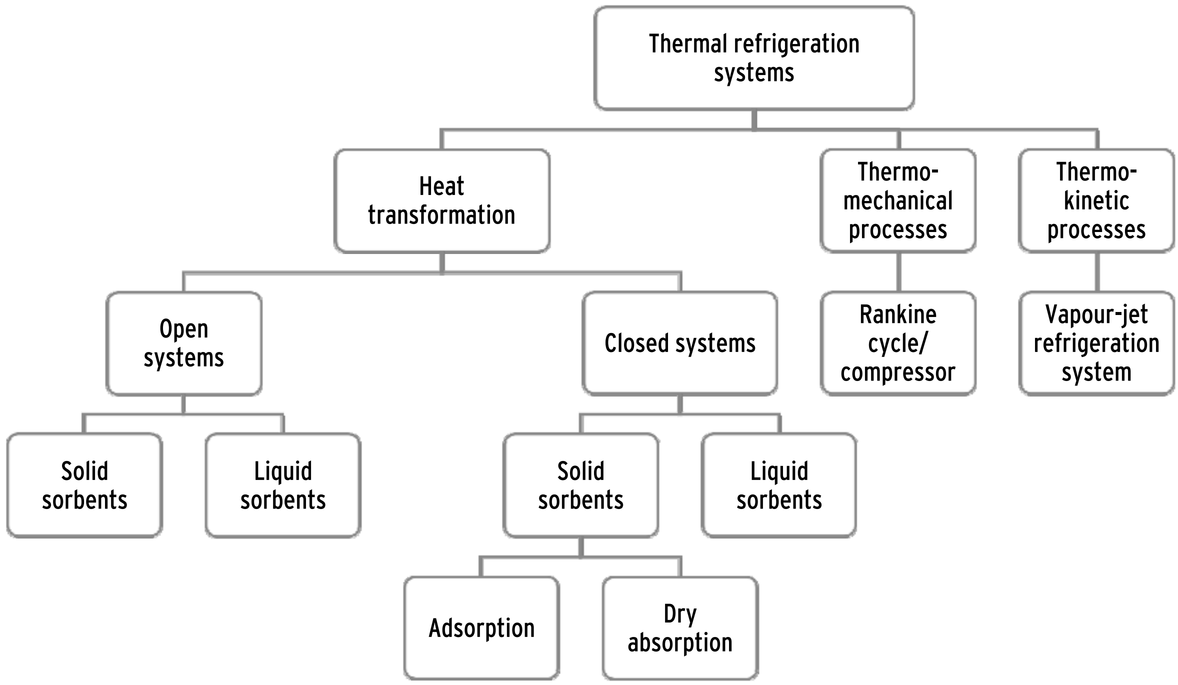


Figure 36: Classification of thermally driven cooling generation processes

The different processes for cooling generation are presented in the following subsections.

For compression type chillers, the effect of the refrigerant on the climate-friendliness is discussed, and a market overview of compressors and chiller for natural refrigerants provided. Finally, a table summarises installed refrigeration systems with natural refrigerants.

The explanation of thermally driven refrigeration systems is limited to closed systems of both ad- and absorption types, which are mostly employed for air conditioning and cooling purposes. Special emphasis is on set-up variants and refrigerant-absorbent combination with their influence on the system behaviour and the obtainable heat ratio. Afterwards, a market overview illustrates already installed system concepts. Moreover, closed system processes, which are chiefly employed for building air conditioning, are explained.

6.2 Compression type chillers

6.2.1 Comparison of refrigerants

The properties of the refrigerant influence both the energetic efficiency and the construction of the refrigeration system regarding pressures, size, and compression principle. An exchange of the refrigerant in already existing systems is only limitedly possible (see so-called drop-in refrigerants[4]). Figure 25 on page 39 provides an overview of both the global warming impact and the safety classification according to DIN EN 378 of common refrigerants. The now following figures provide information about the saturation pressure dependence on the evaporation temperature (Figure 37), the critical temperature and pressure (Figure 38), the cooling-power-specific suction flow rates (Figure 39), and the pressure ratio depending on the evaporation temperature (Figure 40). Fluorinated refrigerants are indicated with red (HFC) and orange (non-saturated HFC) colour, whilst natural refrigerants have blue colour.

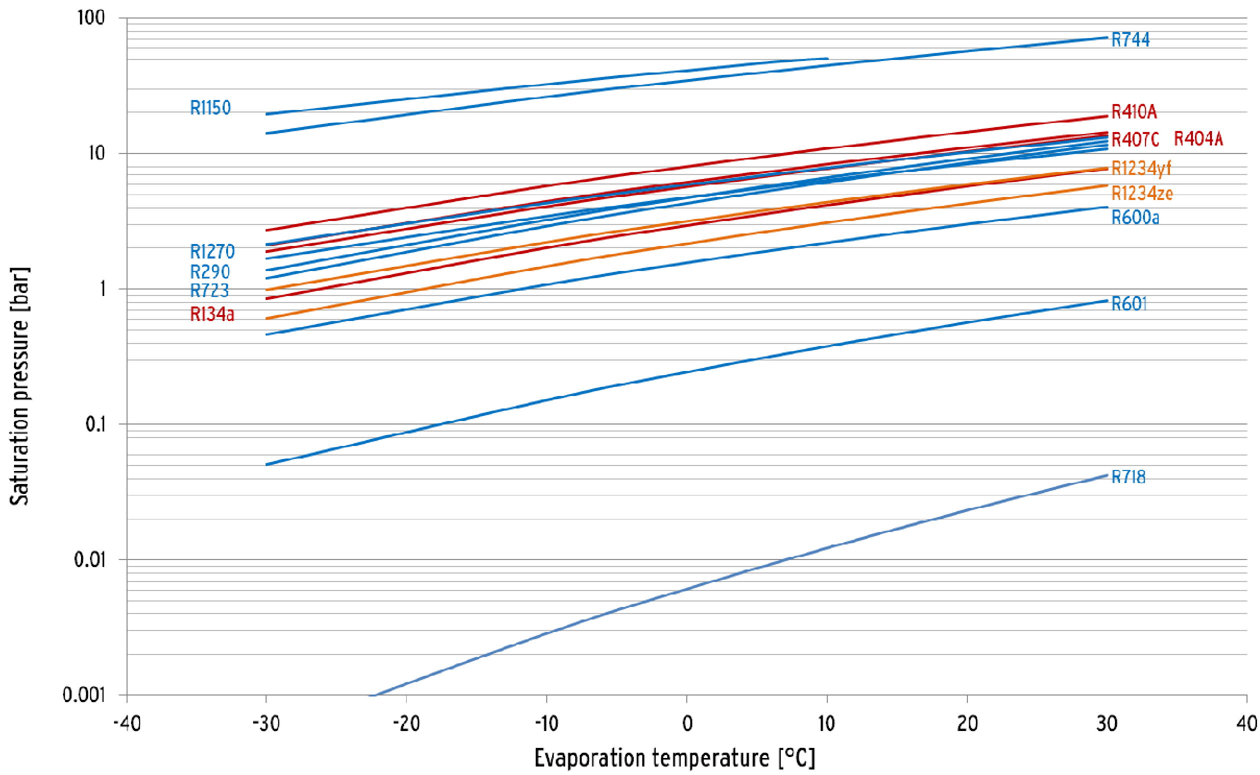


Figure 37: Saturation pressure of various refrigerants (natural (blue), HCF (red), non-saturated HFC (orange))

[4] Drop-in refrigerants are refrigerants which are suitable to substitute another refrigerant directly (e.g., a HCF refrigerant, which has been prohibited in the Montreal protocol) and have been chosen by refrigerant and compressor manufacturers. After refilling, evacuation, and oil exchange it is possible to utilise the chiller without change of the major process parameters, such as pressure and cooling power. Common examples are R134a for R12, and R507 for R502.

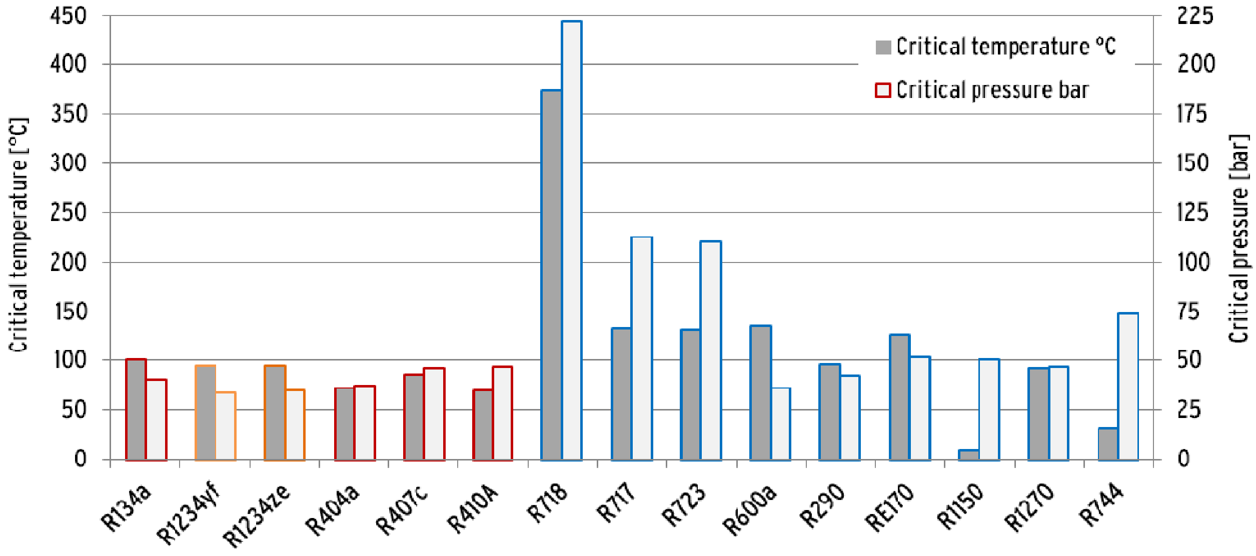


Figure 38: Critical temperature and pressure of various refrigerants (natural (blue border), HFC (red border), non-saturated HFC (orange border))

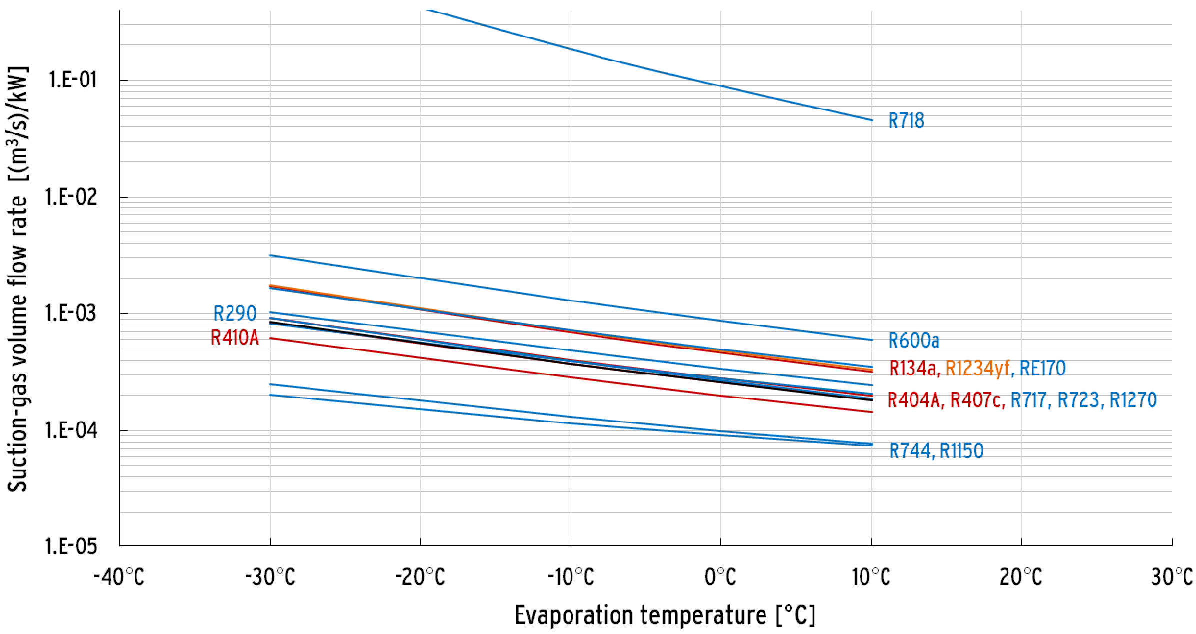


Figure 39: Suction volume rate referred to cooling power versus the evaporation temperature

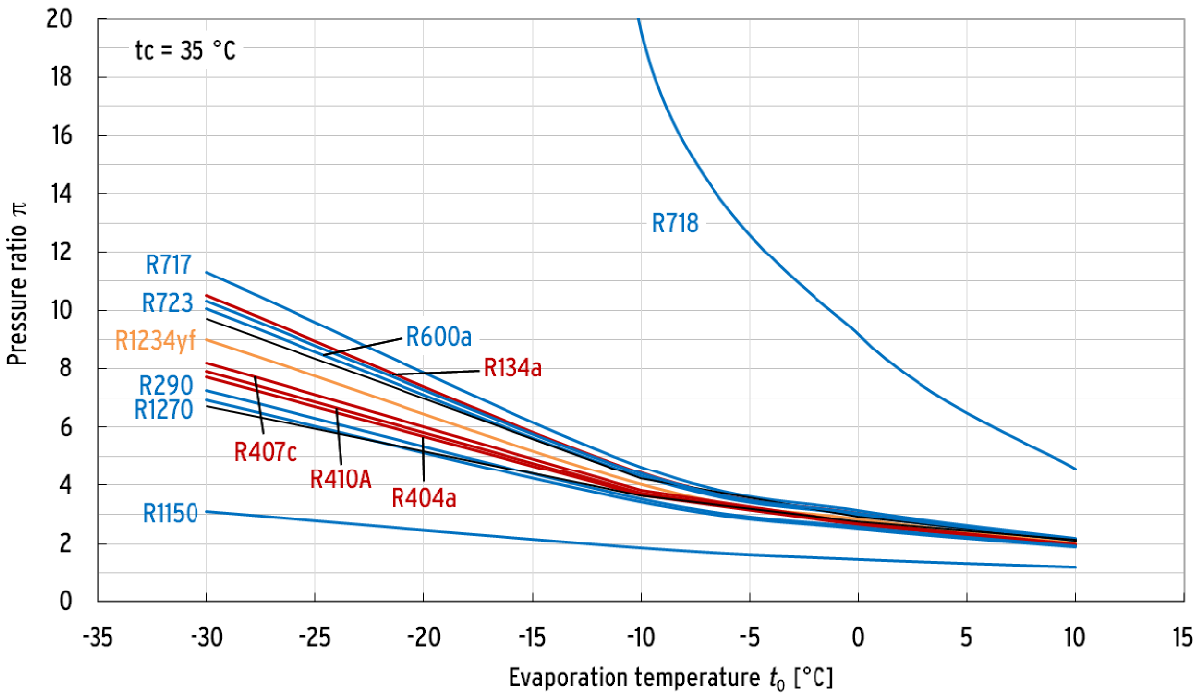


Figure 40: Pressure ratio of condensation and evaporation pressure versus the evaporation temperature at a condensation temperature of 35 °C

In order to estimate the energy efficiency ratios for different refrigerants, a single-staged thermodynamic cycle is simulated (Figure 41). It has to be stressed that the results are approximate values. Refrigerant-specific modifications of the system can yield better efficiency ratios. For instance, modifications such as internal heat exchanger for some refrigerants (e.g., R600a, isobutane) provide a large potential for increasing the energy efficiency.

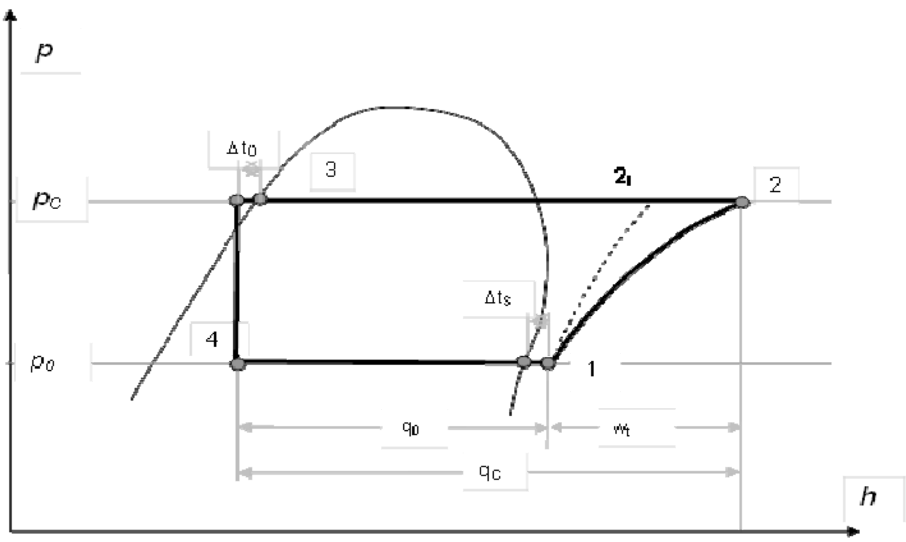


Figure 41: Thermodynamic cycle with single-staged compressor for calculation of the process energy efficiency

Figure 42 illustrates the coefficients of performance for various evaporation temperatures of a single-staged thermodynamic cycle with a condensation temperature of 30 °C and a turbine isentropic efficiency factor of 0.7. The sub-cooling of the liquid is assumed zero, whilst super-heating of the vapour (Δt_0) is 1 K. Such a super-heating is typical for refrigeration system with flooded evaporator. The coefficients of performance are related to those of the HFC R134a in Figure 43.

The saturation temperature is coupled to the saturation pressure in sub-critical processes, where the temperature is below the critical temperature (see Figure 38, left ordinate), which is true for most of the refrigerants. However, in transcritical processes, as it can be observed with R744 (carbon dioxide) and R1150 (ethene) due to their low critical temperatures compared to ambient conditions in Germany, the pressure becomes independent of the temperature. In this case, the pressure is a free parameter, which has to be adjusted properly by a controller. A comparison of the coefficients of performance with sub-critical processes is not useful any more, which is the reason why its expression for trans-critical process is foregone in the following. Due to the process characteristics, it is common that the thermodynamic process differs from the ideal one presented in Figure 41. For instance, modifications might be a so-called booster set-up, various kinds of heat recovery, or intra-process utilisation of the expansion power. Depending on the choice of various parameters, a vast range of results arises.

Figure 42 and Figure 43 indicate clearly that the utilisation of HFC refrigerants cannot be justified with their energy efficiency compared to natural refrigerants. Already in single-staged processes, many natural refrigerants are superior to the common refrigerants R134a, R404A, and R410A. This effect is even more evident with some natural refrigerant, such as R600a or R290, by employing an internal heat exchanger. Especially, R1234yf, which is considered a substitute for R134a in mobile air conditioning systems, has a significantly poorer performance than natural refrigerants. RE170 (dimethyl ether) is presently only employed for azeotropic mixtures in R723. The official listing as refrigerant by ASHRAE[5] has not been carried out yet due to an incomplete toxicological rating (Germanus 2011). Nonetheless, RE170 provides the largest calculated coefficients of performance in single-staged processes, which are also confirmed in recent publications (Baskaran & Mathews 2012; Cox et al. 2009). However, the reasons why it has not been employed purely are not mentioned in the publications. High explosiveness and material incompatibility (only few materials for sealing are possible) might be the reason.

For the cases of large pressure ratio and refrigerants with a gently inclined isentropic line (high discharge temperature), it is useful to have a double-staged compression with intermediate cooling as illustrated in Figure 44. The resulting coefficients of performance are presented in Figure 45. Figure 46 provides a comparison to a single-staged process with R134a.

[5] ASHRAE: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. is the American pendant of the IIR. ASHRAE provides an official listing of authorised refrigerants.

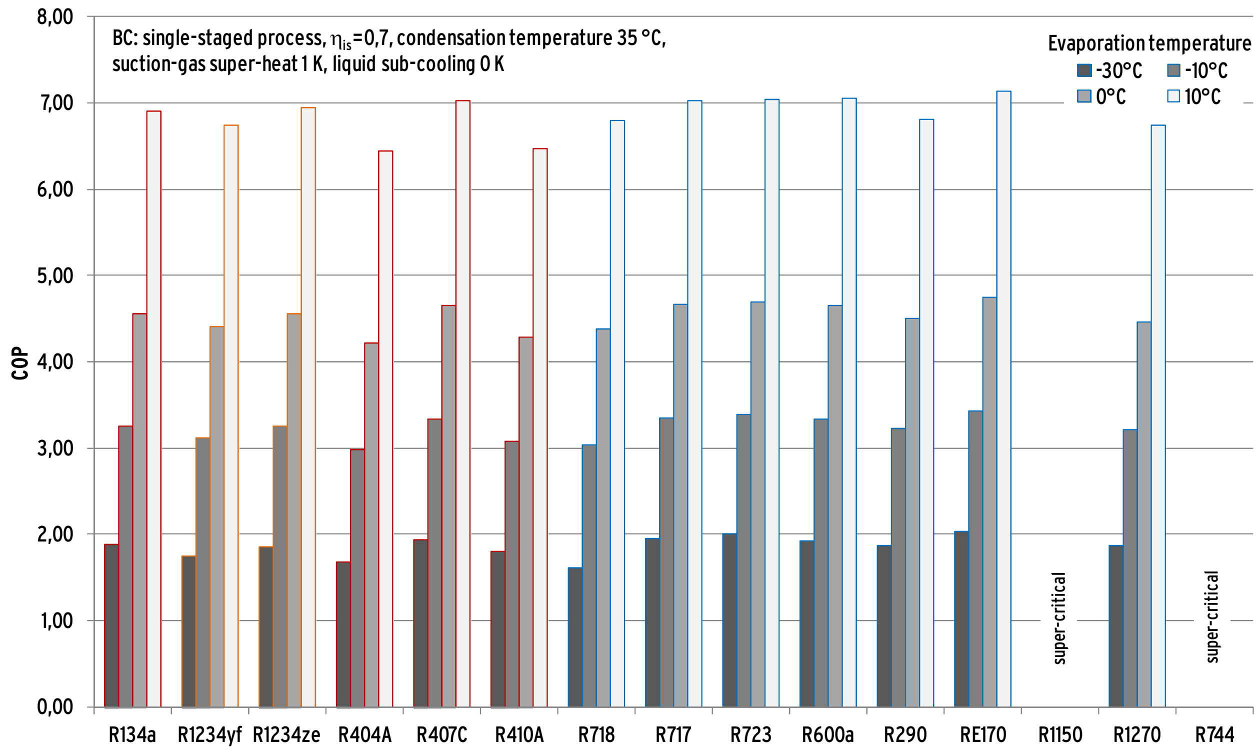


Figure 42: Coefficients of performance for single-staged processes of various refrigerants

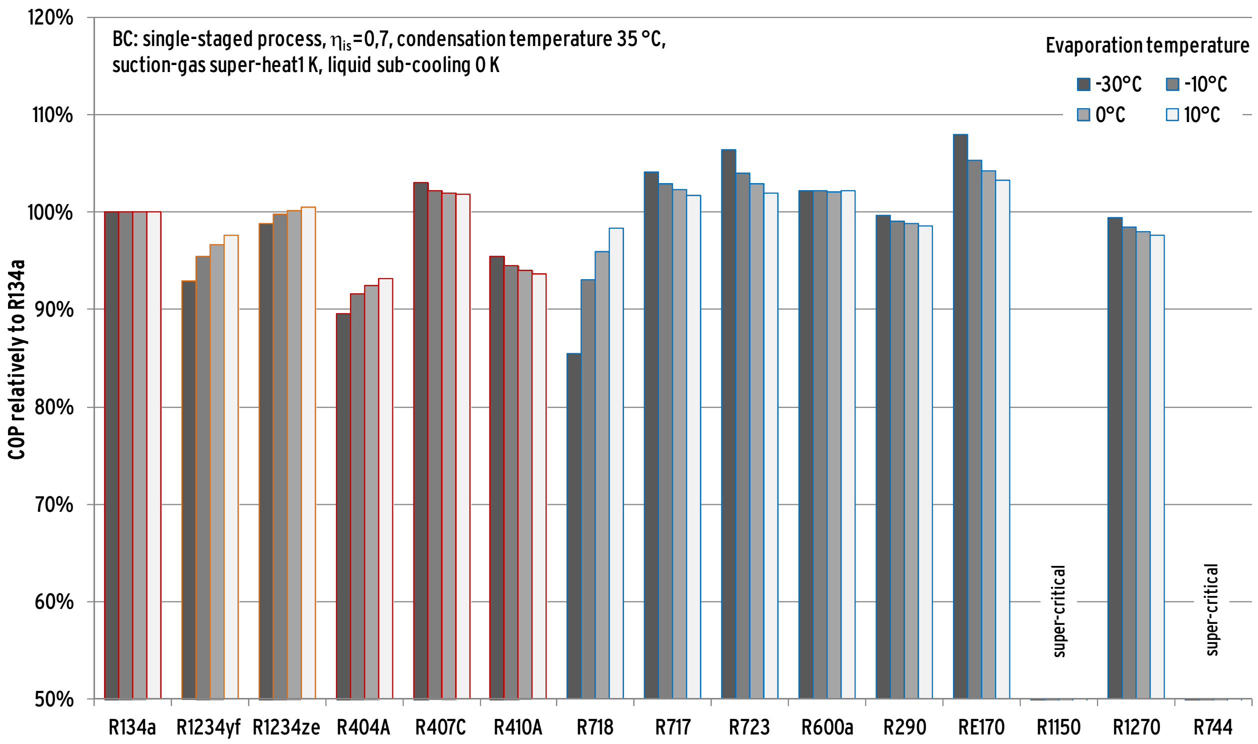


Figure 43: Relative coefficients of performance of single-staged processes of various refrigerants compared to R134a

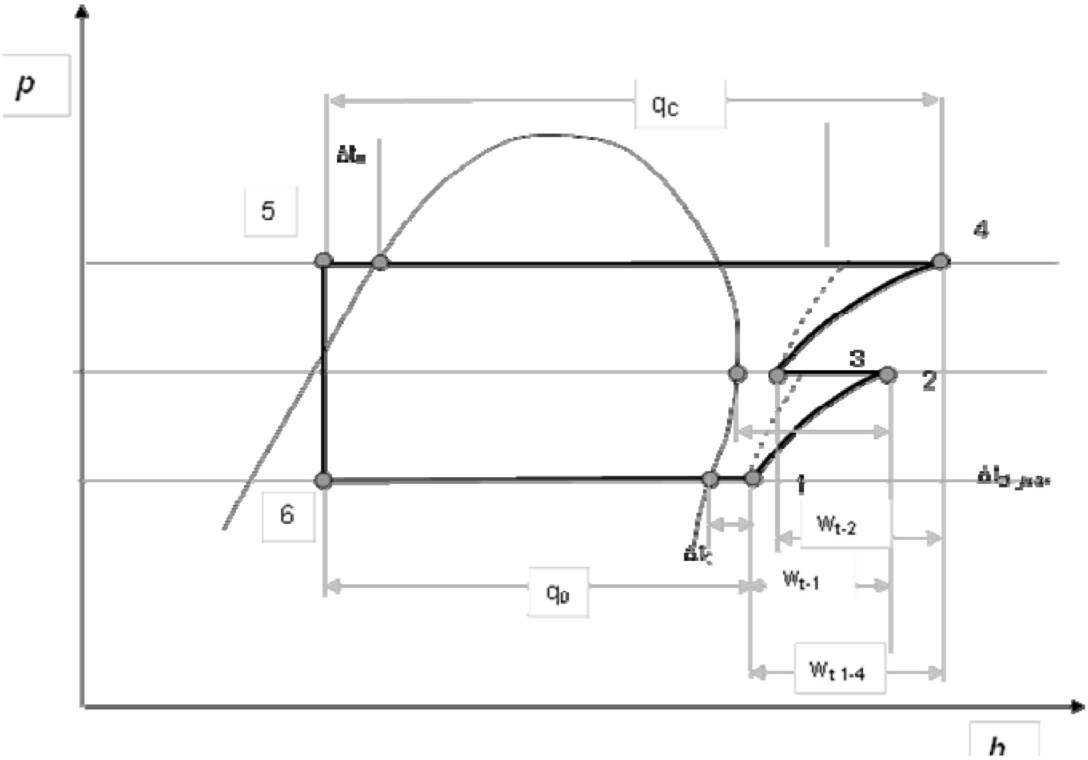


Figure 44: Double-staged compression with intermediate cooling

Comparing the coefficients of performance of double-staged compression processes, clear advantages of the refrigerants R718 (water), R717 (ammonia), R723 (ammonia and dimethyl ether), and RE170 (dimethyl ether) become visible obverse to all HFC refrigerants. The coefficient of performance of those natural refrigerants supersedes those of the best HCFs (R134a and R407C) by 10 % in average.

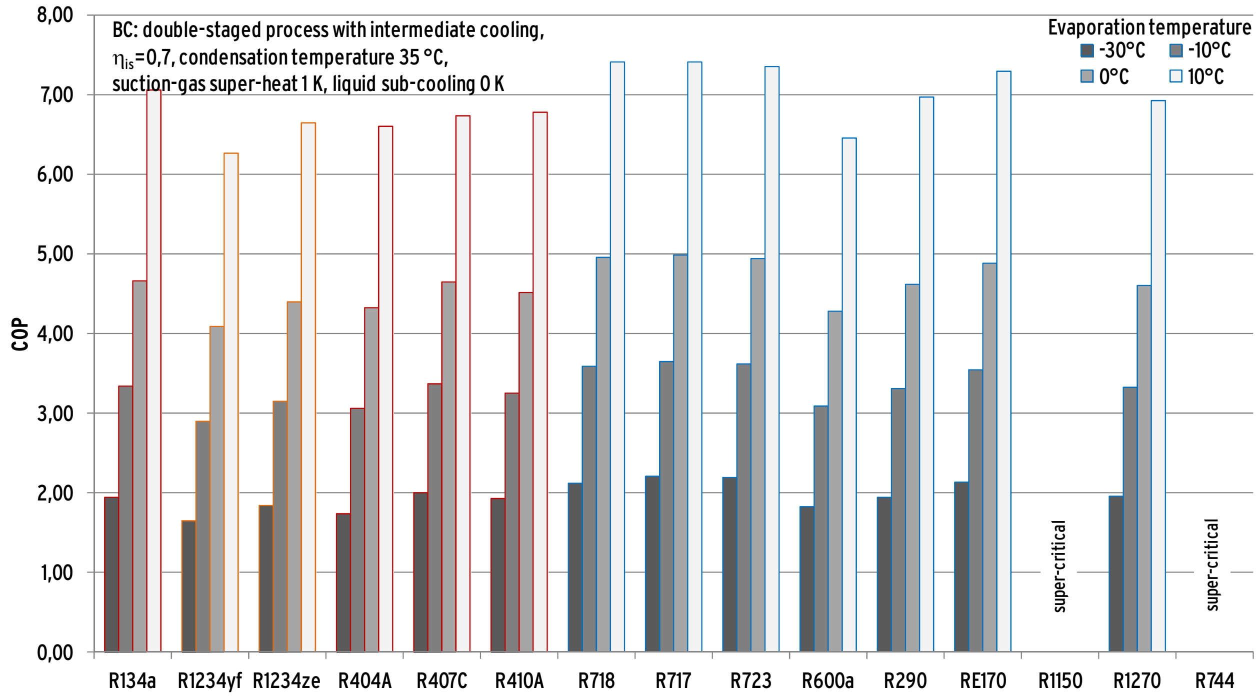


Figure 45: Coefficients of performance for a double-staged process with intermediate cooling for various refrigerants

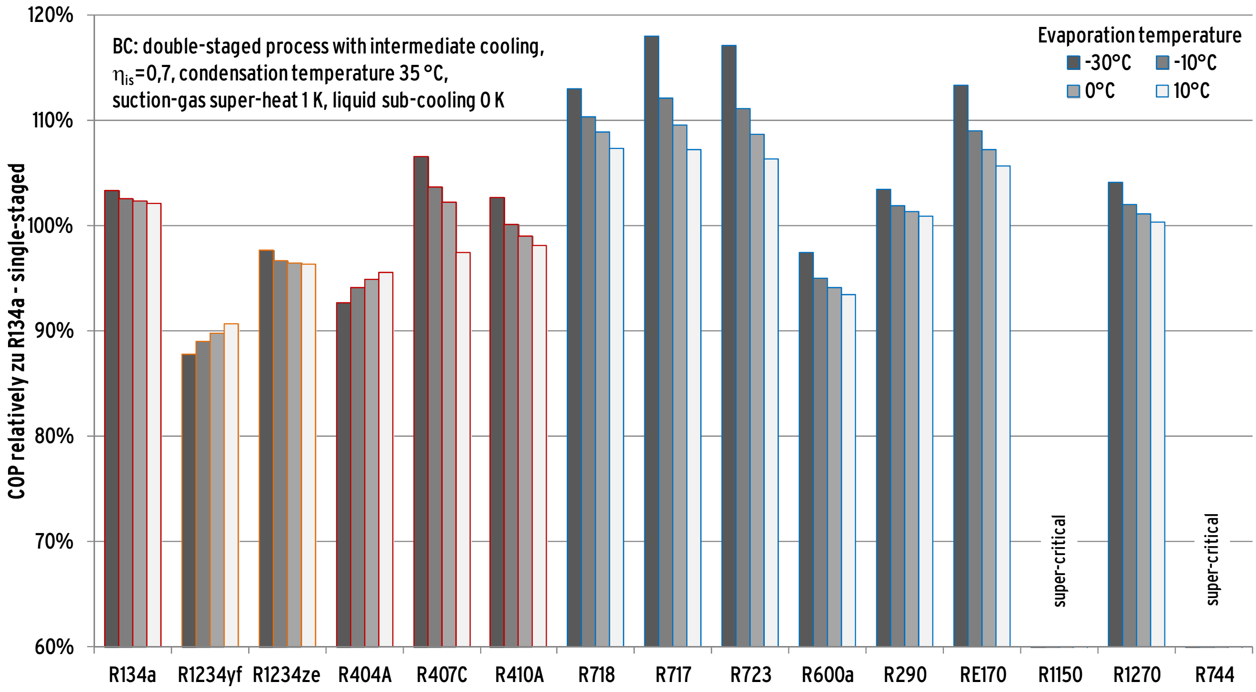


Figure 46: Relative coefficient of performance of a double-staged process with intermediate cooling compared to a single-staged process with R134a

6.2.2 Compression type refrigeration plant – Compressors for natural refrigerants

The compressor is technically the major part in a compression type chiller. For the set-up of chillers with natural refrigerant, the availability of compressors being accredited for the particular refrigerant is the major limiting factor. Within the scope of this study, a database[6] of available compressors has been raised. It allows researching the availability at boundary conditions to be defined and comprises 370 compressors from 30 series of nine producers. The following illustrations provide an overview of the compressors depending on refrigerant and power range.

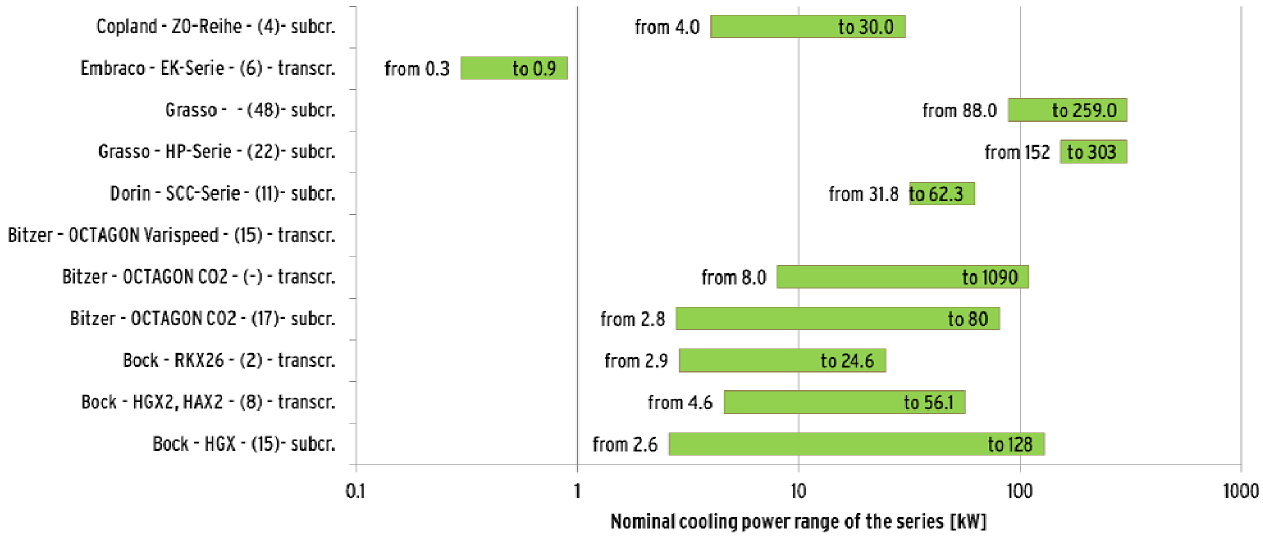


Figure 47: Range of cooling power of available compressor for R744 (CO_2) (without claim to be complete)

[6] A publication of the database by the ILK Dresden has not been conducted yet, but it is considered to be carried out in the future.

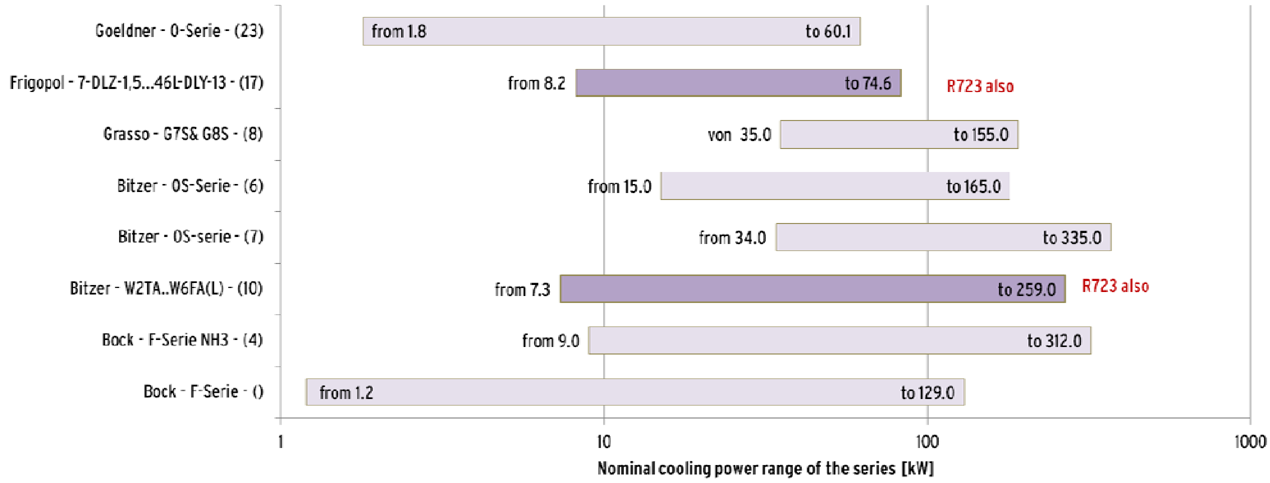


Figure 48: Range of cooling power of available compressor for R717 (NH_3) (without claim to be complete)

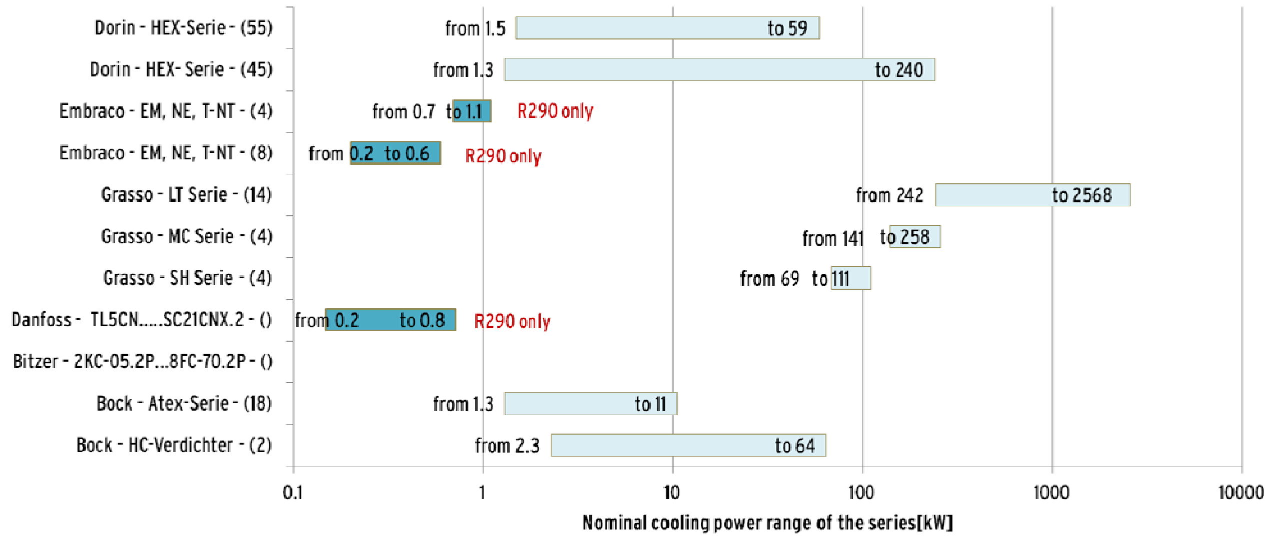


Figure 49: Range of cooling power of available compressor for pure hydrocarbons – R290 (propane) and R1270 (propene) (without claim to be complete)

6.2.3 Compression type refrigeration plant – chillers

Compared to the large number of available compressors for systems with natural refrigerants in a large power range, the number of available chillers is small. The refrigerants R717 (ammonia), R290 (propane), and the mixture R723 are employed for this kind of plants. It is visualised in Figure 50 that ammonia systems are chiefly offered for a larger power. However, one shall consider that systems reviewed here, are series products, whilst many systems for industrial refrigeration are designed and built for a specific application; e.g., for freezing in the food industry, where R744 is also utilised. This fact is also illustrated in Table 11, where some systems with natural refrigerants are listed.

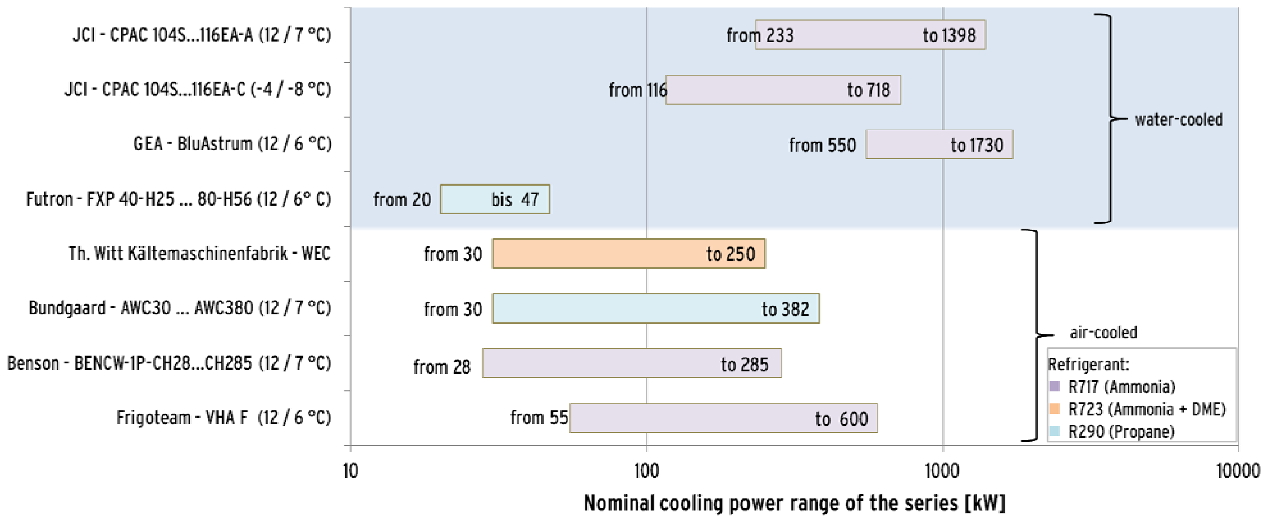


Figure 50: Range of cooling power of available chillers with natural refrigerants (without claim to be complete)

Beside chillers, there are further compression type refrigeration plants with natural refrigerants available in series production. Among those are:

- Multi-split system with CO_2 (R744): Daikin has presented a multi-split air conditioning system with CO_2 as refrigerant in 2008. However, the coefficient of performance remains below standard systems utilising R410A. Presently, there are seven serially produced units (FXSN20AV until FXSN100AV1) available with nominal cooling powers between 2.2 and 11.2 kW. Other producers of multi-split systems with R744 are not known.
- Single-split systems with propane (R290): First serial-products with propane are sold by *Godrej* (India) (hydrocarbons21.com 2012b), which cover nominal cooling powers between 3.4 and 4.9 kW. Further systems with R290 by the Chinese company *Gree Electric Appliance* reached the series production readiness. However, the production did not start by now (as of June 2013). The product has a nominal cooling power of 2.6 kW and is VDE certified.

6.2.4 Existing refrigeration systems with natural refrigerants

As already indicated in subsection 6.2.2 on p. 68, many industrial refrigeration systems are designed and built for their particular application. Table 11 provides an overview of some reference plants utilising natural refrigerants, including information about location, date of commissioning, type and amount of refrigerant, industrial sector and installed cooling power. The data has been taken from articles in KI Kälte – Luft – Klimatechnik, KKA Kälte Klima Aktuell, KK Die Kälte & Klimatechnik, and the websites www.cci-dialog.de/branchenticker/ and www.eurammon.com.

Table 11: Examples of realised refrigeration systems with natural refrigerants (no responsibility is taken for the correctness of this information)

Name	Commissioning	Refrigerant								Industrial sector				Nominal cooling power [kW]
		R290	R601	R717	R718	R723	R744	R1270	Charge size [kg]	Industrial refrigeration	Food production	Building air conditioning	Others	
Air conditioning at Greenpeace	1998		X						6		X			30
Versandschlächterei Neumarkt/Obpf.	1998			X					480		X			400
Stadtwerke Gera	1998								-				X	5,000
SB-Warenhäuser	1998			X					400				X	276
Refrigeration system J. Bauer KG	1999			X					1,400		X			-
Eiswerk Fischindustrie Heiligenhafen	1999			X					-		X			295
Food storage Luxemburg	1999			X					-		X			400
Super market distribution centre Brandenburg	1999			X					-		X			800
Obstbau-Versuchsgut Heuchlingen	1999			X					46		X			75
Parliament Nordrhein-Westfalen	1999			X					280			X		680
Fleischkombinat Kostelecke uzeniny a.s.	1999			X					-		X			1,800
Air conditioning CargoLux	1999				X				-	X				400
Air conditioning University Essen	1999				X				-			X		800
Berlin-Ostbahnhof	2000			X					-			X		1,250
Deep freezing storage for pastries	2000			X					56				X	180
Allgäuer Brauhaus	2000			X					700		X			680
Hermannsdorfer Landwerkstätten Hannover	2000			X					130		X			360
Nestle deep freeze storage Beauvais/F	2000			X			X		6,000/ 1.00		X			1,200
Danone Ochsenfurt	2001			X					1,000		X			2,400
Frostung und Kühlung M+P Tiefkühlkost	2001			X			X		2,000		X			475
Nestle Product Technology Centre	2001			X					7			X		230
Carbon dioxide recovery of Yara	2001			X					-		X			200
Sachsenmilch	2001			X					3,000		X			10,600
Froster und Tiefkühllager Bielefeld	2001			X			X		1,200/ 150		X			475
M+P Tiefkühlkost Bielefeld	2001			X			X		1,200/ 150		X			600
Nestle coffee freeze-drying Hayes/GB	2001			X			X		-		X			2,400
Air conditioning VW Manufacture Dresden	2001				X				-	X				2,000
Berglandmilch	2002			X					9,000		X			3,500
Office building Frigopol	2002					X			30			X		60
Cooling and storage of fruit	2002			X					1,800		X			460
Pizza Wagner	2002			X					3,000		X			1,070
Arianezentrum Ottobrunn	2002	X							34	X				1,200
KÜBA Wärmeübertrager-Testzentrum	2002			X					-			X		-

Name	Commissioning	Refrigerant								Industrial sector				Nominal cooling power [KW]
		R290	R601	R717	R718	R723	R744	R1270	Charge size [kg]	Industrial refrigeration	Food production	Building air conditioning	Others	
Alpincenter Bottrop	2002			X					-				X	1,400
Airport Stuttgart	2003			X					-			X		2,300
Fischverarbeitungszentrum Sassnitz	2003			X			X		21,000/ 12,000		X			5,150
Kunsteisbahn Ravensburg	2003			X			X		7,000/ 600				X	618
Brauerei Zipf / Austria	2004			X					500		X			570
Pasta & Co	2004			X			X		435		X			325
Candies manufacturer	2004			X					1,800		X			4,300
Yoghurt processing	2004					X			18		X			230
Milchunion Hocheifel Pronsfeld	2004			X					12,000		X			3,660
Tube station Marienplatz, Munich	2004			X			X		-	X				550
Deep freeze storage Galliker Transport AG, Dagmersellen/CH	2004			X					-		X			1,200
Poultry producer	2005			X					2,850		X			2,600
Meat processing Edeka	2006			X					10,000		X			5,500
Snow Dome Bispingen	2006			X					1,200				X	2,280
CC GROWA, Bern	2006						X		400		X			140
Atlantic Pelagic Seafood, Ship	2006			X			X		10,000/ 1,500		X			6,500
Mail handling centre Müllingen Switzerland	2007			X					880			X		4,300
Deep freeze storage Frigosuisse, Switzerland	2007			X					2,100		X			540
Aktiv-Hotel, Altis/Slowakei	2007					X			15				X	157
Lekkerland Stapefeld NK	2008			X					400		X			600
Lekkerland Stapefeld TK	2008			X			X		350		X			210
Lekkerland Oberhausen NK	2008			X					400		X			600
Lekkerland Oberhausen TK	2008			X			X		350		X			210
Lekkerland Allershausen NK	2008			X					400		X			600
Lekkerland Allershausen TK	2008			X			X		350		X			165
Gartner KG, Kehl	2008			X					365		X			853
Franken-Gut (EDEKA), Trunstadt	2008			X			X		-		X			1,160
Ozeaneum, Stralsund	2008			X					-			X		900
Südbayerische Fleischwaren GmbH, Obertraubling	2009					X			-		X			300
Magnetic resonance tomography, Ärztezentrum Bad Reichenhall	2009					X			-					-
Danish Crown, Fleisch-Zerlegebetrieb, Boizenburg	2009			X					-		X			500
EDEKA Fleischwerk Rheinstetten	2010			X					-		X			9,000
Logistics centre Lekkerland, Großbeeren/Berlin	2010			X			X		400/ 550		X			650
Netto Denmark	2010						X		-		X			1,430

Name	Commissioning	Refrigerant								Industrial sector				Nominal cooling power [kW]
		R290	R601	R717	R718	R723	R744	R1270	Charge size [kg]	Industrial refrigeration	Food production	Building air conditioning	Others	
Motor test facility TH Zwickau	2011				X				-	X				50
Steweag-Steg Graz, Schaltwarte	-					X			-			X		-
Wolf-ButterBack, Fürth	-						X		-		X			1,000
Poultry butchery Georg Stolle, Neutrebbin	-			X					250		X			1,750
Poultry butchery Wiesenhof, Lohne, Zentrale A	-			X					-		X			1,400
Poultry butchery Wiesenhof, Lohne, Zentrale B	-			X					-		X			1,860
Emmi Dagmersellen	-								-		X			0
Dachser storage, Langenhagen/Hannover	-			X					600		X			1,080
Phillips Petroleum, crude oil processing ship	-	X							-	X				5,000

The following figures illustrate the data of Table 11 by means of refrigerant and industrial sector. It can be observed in Figure 51 that approx. three quarter of the installed cooling power is provided by ammonia systems. Approximately 18 % of the installed cooling power is provided by CO_2 systems, whereas the shares of propane, water, R723, and n-pentane are small. The majority of the cooling capacity is installed in the food production sector (Figure 52). A further itemisation of the food sector is displayed in Figure 53.

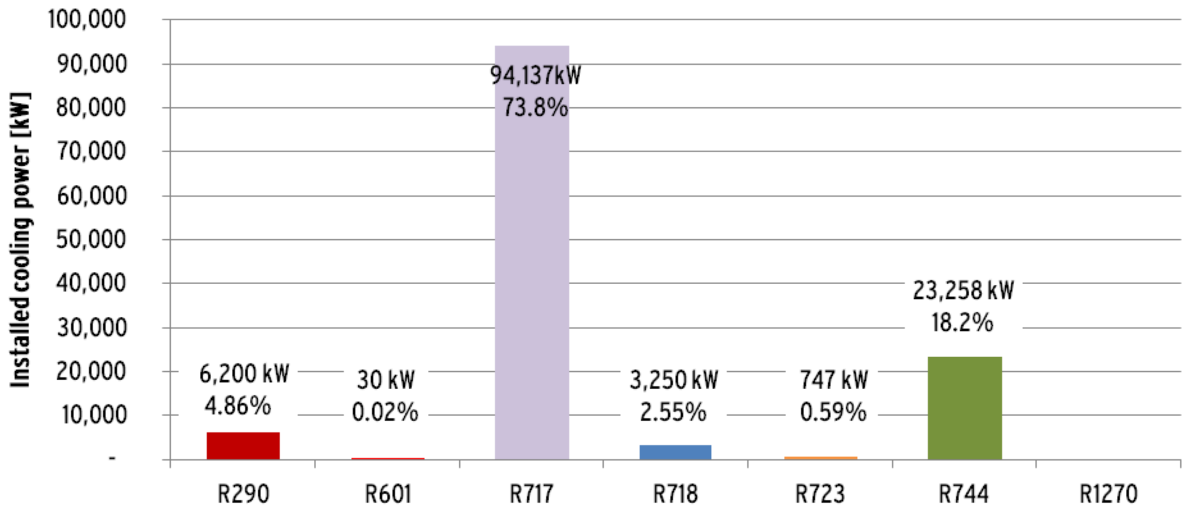


Figure 51: Share of the installed cooling power depending on the natural refrigerant (based on data of Table 11)

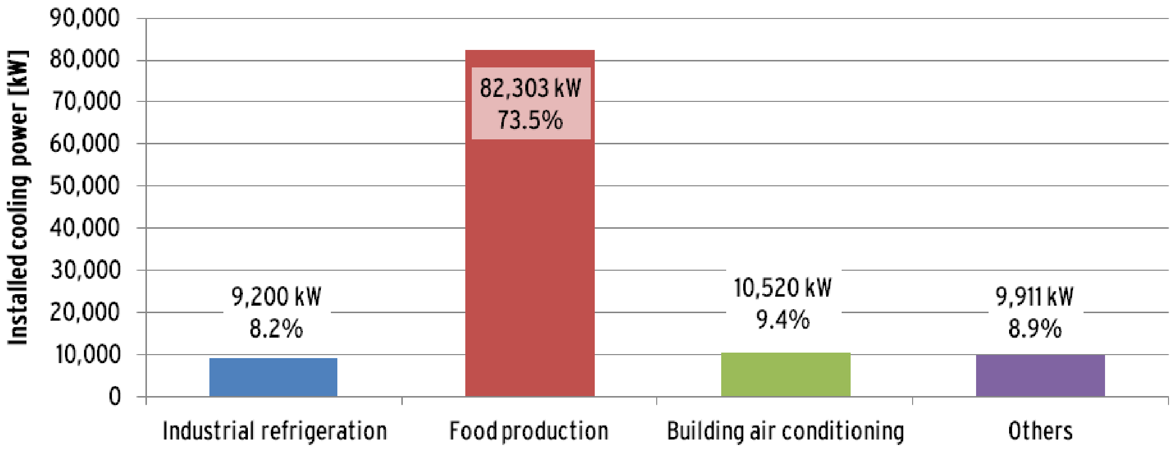


Figure 52: Share of the installed cooling power depending on the industrial sector (based on data of Table 11)

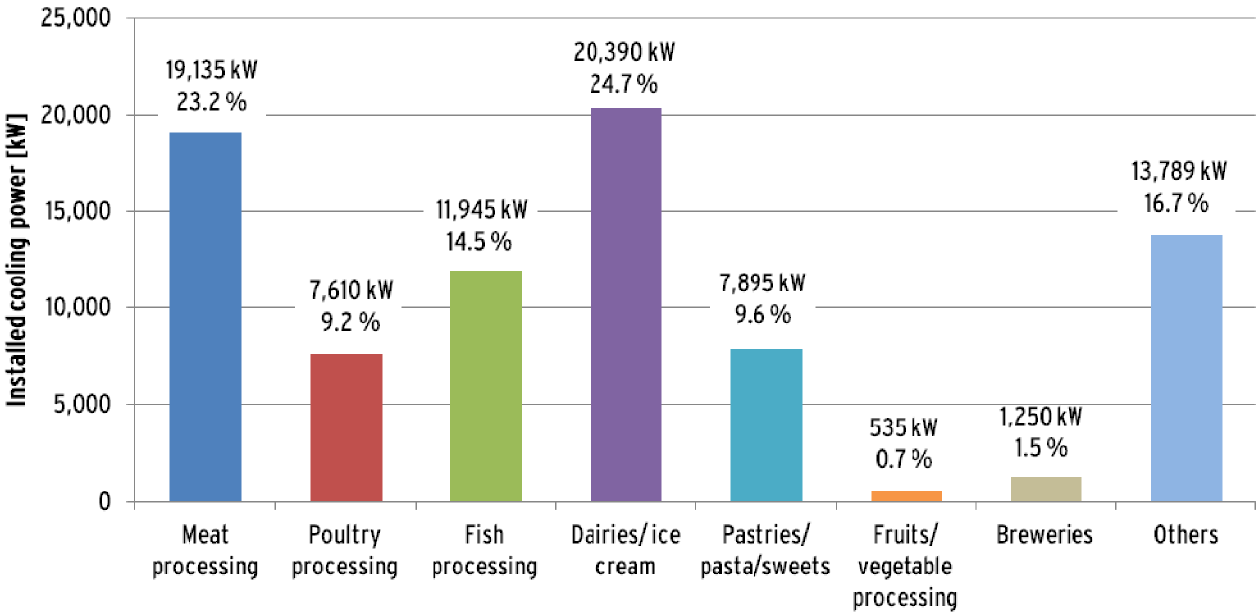


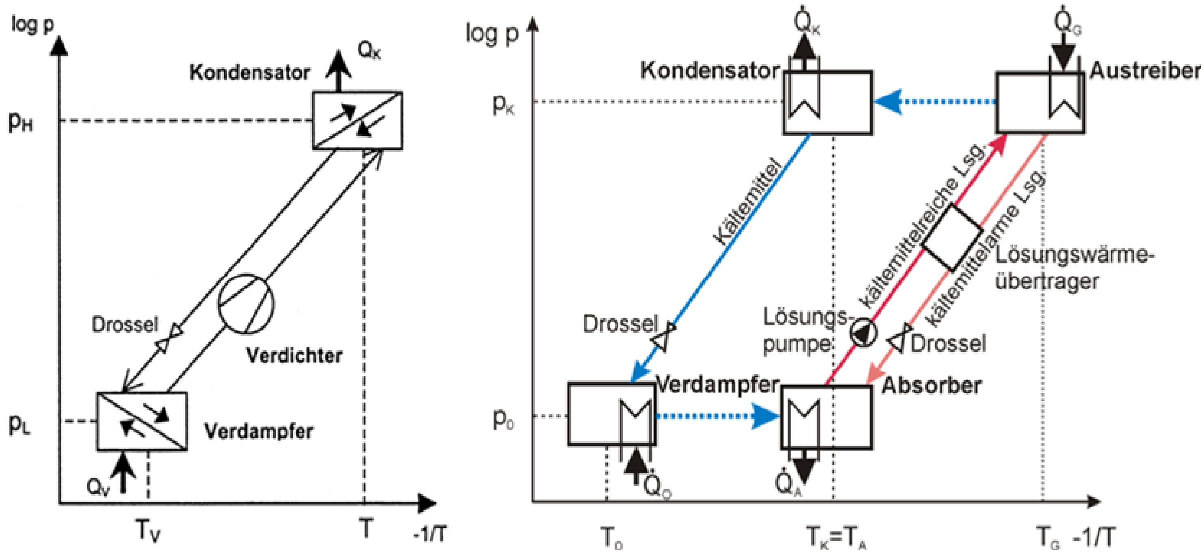
Figure 53: Share of the installed cooling power depending on the application within the food production sector (based on data of Table 11)

6.3 Thermally driven refrigeration systems (ad- and absorption type refrigeration plants)

6.3.1 Closed processes

Chilled water is produced in closed processes for various cooling and air conditioning applications. An absorption type refrigeration plant employs the absorbability of a liquid for a gaseous refrigerant leaving the evaporator. The increase of pressure until condensation pressure is carried out with a solvent pump pressurising the liquid phase in which the refrigerant is absorbed. At condensation pressure, the refrigerant is casted out of the sorbent by heat and afterwards liquefied.

Sorption processes are separated into continuous (absorption of the refrigerant vapour by a solvent) and discontinuous systems (adsorption of refrigerant vapour by a solid surface). Already in the 19th century, the absorption process had gained large-scale importance until developments of the mechanical engineering of mechanical ventilators dominated the refrigeration technology. Figure 54 provides a comparison of a vapour compression refrigeration process, (left-hand side), and the absorption type refrigeration process (right-hand side). The basic principle, which both processes have in common, is clearly visibly. Therefore, the sorptive processes are also called "thermal compressors."



Absorber: absorber, Austreiber: generator, Drossel: throttle, Kältemittel: refrigerant, kältemittelarme Lösung: solution with low refrigerant share, kältemittelreiche Lösung solution with large refrigerant share, Kondensator: condenser, Lösungspumpe: solution pump, Lösungswärmeübertrager: solution heat exchanger, Verdampfer: evaporator

Figure 54: Schematic log p-(1/T)-diagrams of a compression type refrigeration plant (left-hand side) and an absorption type refrigeration plant (right-hand side)

In the area of thermally driven refrigeration processes, to which adsorption refrigeration systems are also assigned, different types of plants with various refrigerant-absorbent combinations are available. Each of those combinations has their own characteristics.

6.3.1.1 Refrigerant-absorbent combinations

Adsorption type refrigeration plant

The refrigerants water, methanol, and ammonia as well as the sorbents activated carbon, zeolite, and silica gel are utilised for adsorption type refrigeration plants (Fan et al. 2007). However, chiefly water/zeolite and water/silica gel are offered on the market.

The utilisation of water as a refrigerant limits the chilled water exit temperatures to minimum 4 °C due to the lower limit of the evaporation temperature to be greater than 0 °C. The application of methanol or ammonia allows for temperatures below 0 °C. The danger of crystallisation, as it can occur in absorption type refrigeration plants with solvents, cannot be observed here. Due to the utilisation of water, it is necessary to carry out the process at vacuum conditions. This demands a large degree of tightness and large cross-sections for the vapour flow.

The refrigerant-absorbent combinations water/silica gel and water/zeolite require driving temperatures between 60 and 90 °C leading to heat ratios between 0.4 and 0.7 (Henning 2004).

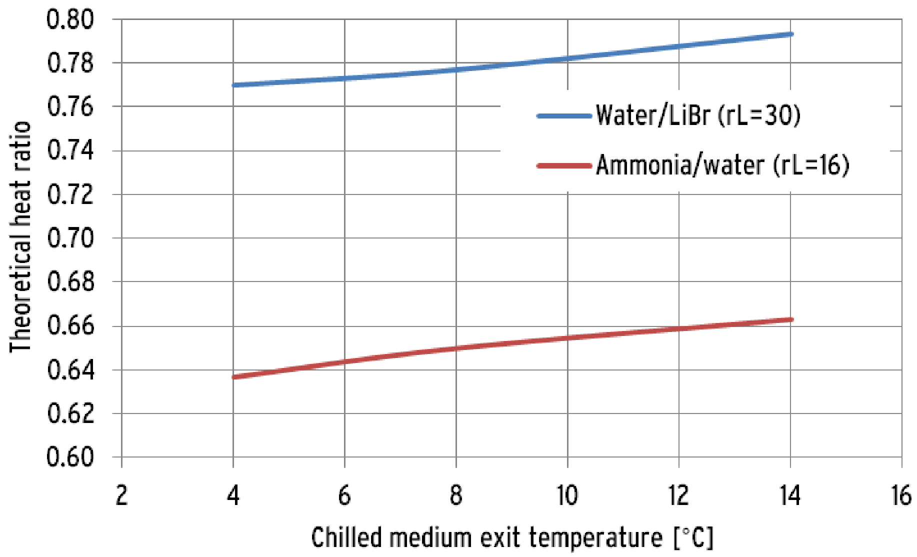
Absorption type refrigeration plant

The refrigerant-absorbent combinations water/lithium bromide and ammonia/water are commonly employed in absorption type refrigeration plants. Beside these particular combinations, water/lithium chloride has been employed for experimental plants.

For the application of water/lithium bromide, some technical limitations have to be considered due to the triple point of the refrigerant and the solubility of lithium bromide. The triple point of water limits the evaporator temperature to be greater than 0 °C, similar to adsorption type refrigeration processes. However, with a re-sorption process, in which a solvent circulates also in the evaporator, it is possible to obtain evaporation temperature below 0 °C (Richter 2008). The temperature spread among chilled water, cooling water, and heating water is limited by the solubility of lithium bromide. As soon as the limit of solubility is exceeded, the danger of crystallisation arises. The heat ratios of single-staged processes are approx. between 0.6 and 0.8 and depend on the temperatures of the chilled, cooling, and heating water, as well as on the kind of solvent circulation (see Figure 55, blue line). Similar to the adsorption type refrigeration plants utilising water as a refrigerant, vacuum conditions demand high tightness and large flow cross-sections.

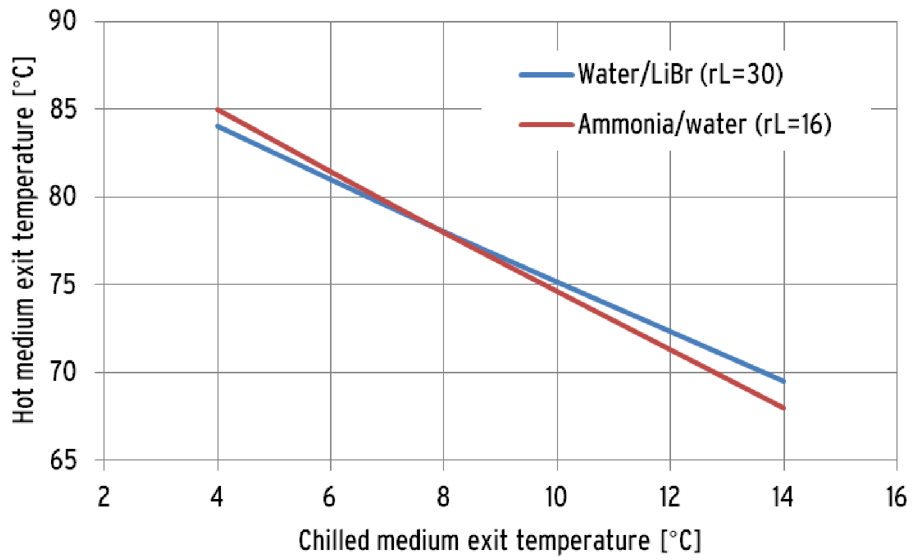
By employing the refrigerant-absorbent combination ammonia/water, evaporation temperatures below 0 °C are obtainable since ammonia is the refrigerant. Due to this, there is also no danger of crystallisation and due to the larger saturation pressures, smaller systems can be built compared to water/lithium bromide. However, the separation of water and ammonia has to be carried out with a rectification process, to keep the water out of the condenser and evaporator. The heat ratios are a little bit smaller than those of water/lithium bromide are and are observed to be between 0.5 and 0.7 (see Figure 55, red line).

The application of so-called *ionic liquids* is relatively new. These liquids contain large and complex ions. Due to a large number of possible combinations, the variety of ionic liquids is accordingly very large (Merck 2005). First tests in an absorption type refrigeration plant with water as a refrigerant and 1-ethyl-3-methylimidazolium-ethylsulfat as the absorbent yielded promising results, which are, however, behind those of water/lithium bromide. The reason for this is the demand for adaption due to the partially very different material properties, e.g., the much larger viscosity and surface tension (Radspieler & Christian Schweigler 2010).



Boundary conditions: cooling water entry temperature: 29 °C, relative solvent circulation (r_L) = 30 for water/lithium bromide and 16 for ammonia/water, common temperature difference to external media, and degree of absorption efficiency: 0.4

Figure 55: Obtainable heat ratios depending on cooling medium exit temperature and refrigerant-absorbent combination



Boundary conditions: cooling water entry temperature: 29 °C, relative solvent circulation (r_L) = 30 for water/lithium bromide and 16 for ammonia/water, common temperature difference to external media, and degree of absorption efficiency: 0.4

Figure 56: Correlation of heating and cooling media temperatures

6.3.1.2 Set-up variants of absorption type refrigeration plants

Beside a *single-effect-single-lift* absorption type refrigeration process, as visualised in Figure 54 (p. 75), multiple-stage processes are possible (Figure 57). These lead to an increased temperature difference or heat ratio.

The so-called *effect* (*single*, *double*, and *triple*) determines the number of temperature levels at which the heating energy is provided to the process. Multiple stages lead to a better degree of heat utilisation and increased heat ratio (Figure 58). The so-called *lift* is a measure for the temperature difference to be obtained. The number of *stages* as been sometimes employed for compression type refrigeration systems is an umbrella term for both *effect* and *lift*. The actual design of the absorption type refrigeration process cannot be deduced from the number of *stages*.

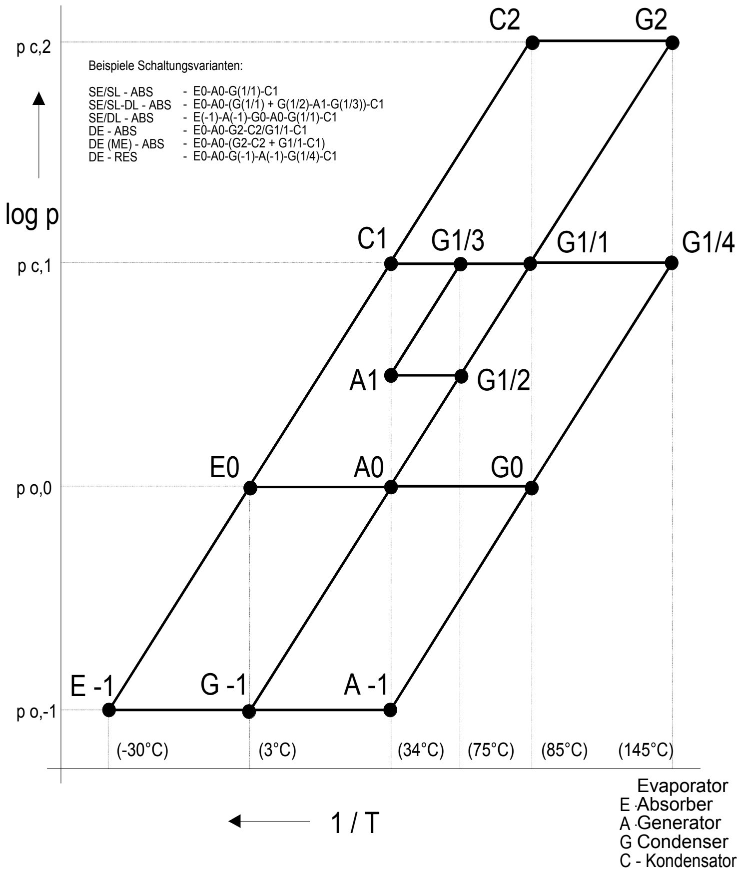


Figure 57: Set-up variants of different thermodynamic cycles of sorption type refrigeration plants

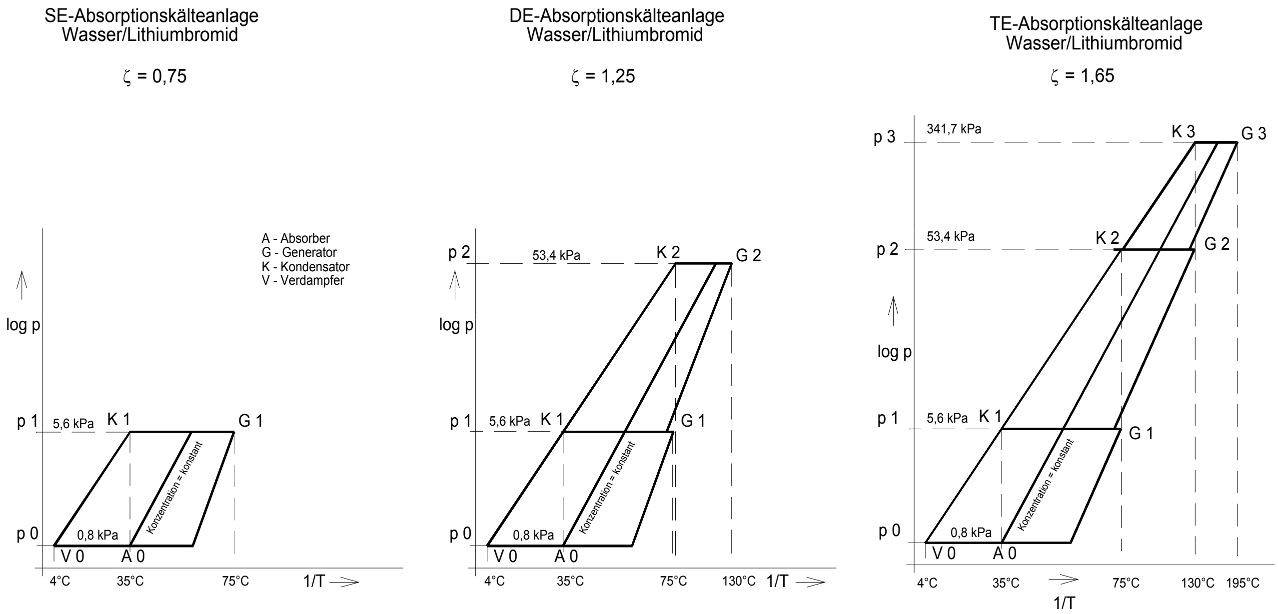


Figure 58: Visualisation of absorption type cycle in log p-(-1/T)-diagram depending on the *effect*

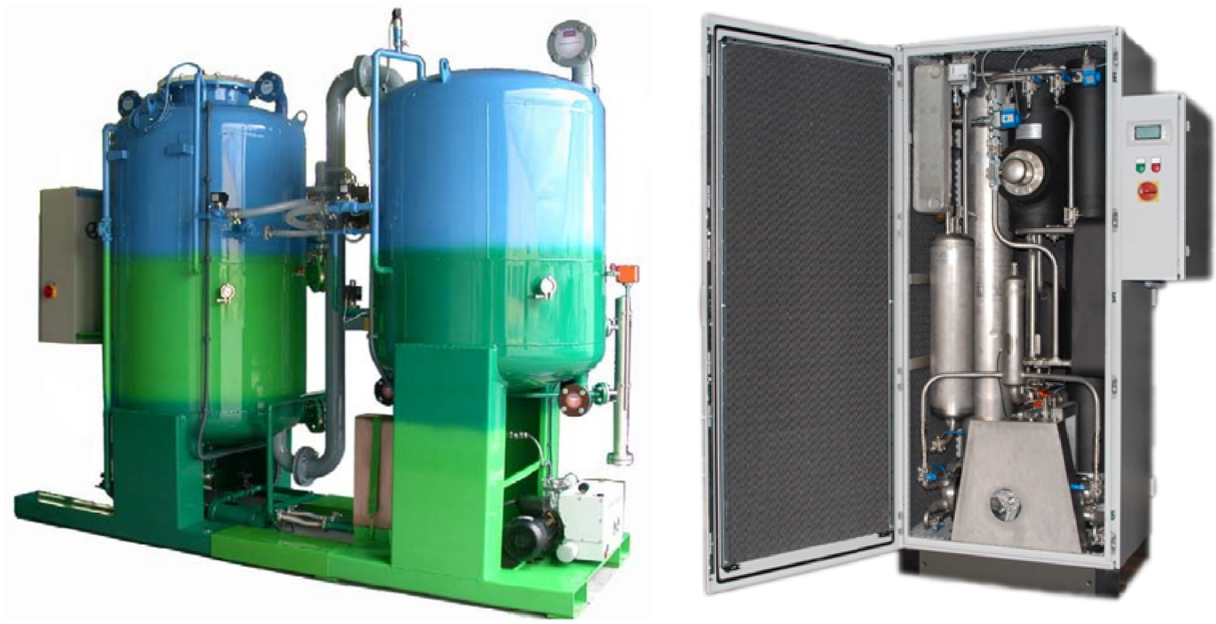


Figure 59: Examples of absorption type refrigeration systems; left-hand side: single-effect water/lithium bromide ACh by EAW Westenfeld, cooling power: 80 kW (EAW 2010)
right-hand side: single-effect ammonia/water Ach PinkChiller PC19 by Pink GmbH, Austria

6.3.1.3 Market overview of ab- and adsorption type refrigeration plants

The following figures provide an overview of systems and their power ranges being available on the market. Figure 60 considers single-staged ab- and adsorption type of refrigeration plants. It is obvious that adsorption type plants are chiefly applied in a low power range up to 50 kW, whereas absorption type plants are employed between 15 kW and more than 20 MW cooling power. Most of the systems are of water/lithium bromide absorption type. Systems with ammonia/water are less frequently used but cover a similar power range.

Double-effect systems are available only with water/lithium bromide. Figure 61 illustrates the power range provided, which starts far above 100 kW. To the best knowledge of the present authors, *double-effect* absorption type refrigeration plants of small power are not available on the market.

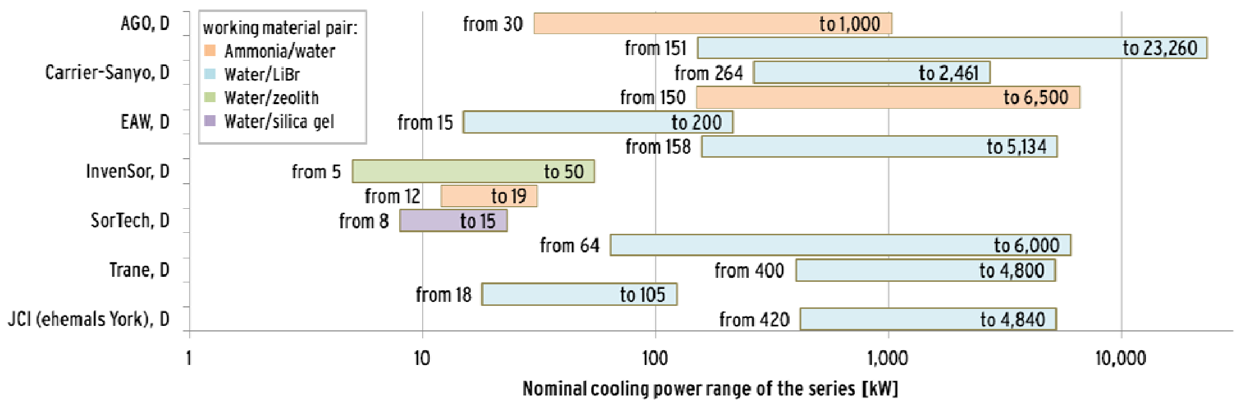


Figure 60: Cooling power range of existing single-staged ad- and absorption type refrigeration plants (without claim to be complete)

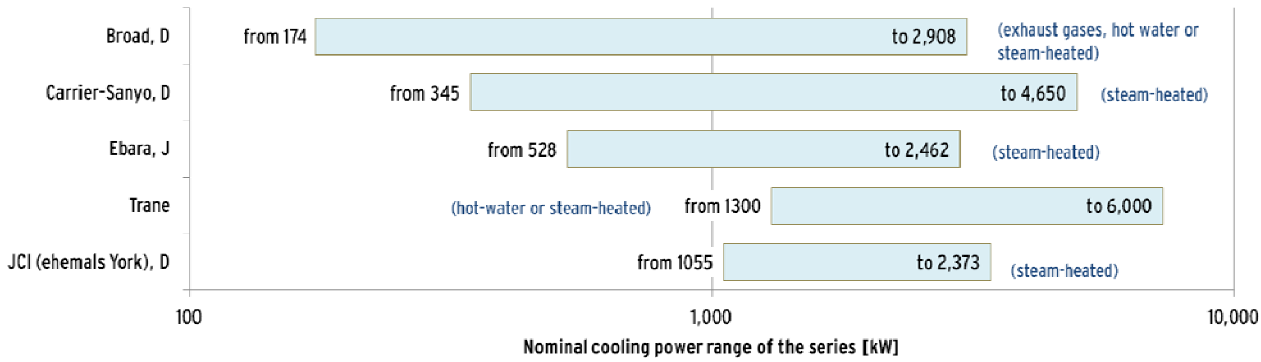


Figure 61: Cooling power range of existing double-effect absorption type refrigeration plants (without claim to be complete)

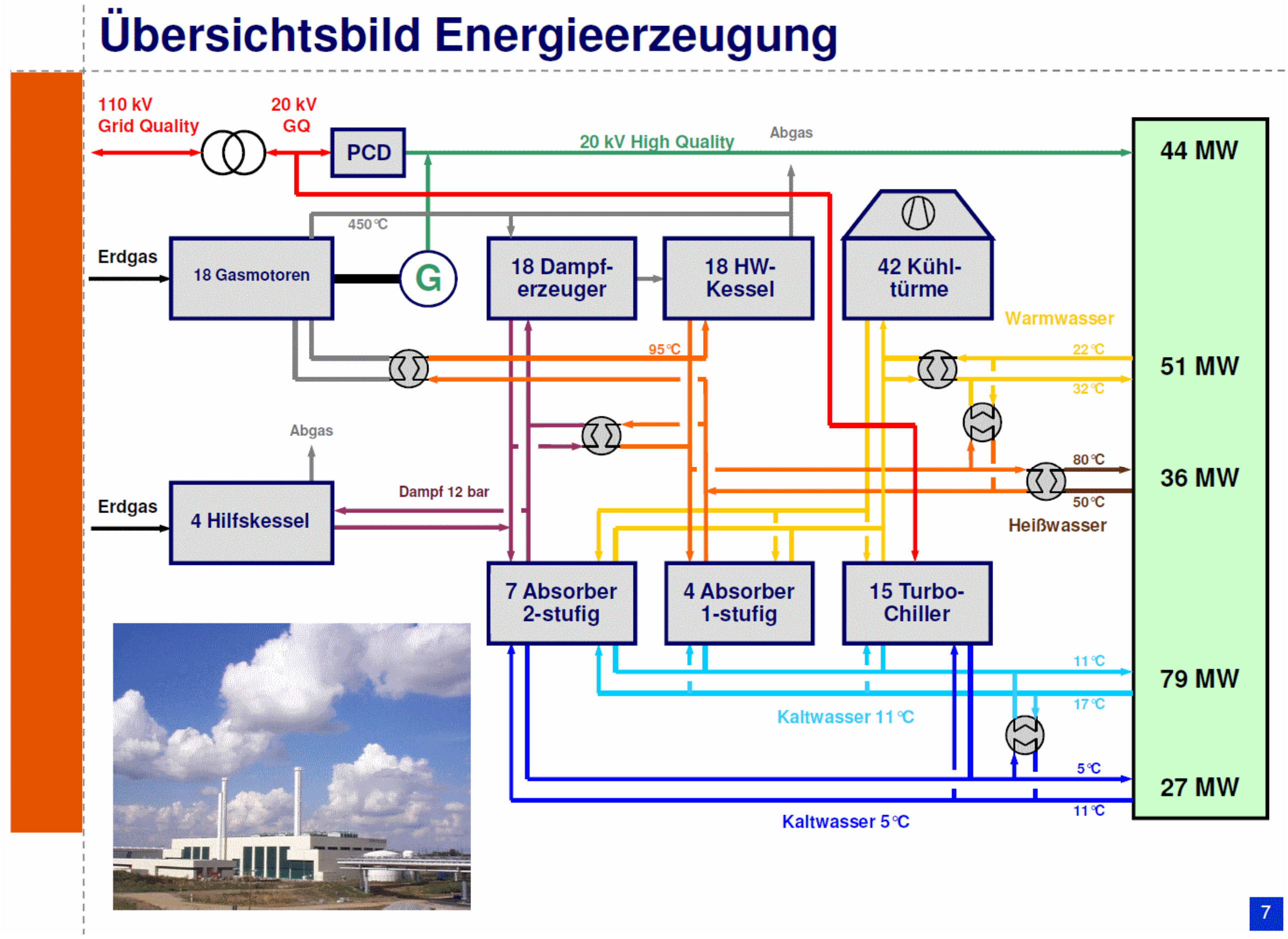
6.3.1.4 Applications of absorption type refrigeration plants in industry and building air conditioning

In the following paragraphs, two applications of absorption type refrigeration plants are described. Figure 62 illustrates a natural gas-driven combined heat, power, and cooling plant (CHPC) with a small cooling power, which supplies a non-residential building in Tettnang with electricity, heat for heating and hot water, and air conditioning cold. The heat-controlled CHP provides an electrical power of 30 kW and a thermal power of 65 kW. This implies that the CHP works only if heat is required. The heat is employed year-round for hot water generation, for heating in the winter, and for cooling in the summer. With 65 kW thermal power, the single-staged water/lithium bromide ACh produces a cooling power of 49 kW (heat ratio of 0.75). The more efficient alternative of a double-effect ACh is not economically efficient for this small power.

Figure 62: Combined heat, power, and cooling system (left-hand side picture, right-hand side machine) and absorption type refrigeration system (left-hand side picture, left-hand side machine) for air conditioning of a commercial building (right-hand side picture) (Weidner 2008)

A semiconductor manufacturer in Europe has an own supply for electricity, heat, and cooling, with strong requests on the security of the supply and constancy of the parameters (e.g., ±0.5 °C) whilst the power demand varies a lot. Figure 63 illustrates the interconnection of all media streams for a highly efficient and economic compound energy system. The electricity production is carried out with gas motors of large power (44 MW_{el}), which offer some larger efficiency due to large exhaust gas temperatures. The waste heat is utilised for steam generation, which drives a double-effect absorption type refrigeration plant that has a heat ratio of 1.3. Parallel to that, it is possible to employ the heat from the oil and gas motor in single-effect ACh for cooling generation or for heating purposes.

For covering peak cooling demands, there are additional turbo chillers installed, which are utilised only for few hours in a year. In sum, there are 106 MW of climate and process cold on two temperature levels (5 °C and 11 °C supply temperature) available. Natural gas-driven boilers are installed for covering thermal peak loads.



Herein: Abgas: exhaust gas, Erdgas: natural gas, Dampf: steam, Dampferzeuger: steam generator, Gasmotor: gas engine, Heißwasser: hot water, Hilfskessel : supplementary boiler, Kaltwasser: chilled water, Kühltürme: cooling towers, Warmwasser: warm water

Figure 63: Self-sufficient energy supply system with an extremely efficient CHPC plant of a large semi-conductor plant (Brinckmann 2008)

6.3.2 Open processes

The working fluid (water or steam) is in direct contact with the ambient air in open processes. These processes are chiefly employed in building air conditioning especially in ventilation systems for water desiccation and hence for the direct control of the fresh air, the so-called desiccative evaporative cooling (DEC).

The sorbents under consideration can be either solid or liquid (see Figure 64). Solid sorbents are usually placed on a rotating solid matrix, where the sorbent is moved continuously from fresh air (desiccation of air by adsorption of the vapour) to exhaust air (desorption of the water by warm exhaust air). The advantage of the solid sorbent process is the very easy set-up.

In contrast, liquid sorbents are trickled in the fresh air stream leading to desiccation of air and dilution of the sorbent. Desorption is carried out afterwards in a regenerator by heating. The advantage of this variation is the possibility to use a sorbent storage. In conjunction with solar thermal systems for regeneration, it is possible to cope with periods of low solar irradiation without interruption of the process and system.

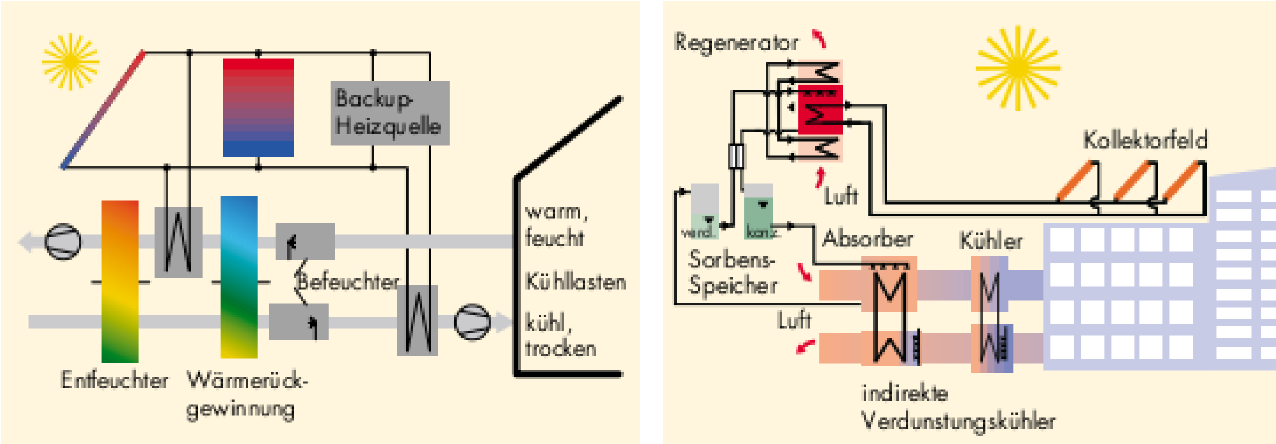


Figure 64: Examples of sorption-based building air conditioning: left-hand side: solid sorbent process; right-hand side: liquid sorbent process (Henning 2004)

Table 12: Processes of sorption-based building air conditioning (Henning et al. 2009)

	Process	
	Solid sorbent	**Liquid sorbent**
Material system (refrigerant/sorbent)	water/silica gel water/lithium chloride	water/calcium chloride water/lithium chloride
State of the art	Components available, design knowledge required	Several test plants
Nominal volume flow rate	typically > 5000 m³/h	> 1000 m³/h
Manufacturer	Many sorbent rotor manufacturers worldwide (Klingenburg, Munters, DehuTech, Seibu Giken, ...) Several unit manufacturers (Robatherm, Munters, Siegle & Epple, ...)	Menerga
Typical driving temperatures	50...95°C	50...70°C

7 Comparative ecological and economical assessment of selected systems

7.1 Introduction and approach

Different refrigeration systems for covering the cooling demand in non-residential building air conditioning and in industrial applications are compared within this chapter. By means of tangible examples, specific assets and drawbacks of various refrigeration systems are determined regarding their climate friendliness. The energy demand, total equivalent warming impact (TEWI), and the total annual costs are investigated and thermal comfort aspects are additionally considered for building air conditioning.

The evaluation basis are dynamical simulations of building and systems engineering, comprising an entire year in various climate regions and different scenarios. For the comparison of building air conditioning systems, an office building with a squarish floor space of 400 m^2 is assumed. The demand scenario corresponds to a typical office. For the comparison in industrial refrigeration, a system is assumed providing 500 kW at cooling medium supply temperature of 2 °C year-round. Such a configuration is imaginable in the food production (e.g., milk cooling).

The profiles of temperature and humidity have been chosen for two climate regions with the following scenarios (Meteonorm 2009):

- Frankfurt a. Main, mean year (F_M)
- Frankfurt a. Main, extreme summer (F_E)
- Hamburg , mean year (H_M)

The simulations of the building and the system have been carried out with the program TRNSYS.

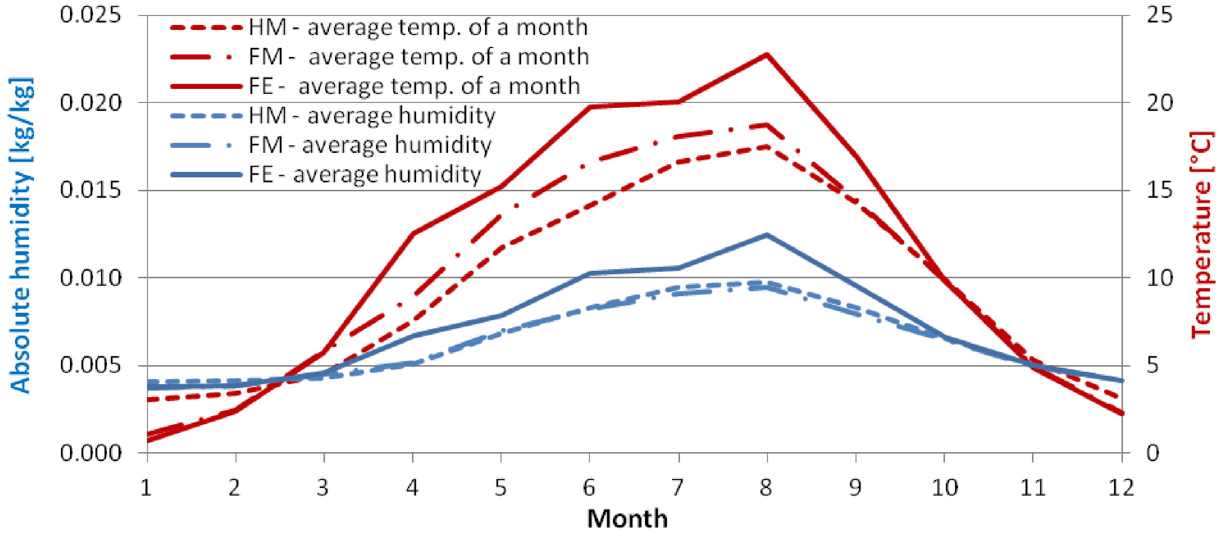


Figure 65: Absolute humidity and temperature depending on the time for two climate regions and different scenarios

7.2 Basis for calculation

7.2.1 Energy demand

As results of the simulation, final energy demands by means of electricity, heat from CHP, and ambient heat (e.g., solar energy) are calculated. Herewith, it is possible to determine the primary energy demand by means of primary energy factors (see Table 13).

Table 13: Primary energy factors (non-renewable share) for occurring final energy form (Robbi & Sander 2012)

Energy form and energy source	Primary energy factor f_p
Electrical energy mixture in Germany	2.4
Heat from fossil-fuel-fired CHP	0.7
Heat from solar thermal systems	0

7.2.2 TEWI

TEWI (Total Equivalent Warming Impact) is a number for estimating the contribution of a particular refrigeration system to the greenhouse effect. It comprises both the direct part by refrigerant emissions (leakages, improper disposal) and the indirect part by carbon dioxide and other gas emission, which occur during the conversion of energy for operating the refrigeration system during the lifetime. The CO_2 equivalent greenhouse gas emissions are determined in mass units of CO_2 (DIN-EN-378-1 2011).

According to DIN EN 378-1, the calculation is carried out by virtue of:

$$TEWI = \underbrace{GWP_{\mathrm{Ref}} \cdot L_{\mathrm{RS}} \cdot n_{\mathrm{RS}}}_{\text{direct share leakage}} + \underbrace{GWP_{\mathrm{Ref}} \cdot m_{\mathrm{Ref}} \cdot (1 - \alpha_{\mathrm{recovered}})}_{\text{direct share disposal}} + \underbrace{n_{\mathrm{RS}} \cdot E_{\mathrm{final,annual,RS}} \cdot \beta}_{\text{indirect share operating power}}$$

with:

L_{RS}	annual leakage rate of the refrigeration system [kg/a]
n_{RS}	operation time of the refrigeration system [a]
m_{Ref}	amount of refrigerant in the refrigeration system [kg]
$\alpha_{\mathrm{recovered}}$	share of recovered refrigerant when changed [%]
$E_{\mathrm{final,annual,RS}}$	final energy demand of the refrigeration system [kWh/a]
β	specific CO_2 emissions based on final energy demand [kg CO_2/kWh]

7.2.2.1 Direct TEWI share – refrigerant leakage and disposal losses

Information about refrigerant losses through leakages differs strongly in the literature. Usually, the information is gathered by one of the following methods:

- Allowed refrigerant losses by law
- Estimation of the losses by means of the yearly refrigerant production (including import and export) and comparison with the demand of newly installed systems

- Field test at already existing systems with systematic search for leakages and determination of leakage rates

Quantitative statements with relevant references can be found in appendix 11.1. Table 14 lists some leakage data, which is the basis for emission reports due to the Kyoto protocol (Strogies & Gniffke 2013).

As one of the few studies investigating systems in the field, (Birndt 1999) observed that most of the leakage with a significant refrigerant loss occurs at detachable junctions. This observation can be utilised for the interpretation of the data in Table 14. Usually, all junctions in mobile room air conditioning systems are soldered by the manufacturer, which is the reason why only small leakage losses occur. However, in multi-split and VRF systems usually screw joints are employed in the indoor units, which exhibit a significantly larger leakage rate.

Table 14: Leakage rate and disposal losses based on the refrigerant amount of various system (data for official reports from 2012 on (Strogies & Gniffke 2013))

System type	Annual leakage rate	Disposal losses	Average life-time [a]
Mobile room air conditioning systems	2.5 %	70 %	10
Split systems	5.0 %	70 %	10
Multi-split systems	6.0 %	30 %	13
VRF systems	7.0 %	30 %	13
Turbo and centrifugal chiller	4.0 %	30 %	25
Chiller > 100 kW	4.0 %	30 %	15
Chiller < 100 kW	4.0 %	30 %	15

7.2.2.2 Direct TEWI share – GWP values

The global warming potential (GWP) is a comparison value in CO_2 equivalents for estimating the greenhouse effect of a particular substance released to the atmosphere. Therefore, it considers the greenhouse effect and also the resting time of the molecules in the atmosphere. The data provided in Table 15 has been taken from the fourth Assessment Report of the IPCC (Forster & Ramaswamy 2007). They are effective bindingly for the reports of the greenhouse gas according the Kyoto protocol from 2013 on.

Table 15: GWP values for various refrigerants being utilised for the comparison (Forster & Ramaswamy 2007)

Refrigerant	Type	Name/Composition	GWP value
R134a	HFC	1,1,1,2-Tetrafluorethan	1,430
R410A	HFC mixture	Mixture of R32 and R125	2,087.5
R1234yf	HFC	2,3,3,3,Tetrafluorprop-1-en	4
R290	Hydrocarbon	Propane	3.3
R717	Natural refrigerant	Ammonia	0
R718	Natural refrigerant	Water	0
R723	Natural refrigerant	Mixture of dimetyl ether and ammonia	8
R744	Natural refrigerant	CO_2	1

7.2.2.3 Indirect TEWI share

The indirect TEWI share comprises the CO_2 emissions for operating the refrigeration system, considering all energy fluxes being required for the operation. Beside the major components such as compressors in compression type refrigeration plants, the auxiliary energy demand of pumps, ventilators, re-coolers, etc. is considered.

Three kinds of operation energy are considered: electrical energy, heat from a CHP plant, and ambient energy by solar heat. The specific CO_2 emissions of different operation energies are presented in Table 4, p. 37, and repeated here for convenience:

- Electrical energy: 583 g/kWh
- Heat from a CHP depending on the assessment method: 0, 44, and 239 g/kWh
- Heat from the ambient: 0 g/kWh

7.2.3 Costs of refrigeration systems

Beside the energetic and ecological properties of a particular refrigeration system, the costs are also of importance. The total costs of a refrigeration system comprise four components:

- Investment costs (acquisition, assembly, beginning of operation, etc.)
- operation costs (covering energy and water demand, etc.)
- Maintenance costs (service, inspection, repairing)
- Disposal costs

The investment costs are influenced by different factors. The costs for the acquisition of the system depend on the manufacturer and distributor. Usually, refrigeration systems are offered by the manufacturer including delivery, assembly, commissioning, and warranty. Hence, the price depends strongly upon the situation on-site (e.g., contentation situation, structure, and size of the building, installation location of the refrigeration system, etc.). The literature provides cost functions for refrigeration plants, mirroring common specific investment costs of plants or components depending on the power. One example is the IUTA price compendium (IUTA 2002) (see Figure 66). The functions rest upon various data and statistics about the costs, yielding a large variety. However, costs for special system requirements and configurations of ammonia systems (e.g., safety precautions) are usually not included. A detailed determination of investment costs requires always an individual consideration of the particular plant design.

Bases for the calculation in the present study are IUTA-price-compendium-based cost functions of (Wiemken, Elias & Nienborg 2012), calculations for existing plants which have been made available to the ILK Dresden, and individual tenders. Rising prices since the moment of price determination or issuing the invoice are considered with VCI price index for devices and systems (VCI 2012). The investment costs are converted into yearly costs for the entire lifetime by means of the method of annuities in accordance with VDI 2067. The market rate of interest has been assumed to 6 %. Approximate values for the lifetime of refrigeration plants are provided in DIN EN 13779:2004, VDI 2067, and DIN EN 15459. The prices and references, which are the basis for the cost comparison, are summarised in Table 50 to Table 52 in the appendix.

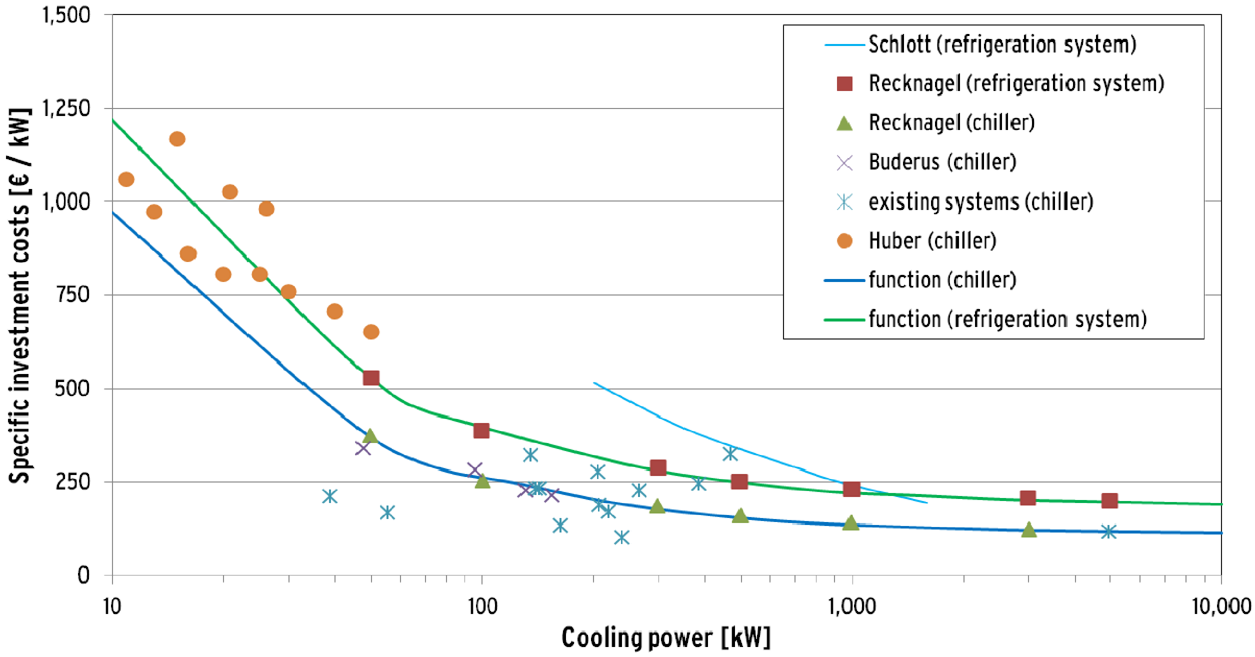


Figure 66: Examples for individual costs and cost function of compression type refrigerant plants (IUTA 2002)

The operation costs are dominated by the costs for providing energy (electricity, heat). There are two major parameters influencing the energy costs: firstly, the energy demands of the refrigeration plant, and, secondly, the costs per energy unit. The energy demand of a particular system can be calculated. However, the energy costs depend on the conditions of purchase. Energy-intensive industrial companies have to pay 0.05 EUR/kWh, whereas private households have to pay 0.26 EUR/kWh (Erneuerbare Energien 2012). The future development of energy prices is discussed controversially. The BMWI energy estimate (BMWI 2010) expects almost constant prices for electricity for 2012 until 2030. In contrast to that, (Kafsack 2011; E.-K. EC 2011) forecast in a study financed by the European Parliament an increase by approx. 50 % until 2030. Similar results have been published in a study for the Federal Ministry for the Nature, Climate, and Energy Industry of Baden-Württemberg (Energie GmbH 2012). Depending on the model, the authors forecasted and increase of 3 to 34 % for industry and 3 to 24 % for commercial companies until 2020. These numbers yield a vast field of possible scenarios for electricity price development. Reasons for this uncertainty are unforeseeable factors, such as fuel cost development or the share of electricity from renewable sources and its costs.

Different scenarios are considered in the present study. For scenarios with energy-intensive companies, a relatively low price of 0.05 EUR/kWh is assumed, whereas for smaller companies a price of 0.20 EUR/kWh is expected. An annual rise of costs of 2.28 % has been assumed, which corresponds to a total rise of 50 % until 2030. Therefore, average electricity prises of 0.06 EUR/kWh and 0.25 EUR/kWh are employed for the years 2012 and 2030. The scenarios are visualised in Table 16.

There are also large price differences for district heat, which, however, do not originate in the energy demand, but in regional circumstances. The prises for private households in German cities have been evaluated in 2009. Two of the extreme cases are 68 EUR/MWh (Chemnitz) and 138 EUR/MWh (Schwerin) (BBU 2009). The average prices vary also among the Federal States. An investigation of the “Arbeitsgemeinschaft für Wärme und Heizkraftwirtschaft” (AGFW, working group for heat and heating power economy) in 2010 revealed 58 EUR/MWh (Rhineland-Palatinate, power demand 600 kW) to 83 EUR/MWh (Thuringia, power demand 15 kW) (Wärme 2010). The German Energy Agency expects a rise of fuel prices of approx. 20 % until 2030 (Höflich et al. 2012), which corresponds to an annual rise of 1.02 %. These numbers can be applied also to district heat.

Different prices for heat are assumed for the scenarios of the operation cost calculation (see Table 16). For scenarios with heat free of charge (scenarios 1 and 3), a heat price of 0 EUR/MWh is assigned. In any other case, a Germany-wide average value of 69 EUR/MWh (power demand 600 kW) is employed. Applying the annual rise of 1.02 %, the mean price is 77 EUR/MWh for the timespan from 2012 until 2030.

Table 16: Prices for electricity and heat for the operation cost calculation of various scenarios

	Energy-intensive company	Waste heat without charge	Energy carrier	price [EUR/kWh]	Maintenance cost (relative to investment cost)	Water costs [EUR/m^3]	Disposal costs [EUR/a]
Scenario 1	•	•	electricity heat	0.06 0.00	4 %	4.81	54

Scenario 2	●	○	**electricity**	**0.06**			
			heat	**0.08**			
Scenario 3	○	●	**electricity**	**0.25**			
			heat	**0.00**			
Scenario 4	○	○	**electricity**	**0.25**			
			heat	**0.08**			

Beside the energy costs, there are additional costs for water (e.g., for open re-cooling of ACh systems or adiabatic cooling of DEC systems) in some systems under consideration. Water prises rise by approx. 1 %/a in Germany (DeStatis 2010). The specific fresh water costs including chemical additives are assumed 3.50 EUR/m^3 and those of wastewater are assumed 2.00 EUR/m^3 (Asmus 2010). It is assumed that 40 % of the delivered water is elutriated, in order to avoid concentrating of solids (e.g., salts, minerals) (Asmus 2010). The results of the simulations provide the amount of water to be evaporated. However, this is only a part of the required fresh water. In addition, it is required that the amount of elutriated water has to be considered both as fresh and waste water within the cost calculation. With the elutriation share, the fresh water amounts 5/3 of the evaporated water. The costs for providing water and water disposal can be calculated to be 4.81 EUR/m^3 based on the fresh water amount and considering the above-mentioned prices. This particular price is the mean value for the timespan from 2012 until 2030, assuming a rise of the costs of 1 %/a.

The estimate of the maintenance costs of various systems is explained in the following. In maintenance contracts, the annual costs for service and inspection of compression type refrigeration plants are specified to be between 1.1 % and 2.6 % of the investment costs. These contracts cover usually the annual survey of function and safety and the cleaning measures of filters and condensers. In addition, the compulsory survey of the tightness of the refrigeration cycle in dependence of the refrigerant charge (in case of HCF refrigerants) is included, in accordance with the F-gas regulation. Design of the refrigeration systems (chiller or direct evaporation), power, and constitution of the cooling grid influence the costs significantly. Repair work is explicitly not covered by those contracts and lead to larger maintenance costs in case of dysfunctions. Furthermore, DIN EN 13779:2004, VDI 2067, and DIN EN 15459 provide guide values for maintenance costs of various components. For complete systems, costs of 4 % of the investment costs are mentioned in the standards, which appears to be plausible when compared to investigated maintenance costs. Hence, this value is employed in the following.

Disposal costs are not or only fragmentary considered in the standards (e.g., DIN EN 15459). Furthermore, there is no data available at all for the disposal of entire systems and values, which are generally valid, cannot be provided by the manufacturers, since costs have to be calculated case-specifically. There are similar criteria applicable as for the maintenance (realisation and power of the system, constitution of the cooling grid, kind of refrigerant). Moreover, the disassembling and disposal costs are calculated differently compared to a refurbishment of a refrigeration system. A relatively disadvantageous case is the disassembling of a direct evaporation refrigeration system with a HFC refrigerant. According to some refrigeration system manufacturer, the disposal of the refrigerant is approx. 2 % of the investment costs. Considering a lifetime of approx. 15 years yields an annuity of 1 to 2 ‰ of the investment costs. Comparing this number with the costs for maintenance, it is obvious that the disposal costs are one order of magnitude smaller than those for maintenance. Hence, it can be concluded, that disposal costs are of minor importance, but are considered in the following for completeness. However, the value for the direct evaporation system mentioned above is employed for all refrigeration systems.

7.3 Building air conditioning

7.3.1 Selected refrigeration systems

All systems under consideration are explained by means of schematic drawings and their essential parameters. Detailed information is provided in Chapter 11.2 from page 191 onwards.

7.3.1.1 Variant 1: ACh – System with solar driven absorption type refrigeration plant

Variant 1 is a thermally driven water/lithium bromide absorption type chiller (ACh). The cold distribution within the building is carried out with a chilled water circuit and the room cooling by chilled ceilings. The heat for operation is harvested from a solar collector field. The system has some redundancy in the heat supply. In times of low solar irradiation and concurrent cooling demand, the ACh may also be operated by district heat. The re-cooling is carried out with an open wet cooling tower. Figure 67 illustrates schematically the system with its major components. The nominal supply temperature for the chilled ceiling is 14 °C, whereby a control of the supply temperature is required in order to avoid condensation. Table 17 provides an overview of the most important system parameters.

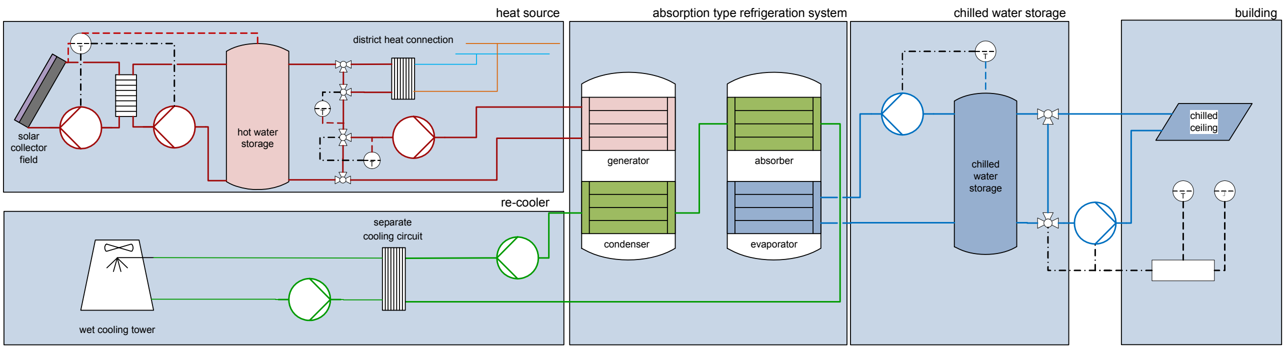


Figure 67: Schematic diagram of variant 1 – solar driven absorption type refrigeration system

Table 17: System parameters of a solar driven absorption type refrigeration system

Parameter	Symbol	Value	Unit
Nominal power of the absorption type refrigeration plant[7]	$\dot{Q}_{0,N,ACh}$	25	kW
Total solar collector area	$A_{o,Col,tot}$	90	m^2
Nominal power of the cooling tower[8]	$\dot{Q}_{RC,N}$	79	kW
Hot water storage	$V_{St,HW}$	1.5	m^3
Chilled water storage	$V_{St,ChW}$	2.0	m^3
Chilled ceiling area	$A_{o,ChC}$	400	m^2

[7] Nominal power at t_{HW} = 85/75 °C, t_{CoW} = 28/33 °C, t_{ChW} = 12/6 °C

[8] Catalogue value of Gohl-Dunsturm 6Z for t_{CoW} = 34/27 °C at t_{FK} = 21 °C (Gohl, 2009)

7.3.1.2 Variant 2: VRF – Multi-split VRF system

Split or multi-split systems belong to the category of compression type refrigeration systems. The heat or cold distribution within the building is carried out directly with the refrigerant in the refrigeration cycle. The cold delivery to the air is carried out by convectors. A multi-split VRF system is considered in the present study, whereby the word “multi” is assigned to a single outdoor device, which is connected to multiple indoor devices (evaporators). These indoor devices can be distributed in different rooms. The cooling power can be adjusted with a speed-controlled compressor and hence with a variable volume flow rate (variable refrigerant flow, VRF). The system under consideration is an air conditioning system without the capability for humidification. However, the uncontrollable dehumidification due to condensation at the evaporator is considered. Furthermore, the system does not comprise any ventilation, which is carried out by a separate system (see building model, subsection 7.3.2.1, p. 96). The composition is visualised schematically in Figure 68. It is assumed for the calculation that ideal air mixing occurs in the single-zone building model (open-plan office). This facilitates the assumption that several convectors in the room can be gathered to a single large one.

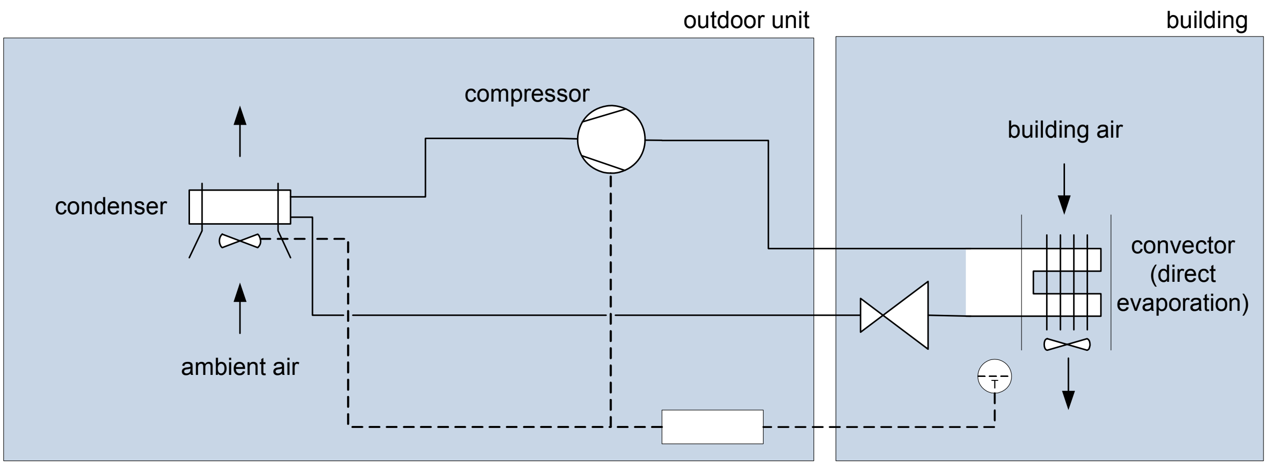


Figure 68: Schematic diagram of a split air conditioning system

Table 18: Major system parameters of the multi-split VRF system

Parameter	Symbol	Value	Unit
Nominal cooling power	$Q_{0,N,VRF}$	25	kW
Evaporation temperature	$t_{0,VRF}$	6	°C
Refrigerant	-	R410A	-
Refrigerant charge	M_{Ref}	20	kg

7.3.1.3 Variant 3: Ch – Chiller operated with a compression type refrigeration plant

Cooling supply with a chiller (Ch) is mainly carried out with a cooling generator in conjunction with a system for cooling delivery in the building. The principle composition of a chiller is similar to that of a direct evaporation split system. The major difference is that the cooling energy is not delivered to the customer directly by the refrigerant, but with a cooling medium (e.g., chilled water) circuit. As illustrated in Figure 70, the evaporator of the refrigerant circuit is located within the chilled water generator. Due to the fact that the cooling medium circuit is operated with liquid media only, different options for cold storages (chilled water storages) and cold delivery (coil fan, chilled ceilings, activated components) arise. Variant 3 consideres different concepts.

Variant 3a – Chiller with convectors for cold delivery

The cold delivery can be carried out by convectors similarly to direct evaporation split systems. As mentioned already for the multi-split VFR systems, all convectors in a room are combined into a single one in the simulation model.

This variant is extended furthermore by applying different refrigerants (R410A, R290, R134a, R717, and R1234yf) and considering the different material properties. The rest of the system is not varied in order to keep the number of variations finite. For instance, refrigerant-dependent constructions (e.g., an internal heat exchanger) can increase the system efficiency.

For compression type refrigeration plants utilising water as refrigerant (R718), modifications of the model are necessary due to the special physical properties of water. The evaporation pressure at 2 $^{\circ}$C is quite small with 7 mbar, whereas the pressure ratio between the saturation pressures of evaporation and condensation are large. Moreover, the small density of water vapour leads to large volume flow rates and demands for special compressors, which, however, have limited pressure ratios only. Therefore, two compressor stages in serial composition are employed (see Figure 70) and an intermediate cooling reduces the dicharge temperature.

A further noteworthiness when applying water is that both refrigerant and cooling medium are of the same fluid, which is the reason why the heat exchanger can be avoided. The refrigerant can be pumped directly into the chilled water circuit, leading to a very small temperature difference between refrigerant and cooling medium exit temperature. In practise, direct evaporators for water as refrigerant are employed for the generation of vacuum ice.

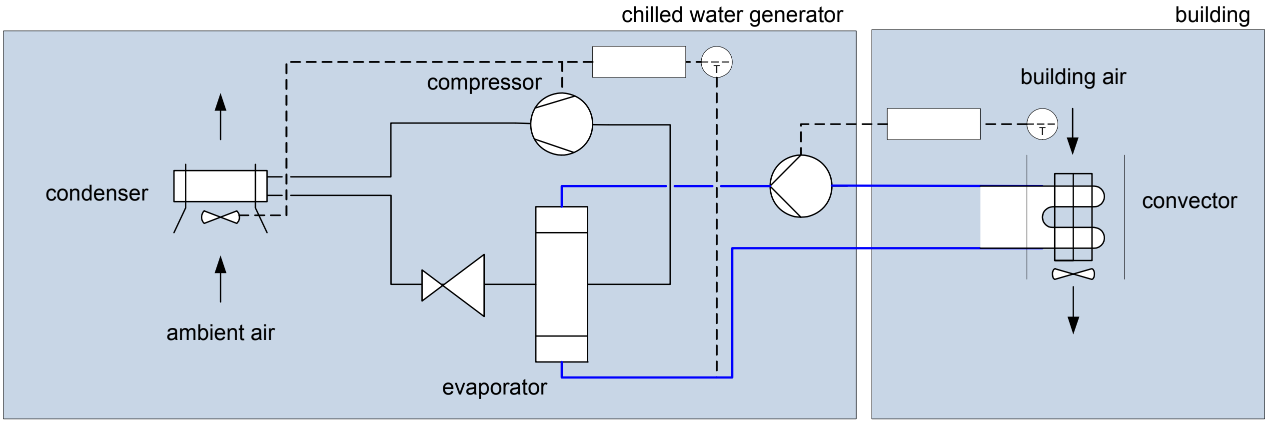


Figure 69: Schematic diagram of a chiller model with cold delivery through convectors

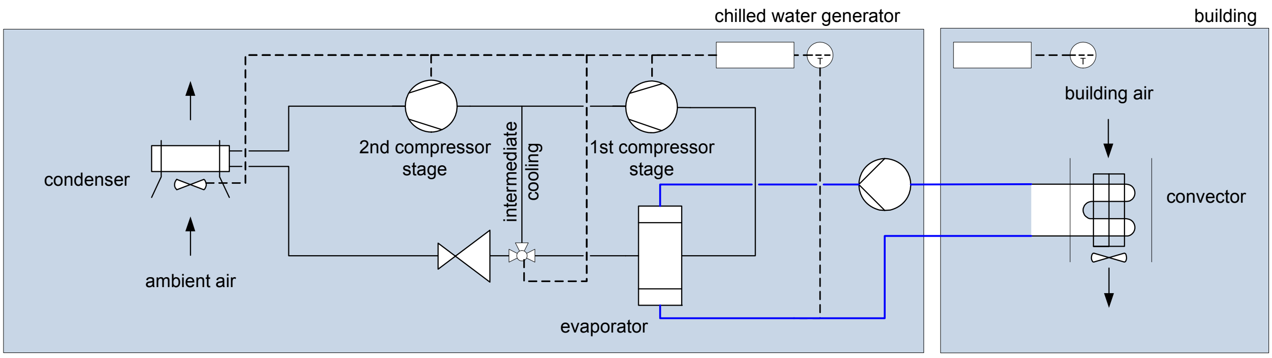


Figure 70: Schematic diagram of a refrigeration system with water as a refrigerant (R718)

Variant 3b – Chiller with chilled ceilings for cold delivery

An alternative to convectors are chilled ceilings, which allow for large chilled areas of relatively high temperature (nominal temperature of 14 °C in the model). Hence, the chiller can be operated at higher evaporation temperatures and coefficients of performance. Furthermore, convectors for distributing the cold can be avoided yielding a dramatic reduction of the electrical energy demand. Drawback of the system is the danger of condensate generation at the ceiling. The model checks the dew point within the room and adjusts the supply temperature of the chilled water in order to avoid condensation.

Variant 3c – Chiller with cold water storage

In this variant, the cooling generation, transport, and delivery are carried out as explained in variant 3a. However, with a cold water storage it is possible to store cold, which is generated at convenient conditions. For instance, the storage can be charged during the night when ambient temperatures are low and hence the efficiency higher, and discharged for cooling purposes during the day when ambient temperatures are high. For the cases of too high storage temperature or when the storage is not intended to be employed for cooling, the storage can be shunted with a bypass.

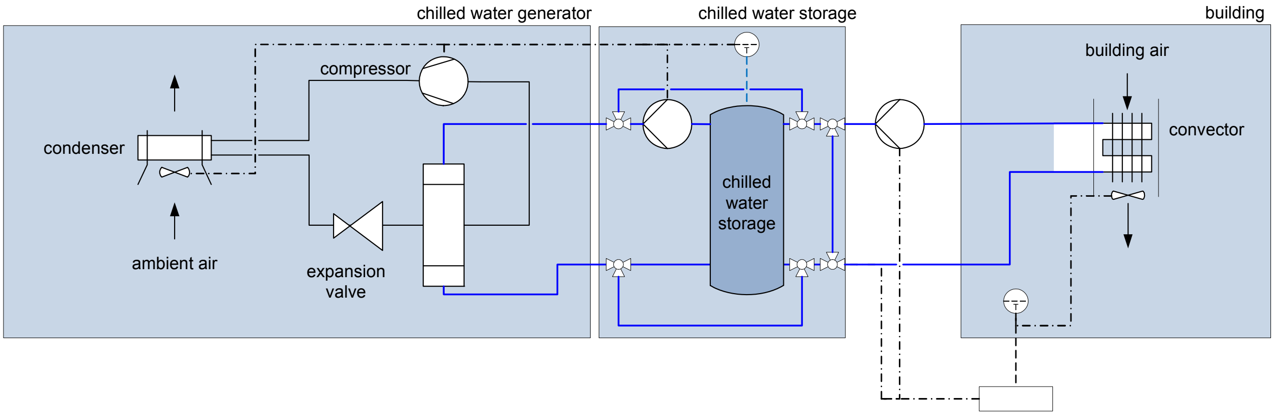


Figure 71: Schematic diagram of a chiller with a cold water storage

Table 19: Important parameters of a chiller system (only variant 3b is considered with various refrigerants)

Parameter	Symbol	Value	Unit
Nominal cooling power	$\dot{Q}_{0,N,Ch}$	25	kW
Supply temperature of cooling medium (convector/chilled ceiling)	$T_{sup,Ch}$	10/14	°C
Return temperature of cooling medium	$T_{ret,Ch}$	variable	°C
Storage volume	V_{St}	7.5	m^3
Charge temperature of the storage	$t_{st,min}$	6	°C
Refrigerants	-	R410A (R290, R134a, R717, R718, R1234yf, R32)	-
Refrigerant charge (R410A, R134a, R1234yf)	$m_{Ref,Ch}$	8	kg

7.3.1.4 Variant 4: HVAC – Humidification, ventilation, and air conditioning with DEC systems utilising CHP waste heat

This system differs significantly from the other systems under consideration. It is a humidification, ventilation, and air conditioning system (HVAC), which is designed in such a manner that the entire cooling demand can be covered by conditioning of the fresh air without an additional cooling medium. Just to remember: In other systems, the air within the building is cooled usually. The system works as follows: The fresh air is dehumidified and cooled in the absorber of a DEC system (desiccative evaporative cooling), which is based on a liquid sorption process. Due to the dehumidification, the temperature rises above the temperature of the cooled exhaust air, but remains below the untreated exhaust air. The cooling is carried out by evaporation of water. The water that has been absorbed by the sorbent is removed in the regenerator by virtue of CHP waste heat.

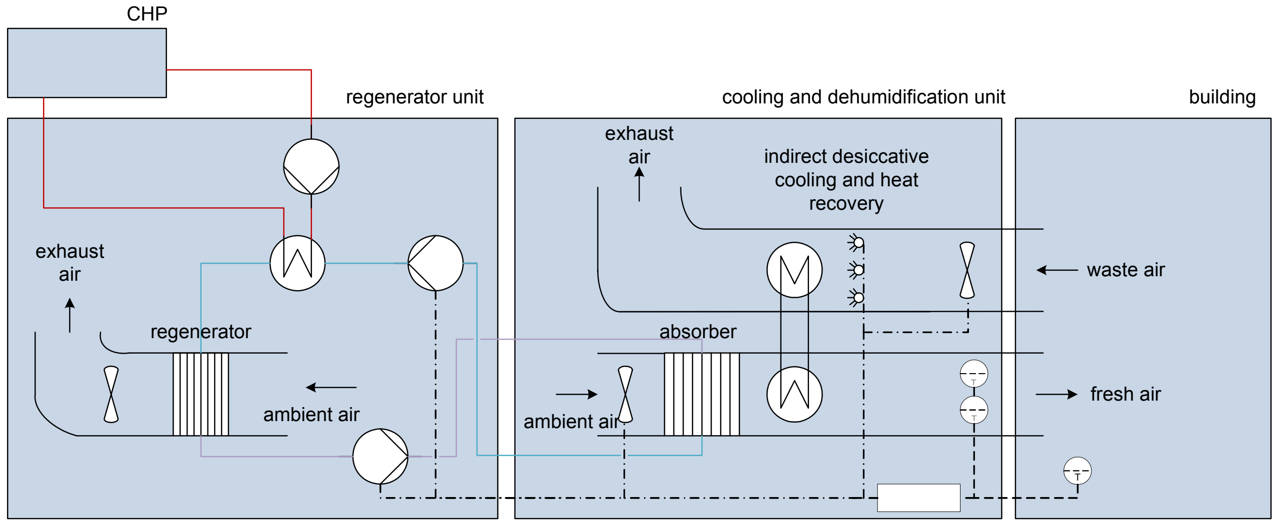


Figure 72: Schematic diagram of an HVAC system with sorption-based air conditioning

The degree of dehumidification of the fresh air can be adjusted to the demand in the absorber. For the case of ambient temperatures below the desired room temperature, free cooling is possible. However, if the ambient temperature is significantly lower than the room temperature, the heat exchanger within the systems allows an efficient heat recovery of the waste air.

Table 20: Important system parameters of the DEC system

Parameter	Symbol	Value	Unit
Nominal cooling power, adiabatic desiccative cooling[9]	$\dot{Q}_{0,ad,DEC}$	37.4	kW
Max. fresh/waste air volume flow rate	$\dot{V}_{fresh/waste,DEC}$	8,300	m³/h
Max. regeneration air flow rate	$\dot{V}_{Reg,DEC}$	2,800	m³/h
Supply/return temperatures of CHP hot water	$T_{sup/ret,CHP}$	70/60	°C

7.3.2 Modelling

7.3.2.1 Building model

An office building oriented towards the cardinal points and with a squarish floor area of 400 m² is assumed for the comparison of different systems for the building air conditioning. Only one representative floor of the building is considered, which is not separated. Hence, it can be considered a single-zone model and an itemisation depending on the cardinal points (due to different solar irradiation) is therefore not necessary. The windows are assumed to be distributed uniformly along the facade, yielding a well-balanced influence of the times of the

[9] Waste air conditions: 24°C, 45 % r.h., fresh air conditions: 32 °C, 40 % r.h., dehumidification of the fresh air to 11 g_{water}/kg_{air}, at maximum fresh and waste air volume flow rates (manufacturer data: (Menerga, 2010) and own calculations)

day and the year onto the indoor climate. Therefore, the objective comparison of various refrigeration systems is possible.

The office comprises 30 workplaces, which are occupied five days a week. The internal loads comprise the rejected heat of the persons (120 W/person), illumination (10 W/m^2), and data processing devices (140 W/workplace). These loads are adapted according to the occupation of the office (see Figure 73 and Table 21). The maximum specific load is 30 W/m^2, whereas the average for a working day (6 a.m. until 8 p.m.) is 19.6 W/m^2.

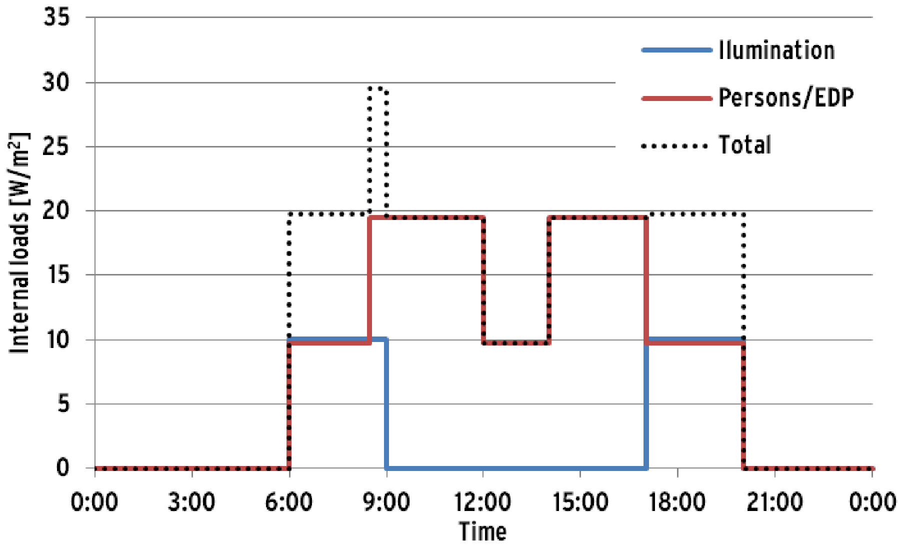


Figure 73: Profile of the internal loads during an office day (Mo.–Fr.)

Data processing devices contribute significantly to the internal thermal loads. It is assumed that each employee has a personal computer with a monitor having a total rejected heat flow of 140 W. Printers and fax devices have powers below 5 W and are therefore negligible. Data centres and servers are explicitly not considered here. Nonetheless, it can be reasonably assumed that load conditions are constant during a year. It is common that the cooling demand of servers is covered separately by an efficient system, since for some of the variants it is not secure to assume that the cooling demand can be covered anytime (e.g., solar cooling with ACh, DEC). It is nevertheless necessary to have a redundant refrigeration system for those systems. Moreover, it is not useful in any case to have a central refrigeration system for the relatively low load of a data centre in the transition period or the winter. Utilising such a system with a low part-load can result in a significant loss of efficiency (Wittig et al. 2012). Contrary, some heat shift, e.g. with a data centre or server room in a peripheral multi-split system, can be positive for the efficiency. All those aspects have to be evaluated for each application and each refrigeration system. In order to avoid some favouritism or discrimination of particular systems by considering data centres or server rooms, these are generally not considered in the calculations.

The building model is generated with “TRNBuild,” which is a graphical user interface for multi-zone building models in TRNSYS. Details of the model are listed in Table 21.

Table 21: Important parameters of the building model

Parameter	**Value(s)**
Geometry/walls	
Floor area	20 x 20 m
Ceiling height	2.8 m
Complexity	1 zone, 1 floor
Outer wall	U-value = 0.28 (EnEV 2009) composition: 10 cm concrete, 16 cm mineral wool, facade plate
Floor/ceiling	adiabatic composition: 17.5 cm concrete
Window area	Window area share 50 % U-value = 1.3 (EnEV 2009) g-value = 0.624 (EnEV 2009) Outer sun blinds: shadow 70% from 200 W/m^2 irradiance in particular window area
Internal loads	
Heat rejection	30 persons, 120 W/person (at 22 °C) (VDI-2078 2003), 30 data processing units, 140 W/data processing unit, 10W/m^2 illumination, → 30 W/m^2 maximum value, 19.6 W/m^2 working day average
Operation time of employees/data processing	100 %: Mo.–Fr.: 8:30 a.m.–noon and 2 p.m.–5 p.m. 50 %: Mo.–Fr.: 6 a.m.–8 p.m. 0 %: other times
Operating time of illumination	100 %: Mo.–Fr.: 6 a.m.–9 a.m. and 5 p.m.–8 p.m. 0 %: other times
Air renewal	
Infiltration	0.2 1/h: every time
Forced ventilation	0.4 1/h: Mo.–Fr.: 5 a.m.–8 p.m. (incl. minimal air renewal rate for an average air quality according to DIN EN 13779) exception: up to 5.4 1/h for DEC systems heat recovery at ambient temperatures below 15 °C (recovery ratio 0.8 for DEC systems (Menerga 2012) and 0.5 for other systems (DIN-V-18599 2007))
Cooling	
Nominal temperature (cooling)	24 °C, at ambient temperatures below 30 °C 6 K below ambient temperature at ambient temperatures above 30 °C
Operation time	Mo.–Fr.: 6 a.m.–8 p.m., when exceeding nominal temperature (cooling)
Heating	
Nominal temperature (heating)	20 °C: Mo.–Fr.: 5 a.m.–8 p.m. 18 °C: other times
Operation time	All-the-week, when undercutting nominal temperature (heating)

The physical properties, the utilisation profile, and the climatic influences determine the heat flows into the building. The values for a warm summer week in two climate scenarios (F_E, F_M) and the office floor under consideration are visualised in Figure 74. It can be observed that solar irradiation through the windows (direct and diffuse) and the internal loads are the most important heat sources. The heat input via ventilation, infiltration, and thermal conduction through the windows (without solar radiation) and walls are of minor importance. Assuming equal office temperatures in close-by floors, the ceiling and flooring act as heat storages (heat from the day is released during the night). Outer walls work in a similar manner. However, much more heat is rejected than gained through them. The reason for this is that average ambient temperatures (F_M: 20.0 °C; F_E: 23.5 °C), are below the maximum internal temperature (24 °C). The illustration indicates also that the windows contribute – beside the above-mentioned heat gain through solar irradiation - to the heat losses. Ventilation and infiltration yield large heat losses and small heat gain for climate F_M. The reason for this behaviour is that the average ambient temperatures are significantly lower than the maximum room temperature. Contrary, during an extreme summer (F_E), components, whose heat flows depend strongly on the ambient temperature, contribute stronger to the heat input than to the heat losses.

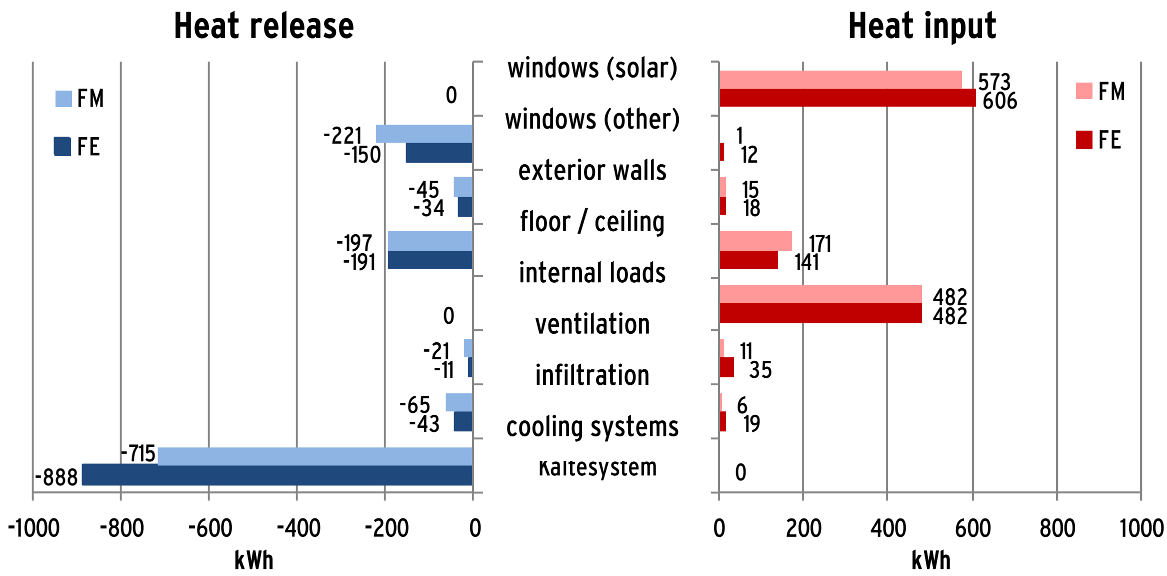


Figure 74: Itemisation of accumulated heat flows for the office floor during a warm summer week, F_M

From the heat flow contemplations, it is obvious that for the building under consideration and its utilisation profile, most of the heat rejection has to be carried out with a refrigeration system. In the subsequent investigations, it is worked out how the heat rejection can be conducted considering climate safety, costs, and thermal comfort aspects.

7.3.2.2 Refrigeration systems

The approach and assumptions for modelling the refrigeration system are summarised in Table 22. Further details are provided in the appendix.

Table 22: Approach and assumptions for modelling the refrigeration system

Model	Values/assumptions
Refrigeration circuit	
Basics	• Major components of refrigeration circuit (compressor, condenser, evaporator, expansion device) are considered for the model. • Points of state of the refrigeration circuit are calculated for the refrigerant by material property databases • Boundary conditions, such as temperatures and pressures, and heat flows are given by the application and assumptions (see below)
Compressor	• Total degree of efficiency: 0.7 • Electrical power is calculated from technical work in Carnot process and losses according to total degree of efficiency • Single-staged compression (except for R718, double-staged here)
Condenser	• Fixed minimal temperature difference between refrigerant and cooling air in counter-current heat exchanger • Sub-cooling of refrigerant: 5 K
Evaporator	• Fixed minimal temperature difference between refrigerant and cooling medium/air in heat exchanger
Expansion device	• Assumed to be lossless • No limitation of controllability (all pressure ratios available)
Absorption type refrigeration system	
Basics	• Absorption type refrigeration plant described with model of the characteristic equation (Heinrich 2004) • Coefficients are based on an investigation of small H_2O/LiBr ACh (Wiemken, Elias, Nienborg, et al. 2012)
Solar collectors	• Single-knot model • Coefficients are based on an investigation of a large number of solar collectors and represent state of the art
Change of state of the moist air	
Basics	• Changes of the state are calculated with separate program (utilising calculation methods behind a Mollier h-x diagram)
Ventilators and hydraulic pump	
Ventilators	• Electrical ventilator power in part-load (TL) with simplified relation between and volume flow rate (V_{Luft}) and electrical power (P_{el}) (Menerga 2012): $\left(\frac{\dot{V}_{air,\,part\text{-}load}}{\dot{V}_{air,\,nominal}}\right)^{1.78} = \left(\frac{P_{el,\,part\text{-}load}}{P_{el,\,nominal}}\right)$ Influences of volume flow rate on flow resistance and degree of efficiency roughly considered
Hydraulic pump	• pump degree of efficiency: 0.8, motor degree of efficiency: 0.75 (well-designed dry-running pump (Oraschewski et al. 2007)) • Electrical pump power calculated with the for the ventilator power defined equation
Chilled ceiling	
	• Limitation of the supply temperature in order to avoid condensation • Adaption of chilled water mass flow rate in order to adjust the power
Balance	

Model	Values/assumptions
Energy demand	• All electronic components, which are not explicitly excluded, are included in annual final energy demand • Electronic components (e.g., control electronics) and electrical components (e.g., electrical expansion valve) are not balanced, since they are system-independent and hence of minor importance

7.3.3 Results

7.3.3.1 Thermal comfort

The application of a refrigeration system for building air conditioning is required when temperature and humidity are large enough to be considered uncomfortable if such a system is missing. For the comparison of different systems, it is important to investigate the adherence of comfort conditions, beside the energetic efficiency and environmental safety.

In the next following diagrams, it is illustrated with which refrigeration system the acceptable room temperature is exceeded, for how long, and at which super-heat. It has to be mentioned that differences smaller than 1 K to the nominal value (24 °C) are not considered an exceeding, since in both simulation and practice such small differences cannot be avoided. For the humidity, values larger than 11.5 g water per kilogram air are considered an exceeding (according to DIN 1946-2 for "thermal comfort"). The situations for the particular systems are illustrated in Figure 75 (room temperature) and Figure 76 (humidity). The following information related to the temperature control shall be considered:

- VRF and chiller systems have short times of temperature exceeding only, which is moreover relatively low (reason: sufficient cooling power, low air renewal rates and hence low input of sensible and latent loads with the fresh air)
- DEC systems exhibit much more exceeding hours (reason: relatively large fresh air temperature not sufficient for covering cooling demand (especially for F_E) → air volume flow rate should be larger (especially for F_E) → even larger energy demand)
- Purely solar driven absorption type refrigeration systems (without backup for heat supply and without large thermal storage) have often and large temperature exceedings. An increase of the solar collector area, and their distribution to different cardinal points (e.g., to harvest more energy in the afternoon) have only a limited effect.
- Utilisation of chilled ceilings yields exceeding room temperatures independent of the refrigeration system (reason: minimal temperature above the dew point limits the cooling power; lower temperature leads to condensation).

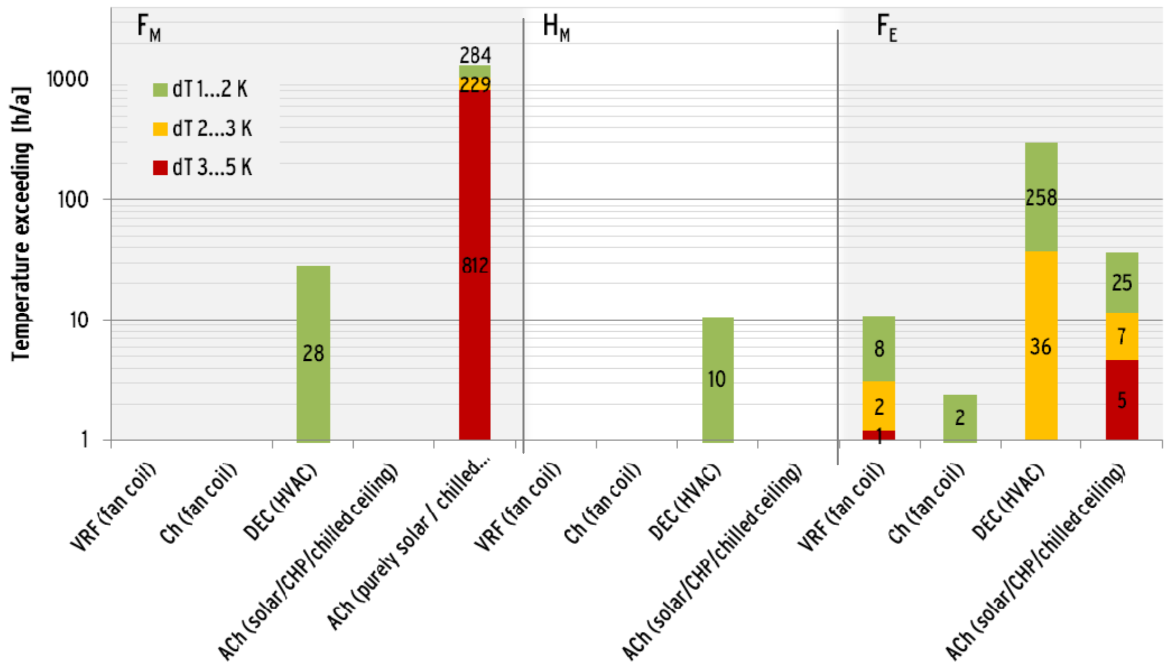


Figure 75: Frequency of exceeding of the nominal room temperature for various refrigeration systems and climates (CHP – combined heat and power, district heat supply)

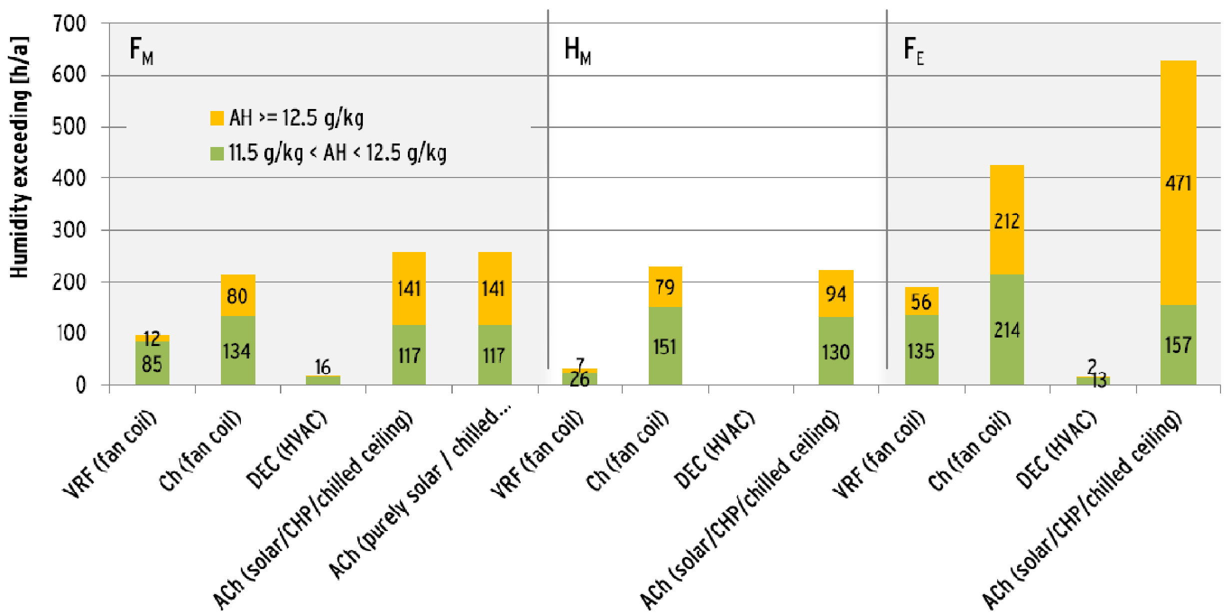


Figure 76: Frequency of exceeding of the maximum absolute humidity (AH) of 11.5 g/kg for various refrigeration systems and climates (CHP – combined heat and power, district heat supply)

The following information related to the humidity control shall be considered:

- VRF and chillers do not provide a controlled dehumidification. The dehumidification of VRF systems is larger than those of chillers due to lower convector temperatures (6 °C and ≥ 10 °C, respectively)
- A controlled dehumidification in HVAC systems with DEC leads to low exceeding frequencies.
- Systems with chilled ceilings cannot dehumidify the air (large supply temperatures in order to avoid condensation at the ceiling) → often exceeding of humidity limits.

The best results of all systems under consideration regarding thermal comfort can be obtained with VRF and chiller systems with air convectors. However, systems with chilled ceilings are not able to provide sufficient cooling power whilst avoiding condensation in warm and humid summers. Moreover, a humidity control is impossible with chilled ceilings. DEC systems are able to dehumidify the air sufficiently well. However, the cooling power is limited due to the limited fresh air temperature.

An optimal configuration might be a separation of cooling and dehumidification. Dehumidifying with a sorptive DEC process and cooling with chilled ceilings or concrete activation, enables thermal comfort with low air velocities.

7.3.3.2 Energy demand

Multi-split VRF systems with convector require the largest amount of electrical energy amongst the compression type refrigeration systems. On the one hand, this is due to the relatively low evaporation temperature and the resulting larger compressor power. Moreover, a low evaporator temperature leads to a larger dehumidification power (see Figure 79), which does not reduce the air temperature, but increases the energy demand. On the other hand, such systems show a strong decline of the COP at part-load conditions below 40 % of the nominal cooling power (DIN-V-18599 2007). For the cases studied, these systems are employed up to 85 % of the total operation time in this power range.

Figure 77 and Figure 78 illustrate the primary and final energy demand, respectively, of various refrigeration systems and climatic conditions. It can be observed that the extension of a chiller by a cold-water storage reduces the electrical energy demand by 10 % and 12 %, respectively. The values originate from calculations, which solely optimised the storage size. With an optimisation of the control strategy, shape, and thermal insulation of the storage, the reduction can be even larger. Moreover, the utilisation of a chilled ceiling for cold distribution leads also to a reduction of the energy demand (27 % for climate F_E). The reason for the reduction is the increased chilled water supply temperature (14 °C instead of 10 °C), and saved electrical energy by substitution of ventilators by natural convection. Furthermore, it has to be stressed that chilled ceilings are generally not allowed to dehumidify, which is the reason why no latent heat has to be rejected by the refrigeration system.

The HVAC system with DEC exhibits the biggest electrical energy demand. Due to the limited adiabatic cooling of the fresh air, it is required to have a large air renewal rate for covering the cooling demand. Beside the electrical energy demand, also thermal energy is required for the regeneration of the sorbent. Especially, in warm and humid climate (F_E) an extraordinary large thermal energy demand is required in order to reduce the exceeding hours regarding humidity (see Figure 79). However, a large degree of utilisation of waste heat (0.8) of the DEC system contributes significantly to the reduction of the heat energy demand (approx. 17 % in comparison to other systems, which have a degree of 0.5 of waste heat utilisation for forced air renewal (but not for infiltration)).

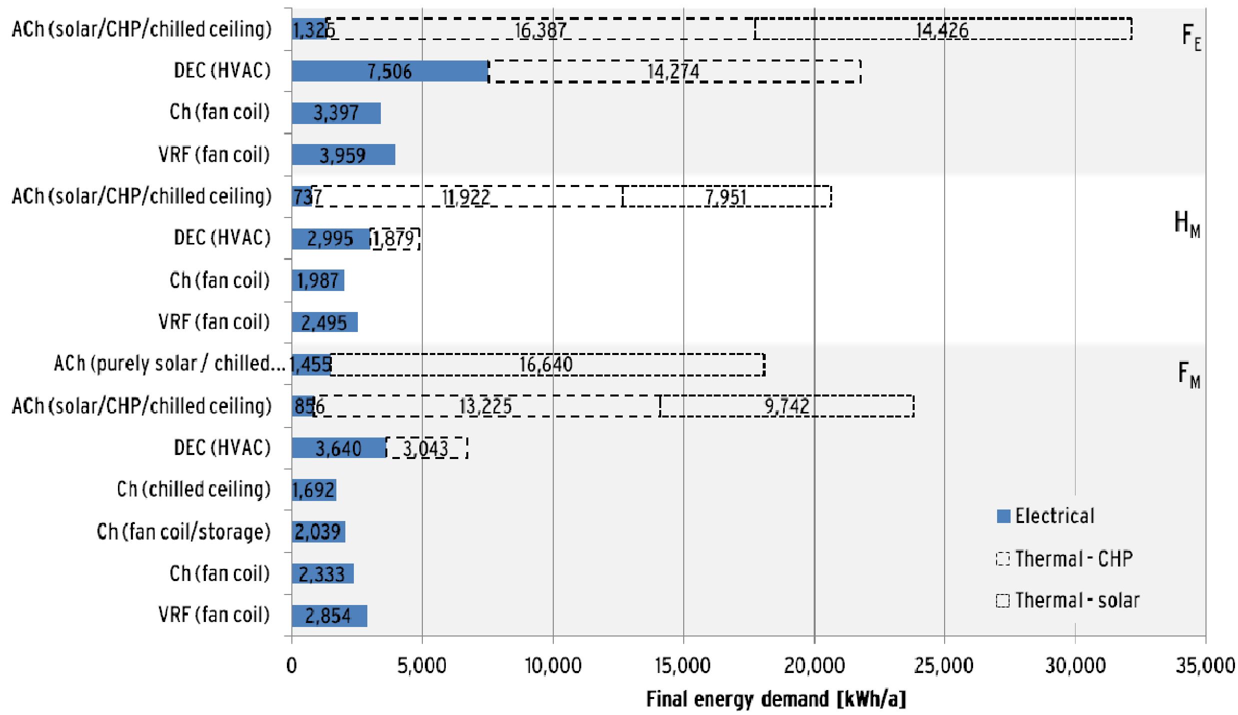


Figure 77: Final energy demand of refrigeration systems for building air conditioning (CHP combined heat and power/district heat supply)

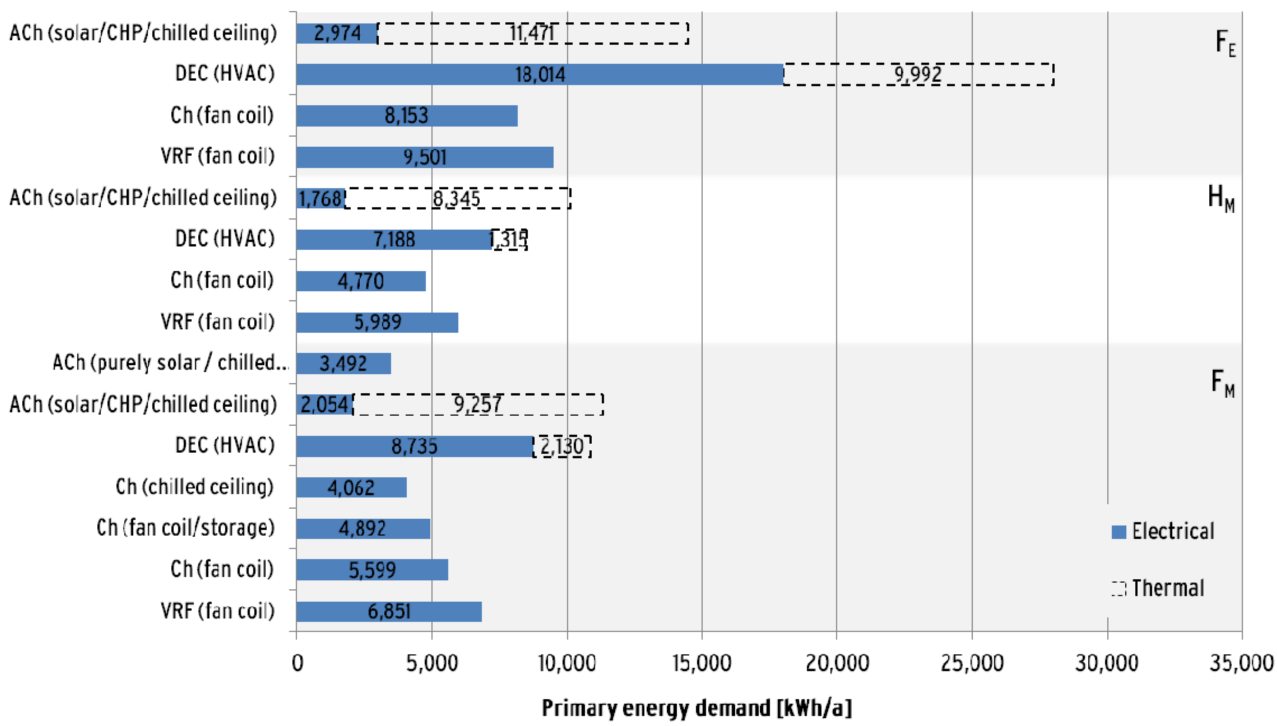


Figure 78: Primary energy demand – compilation (CHP combined heat and power/district heat supply)

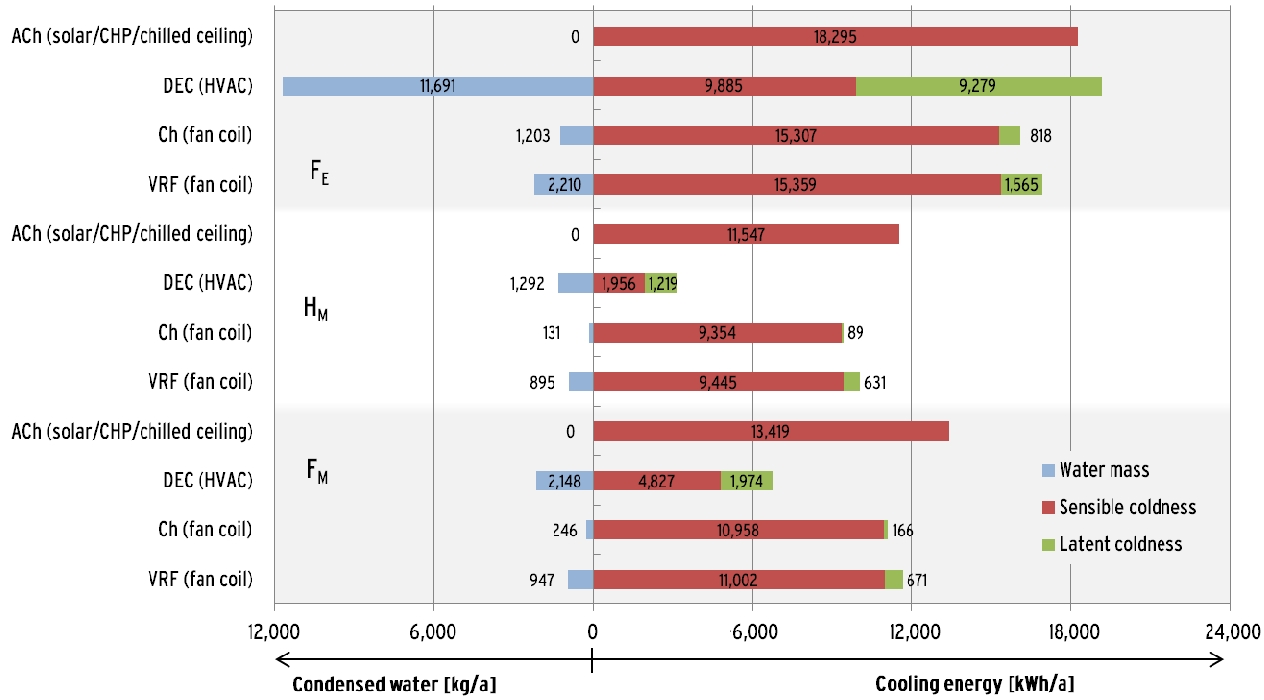
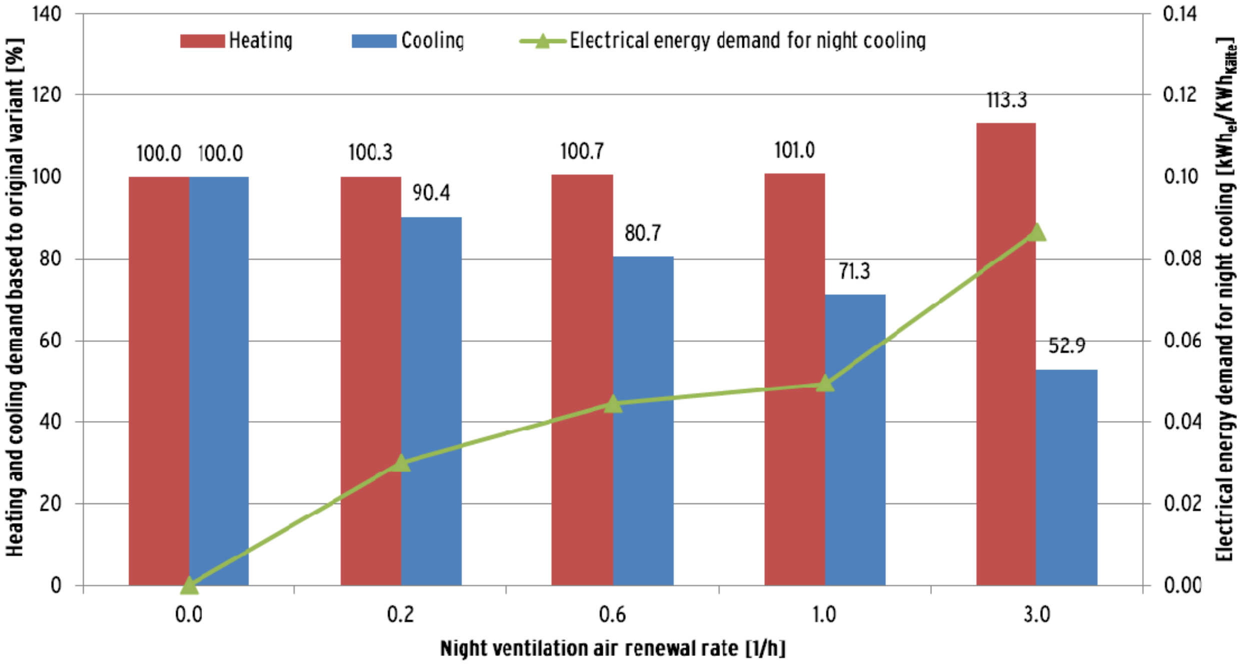


Figure 79: Condensed water, and the latent and sensible cold for various systems variants and climate conditions

Independent of the system for the cooling generation, it is possible to reduce the energy consumption with controlled ventilation during the night. Ambient air has usually a much smaller temperature than the internal temperature of the building. Hence, heat can be rejected from the building by night ventilation. Figure 80 illustrates the results of a calculation for the climate F_E. A moderate air renewal rate of 1.0 h^{-1} reduces the cooling demand by 30 % without significant increase of the heating demand. However, increasing the air renewal rate further reduces the cooling demand, but increases the heating demand even more. Depending on the specific primary energy demand for heating and cooling, an energetic drawback can arise at intensive night ventilation. Moreover, one has to take into account the energy demand for the ventilation. Therefore, it is advantageous to employ ventilation variants with very low electrical energy consumption (e.g., controlled window ventilation). Having an HVAC system installed, it is possible to realise the necessary air renewal rates easily – compared to the energy saving potential (see also Figure 80).



Night ventilation is only active, if the ambient temperature is between 10 °C and building inner temperature, and the office temperature is above 20 °C. The electrical energy demand for the ventilator is calculated according to DIN V 18599-7 and is omitted almost for the case of effective and controlled window ventilation.

Figure 80: Benefit and demand of a concerted ventilation of a building during the night (F_E).

7.3.3.3 TEWI calculation

The TEWI values depend on the refrigerant leakage rate and specific CO_2 emissions of the electricity and heat supply. Beside Figure 81, there are more details to be presented in subsection 7.3.4 in order to illustrate the influence of various factors.

Figure 81 provides an overview of direct and indirect TEWI values of various systems. Values between 850 kg/a CO_2 equivalent with ACh (solely solar) and 5,000 kg/a CO_2 equivalent with DEC system[10] illustrate large differences. However, it has to be recalled, that all the systems fulfil the thermal comfort criteria also quite differently. For a full evaluation of a refrigeration system, further TEWI examinations have to be carried out additionally for the manufacturing. This holds true especially for air conditioning systems with a small number of full-load hours. However, this examination is not within the scope of the present study. The lowest TEWI values are observed for absorption type refrigeration systems and chillers. The direct TEWI shares of VRF systems are twice as high as the indirect ones.

[10] By applying the electricity loss method of the heat.

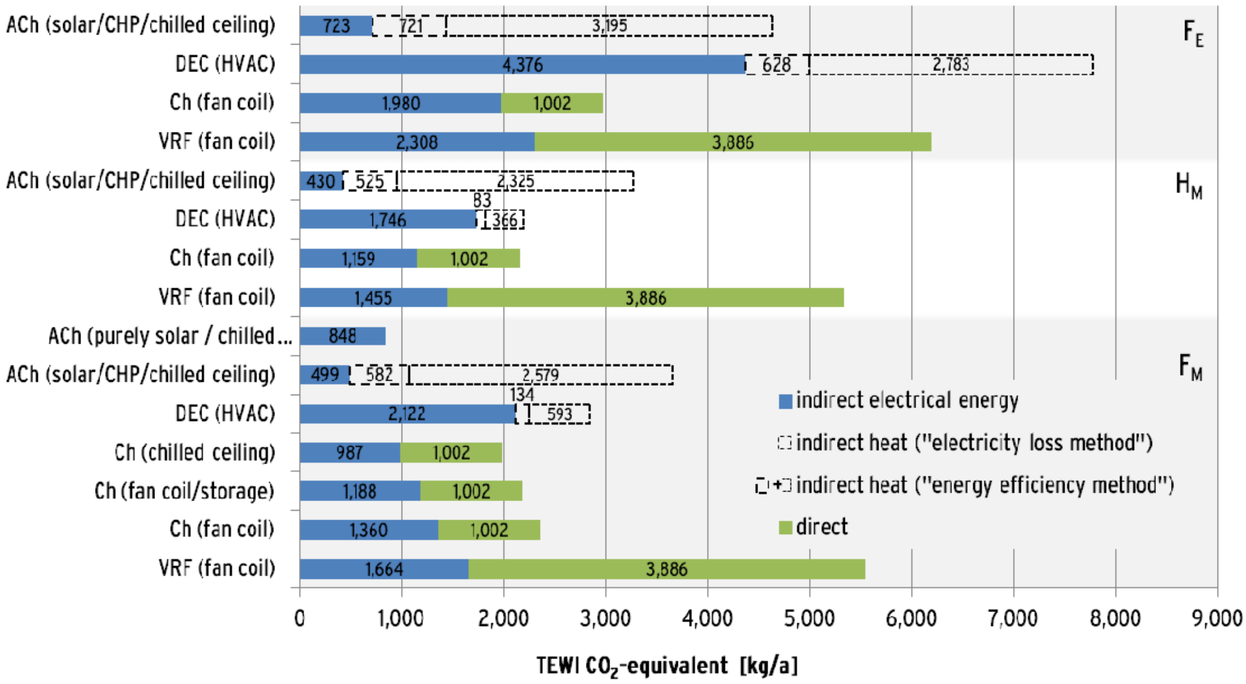


Figure 81: TEWI examinations for various refrigeration systems – itemisation in direct and indirect TEWI shares (the different allocation methods for the heat are explained in subsection 4.2.10, p. 34)

7.3.3.4 Costs of refrigeration systems

The comparison of the total costs of the various office air conditioning systems yields a large spectrum (see Figure 82). From economical point of view, compression type refrigeration systems are much better than the other systems under consideration. Absorption type systems produce more than 50 % and DEC systems more than 130 % additional costs compared to conventional refrigeration systems. The investment costs have the largest influence. Both the ACh and DEC systems have larger investment costs than the compression type refrigeration system total costs. However, in any case, the operation costs contribute only in a limited manner to the total costs. Even with most expensive operation cost scenarios, this statement is valid. The major reason is the small number of full-load hours. For all refrigeration and climate regions, these are below 800 h, which corresponds to a degree of utilisation of less than 10 %. Maintenance and disposal costs do not differ significantly among the various refrigeration systems and are therefore not important for the cost comparison.

A comparison of both compression type refrigeration systems reveals only marginal differences. However, the chiller without storage is slightly cheaper than a VRF system (approx. 7 %). The reasons are both the lower investment and operation costs (see Figure 83). An extension of the chiller with a cold water storage yields significant energy advantages (operation costs are reduced by 10 % in average). However, the investment and maintenance costs increase, leading to larger total costs. The same is true for the substitution of convectors by chilled ceilings. The workings costs are reduced by approx. 27 %, whilst the investment costs are approx. 60 %

larger. For the evaluation of multi-split VRF system one has to consider that those systems provide an additional heating mode of operation (with 3-tube systems also in parallel mode[11]).

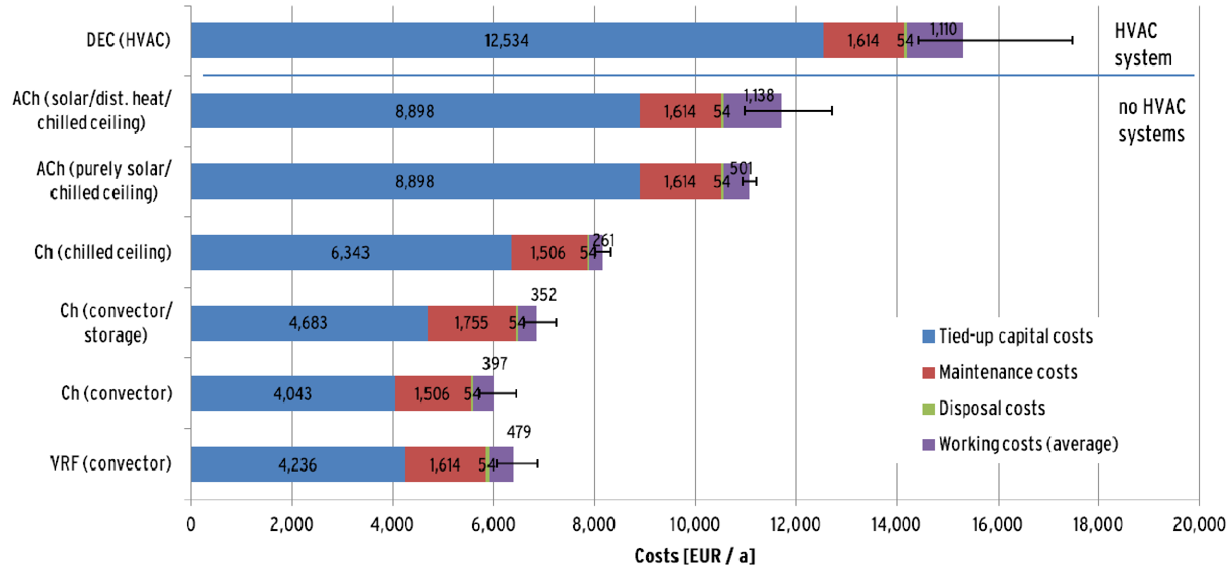


Figure 82: Total annual costs of refrigeration systems for building air conditioning – operation costs are provided with their average value for all cost scenarios (lowest and largest costs are indicated by error bars)

Figure 83 provides an itemisation of the investment costs. It is visible that the costs for the cold circuit and the cold distributor prevail, whereas the costs for the cooling generator are of minor importance. For ACh, the cooling generation costs including re-cooler are significantly larger (the re-cooler is integrated into the cooling generator in VRF and chiller systems). In addition, the use of solar collectors increases the additional costs, yielding a large difference to the classical systems. It is also visible in Figure 83 that the investment costs for DEC systems are mostly triggered by their cooling generator costs. The reason originates in the lower limit for the fresh air temperature at adiabatic desiccative cooling conditions. For covering the cooling demand, large volume flow rates are required which lead to large system dimensions increasing the costs. However, it should be stressed that DEC systems belong to the HVAC category and are therefore not directly comparable. Contrary to the other systems, it is possible here to renew the air continuously. In conjunction with a concerted humidification, this system provides a much better air quality. The efficient heat recovery implemented into the system offers advantages in the heating period, too. However, this effect has not been within the scope of the present study.

The operation costs for climate F_E are visualised in Figure 84. Compression type refrigeration systems are cheapest for cost scenarios, since they have costs neither for water nor for heat. For absorption type refrigeration systems, a wet cooling tower is disadvantageous for the operation costs. The resulting costs for water exceed the savings of electrical energy greatly, compared to

[11] Parallel cooling and heating of particular indoor units which are connected to the same outdoor unit is possible and leads to a reduced load in the outdoor unit heat exchanger. This is particularly interesting for server rooms with their usually demand extending well into the heating period.

a classical system without cold storage. Alternatively, the application of hybrid or dry re-coolers is cogitable. However, hybrid re-coolers are much more expensive and dry re-coolers have the disadvantage to have large re-cooling temperatures at large ambient temperatures, reducing the cooling power of the system.

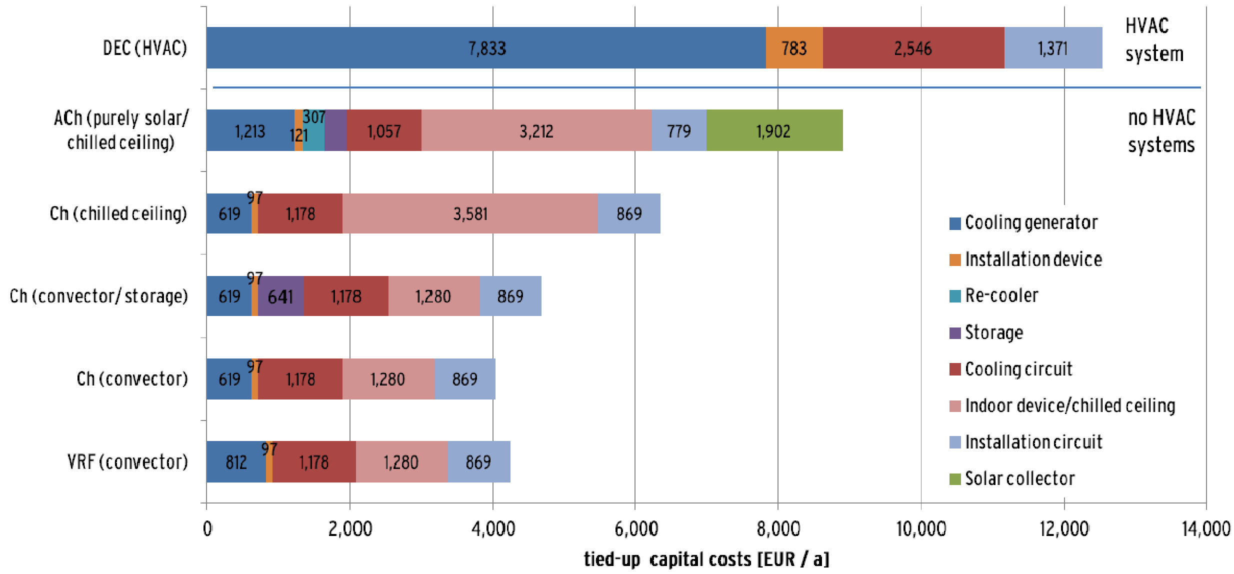


Figure 83: Constitution of the investment costs of various refrigeration systems

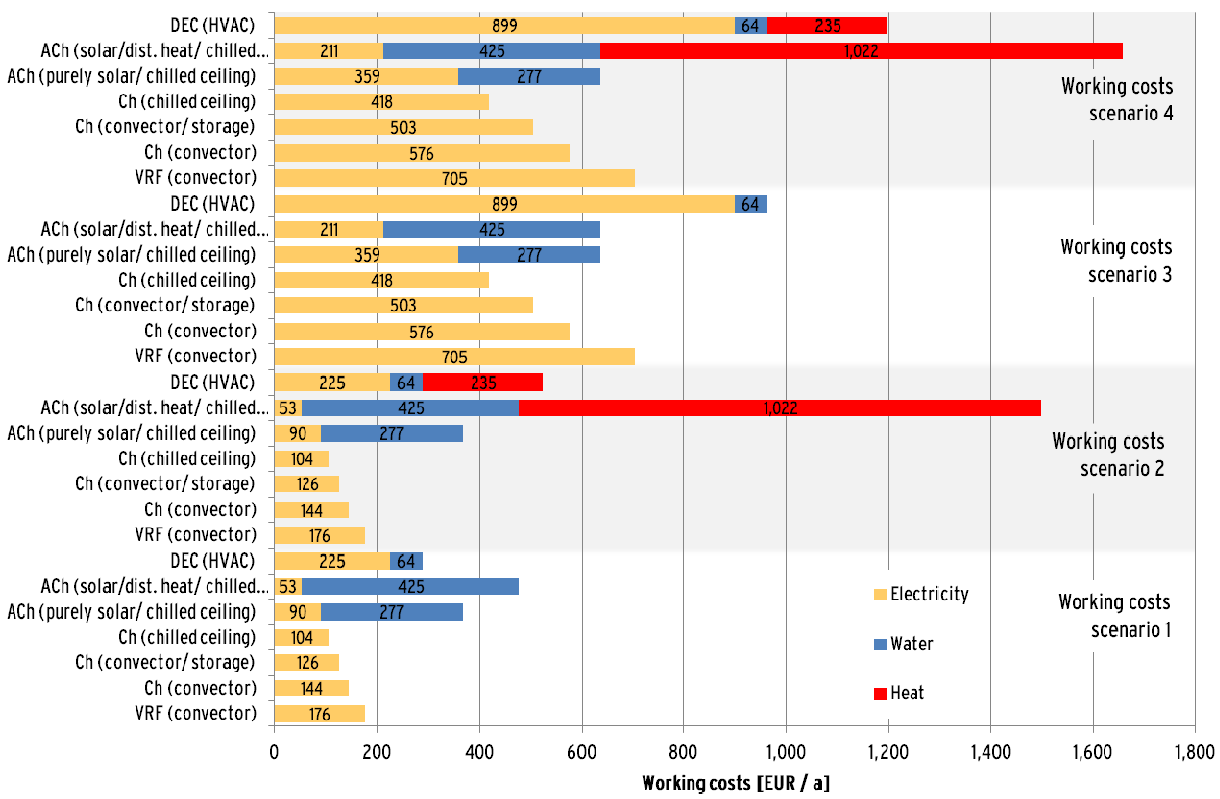


Figure 84: Composition of the operation costs for various scenarios (climate: F_M)

Concluding the cost investigation, it can be stated that due to the low degree of utilisation, the investment costs dominate in the present application. Alternative systems, such as ACh or DEC, are economically disadvantageous due to their larger investment costs.

7.3.4 Discussion of particular systems

7.3.4.1 Variant 1: ACh — System with solar driven absorption type refrigeration plant

The solar driven absorption type refrigeration system in conjunction with backup heat supply from a CHP plant has the lowest TEWI emissions of all refrigeration systems under consideration. Operation without backup heat does not reduce the TEWI and leads to a worse cooling demand coverage. The reason for this is the larger number of utilisation hours of the whole refrigeration system. The power of the refrigeration system decreases due to lower driving temperatures from the solar system. Furthermore, a lower air temperature reduction can be obtained at constant power demand for pumps and ventilators, yielding a longer run-time of the ACh.

Various solar collector variants, including *FM-ColS-DH* (flat-plate collector, 90 m^2, south-orientation, and district heat backup), and *FM-ColS* (flat-plate collector, 90 m^2, without district heat backup) are presented in Figure 85 and Figure 86. The other variants have other collector areas, collector orientations, and various chilled water storage sizes:

- *FM-ColS60-DH*: collector area of 60 m^2 instead of 90 m^2
- *FM-ColS180-St20*: collector area of 180 m^2 instead of 90 m^2, chilled water storage 20 m^3 instead of 2 m^3 (6...12 °C), no backup system
- *FM-ColEW-DH*: collector area separated into two parts that are oriented to east and west (setting angle 45° instead of 30° against horizon)
- *FM-ColEW*: as *FM-ColEW-DH*, but without backup heat supply

Systems with a backup system (*FE-ColS-DH*) have a solar coverage ratio of maximum 40 % only (see Figure 85). The reduction of the solar coverage ratio when reducing the collector area to 60 m^2 is 7 % for scenario F_M. This results in an additional district heat demand of approx. 1,500 kWh/a. Hence, an exclusive utilisation of 90 m^2 collector area for cooling generation is not economical.

Purely solar driven variants obtain cooling demand coverage ratios[12] between 42 % and 91 % (climate F_M). However, despite the large coverage ratio for *FM-ColS180-St20* of 91 %, room air temperature exceedings of above 3 K occur during approx. 200 hours, between 2 and 3 K during 130 hours, and between 1 and 2 K during 150 hours. This implies that also with a large system effort, purely solar driven systems cannot ensure comfortable room air conditions.

Seasonal energy efficiency ratios based on the heat demand are between 0.63 and 0.68 for all variants. However, based on the electrical energy demand, large differences arise (see Figure 86). Both purely solar driven systems *FM-ColS* and *FM-ColEW* have electrical SEER of 5.8 and 3.9, respectively, which is approximately the same as for compression type systems (VRF and chiller). Therefore, purely solar driven systems are hardly justifiable from both economical and ecological point of view. Increasing the collector area and the storage volume leads to larger

[12] The cooling demand coverage ratio is defined to be the ratio of actually transferred cooling energy versus the cooling energy demand for obtaining comfortable conditions.

electrical SEER (*FM-ColS180-St20*: electrical SEER of 9.1). However, such an increase is only approachable with larger investment costs. Systems with district heat backup obtain electrical SEER of more than 15.

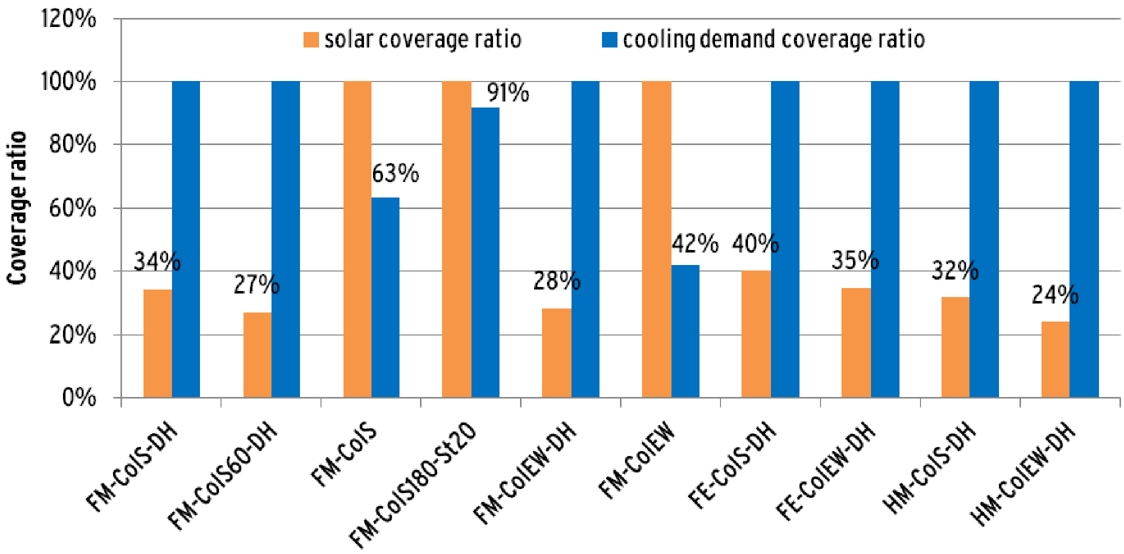


Figure 85: Solar coverage ratio and cooling demand coverage ratio for various system configurations with absorption type refrigeration systems

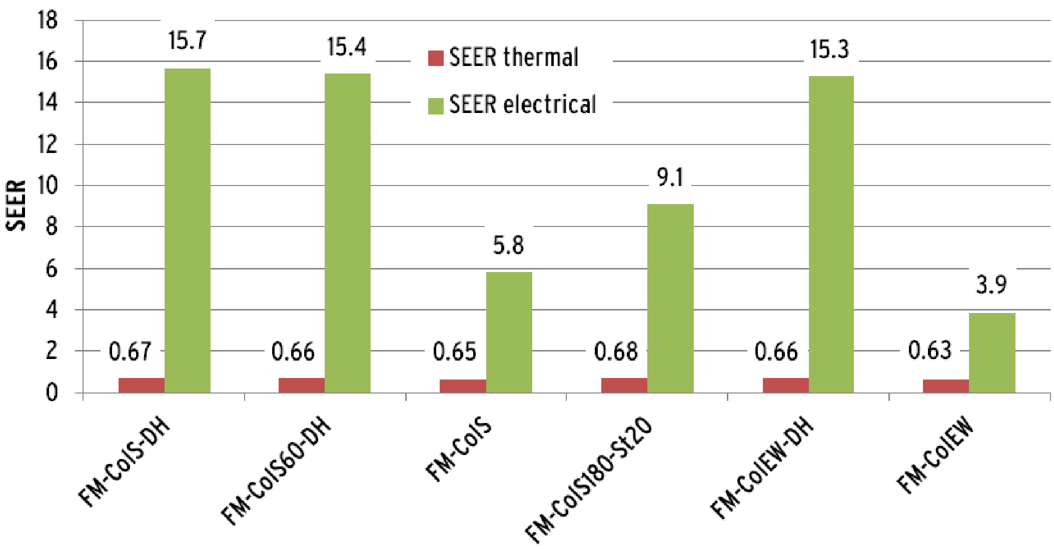


Figure 86: Seasonal energy efficiency ratios (SEER) based on heat and electrical energy demand for various configurations with absorption type refrigeration systems

In summary, the following statements can be made related to thermally driven refrigeration systems:

- Purely solar driven systems are not useful from economical and ecological point of view and do not fulfil the thermal comfort criteria for the climate zones under consideration.
- In conjunction with a district heat backup from CHP systems, the situation is different and observed to be positive from ecological and energy economical point of view. Both the primary energy demand and the TEWI emissions can be smaller than for other systems, but depend strongly on the availability and the evaluation of the thermal energy.
- The utilisation of the solar collector area shall be extended to other applications (heating, hot water supply) during the non-cooling season due to economical advantages.
- Prerequisite for efficiency and cost effectiveness of complex solar driven systems are forecast-based control strategies with a variety of input parameters, such as weather and cooling demand forecasts.

7.3.4.2 Variant 2: VRF – Multi-split VRF system

Figure 87 illustrates the frequency of occurrence of different loads during the operation of a VRF system. It is obvious that VRF systems are usually operated in low part-load conditions and 80 % of the operating time at a part-load ratio below 40 %. Especially at these small part-load ratios, this particular system type has a strong efficiency decline. Among others, the reasons are:

- Missing internal cooling power storage; the cooling power has to be adjusted according to the conditions by a control scheme (compressor speed, hot gas bypass)
- Operation of a motor at low numbers of revolutions and small loads far from efficiency optimum
- Minimal refrigerant mass flow rates required for oil separation. I.e., a bypass operation at very small cooling powers yields a significant reduction of the energy efficiency ratio. For this particular mode of operation, the compressed and hot vaporous refrigerant is lead directly to the inlet of the compressor.
- Upper limit of evaporation temperature (usually max. 6 °C in indoor unit)

Figure 88 illustrates the relative generated cooling energy (based on total cooling energy) at different part-load classes of the VRF system. Compared to Figure 87, the curve is shifted to larger power classes, since there is more cooling energy generated at similar running times in larger power classes. Since lower part-loads originate in lower ambient temperatures (outer loads are lower), the average SEER values are relatively large due to the lower temperature differences to be overcome. The resulting distribution of cooling SEER is visualised in Figure 89.

Of remarkable influence is the diameter of the gas suction pipe. The energy demand increases by 20 % for the system under consideration, by decreasing the suction pipe diameter from 33.7 to 21.3 mm for a length of 50 m. Hence, the design of refrigerant pipes is critical for the system. It shall be stressed that VRF systems have good possibilities for individual control.

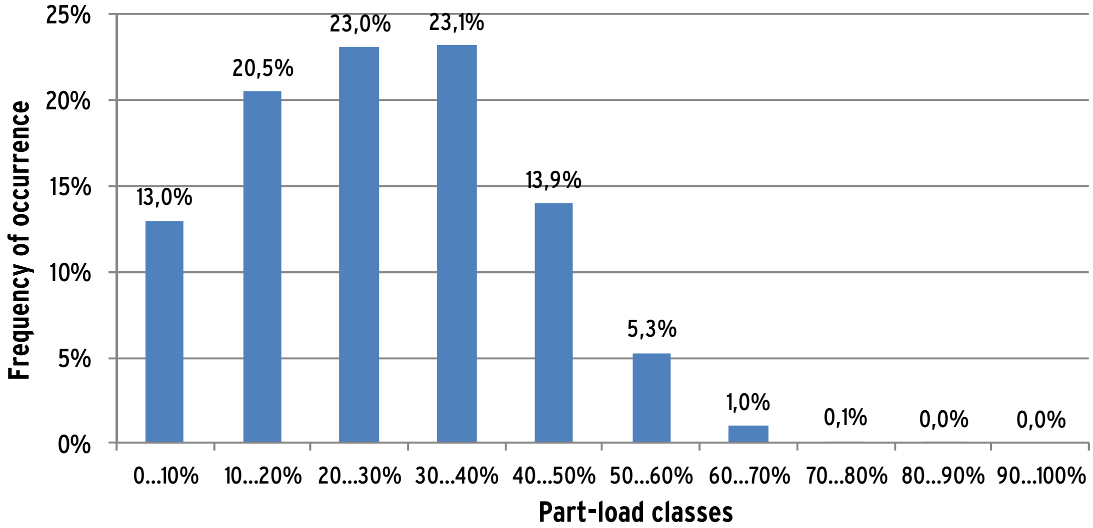


Figure 87: Frequency of occurrence distribution of part-load classes of a VRF system (operation times considered only, climate: F_M)

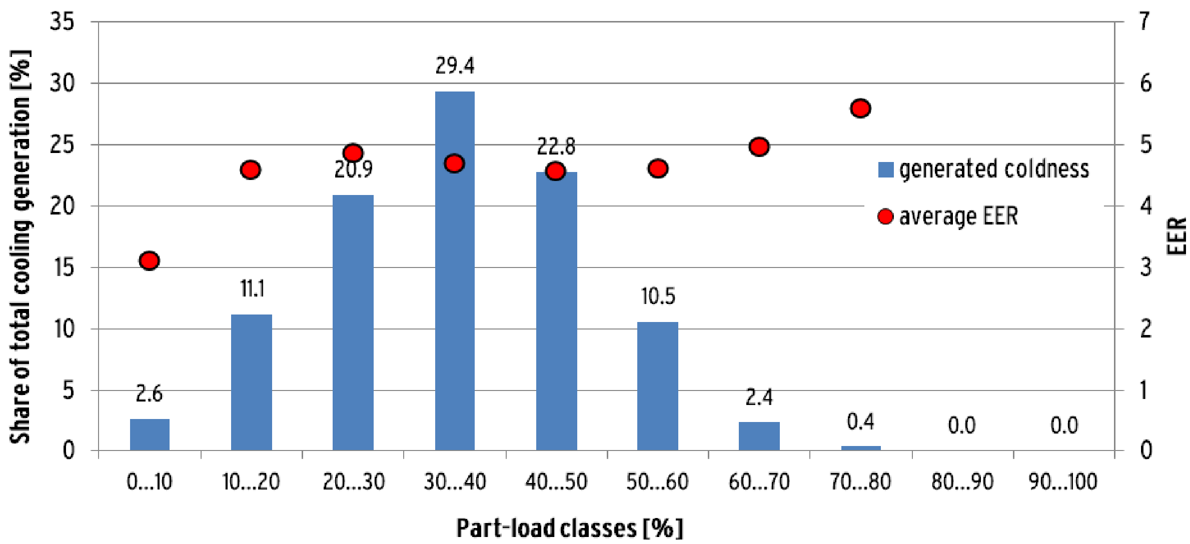


Figure 88: Itemisation of the relative generated cold (based on total cooling generation) versus the part-load classes of a VRF system (climate: F_M)

Major reason for the large TEWI values of VRF systems are the large amounts of refrigerant in conjunction with relatively large leakage rates (see Table 14). Approximately 70 % of the total TEWI is caused by direct emissions. A reduction of the refrigerant charge is hardly possible due to the long pipes filled with refrigerant. Moreover, detachable screw joints are employed for those installations leading to large leakage rates (see subsection 7.2.2, pp. 85ff.). Figure 90 illustrates the influence of the leakage rate on the total TEWI emissions. The total TEWI emissions can be reduced between 20 and 23 % if the leakage rate of maximum 4 % is obtained as required by the German ordinance ChemKlimaSchutzV (see Table 43 in the appendix).

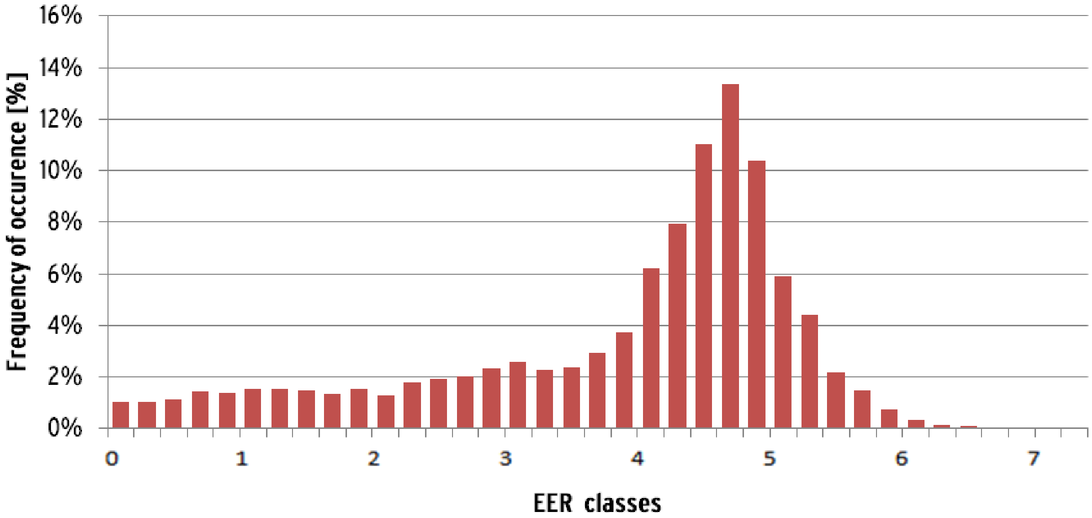


Figure 89: Frequency of occurrence distribution of VRF system EER during operation time (climate: F_M)

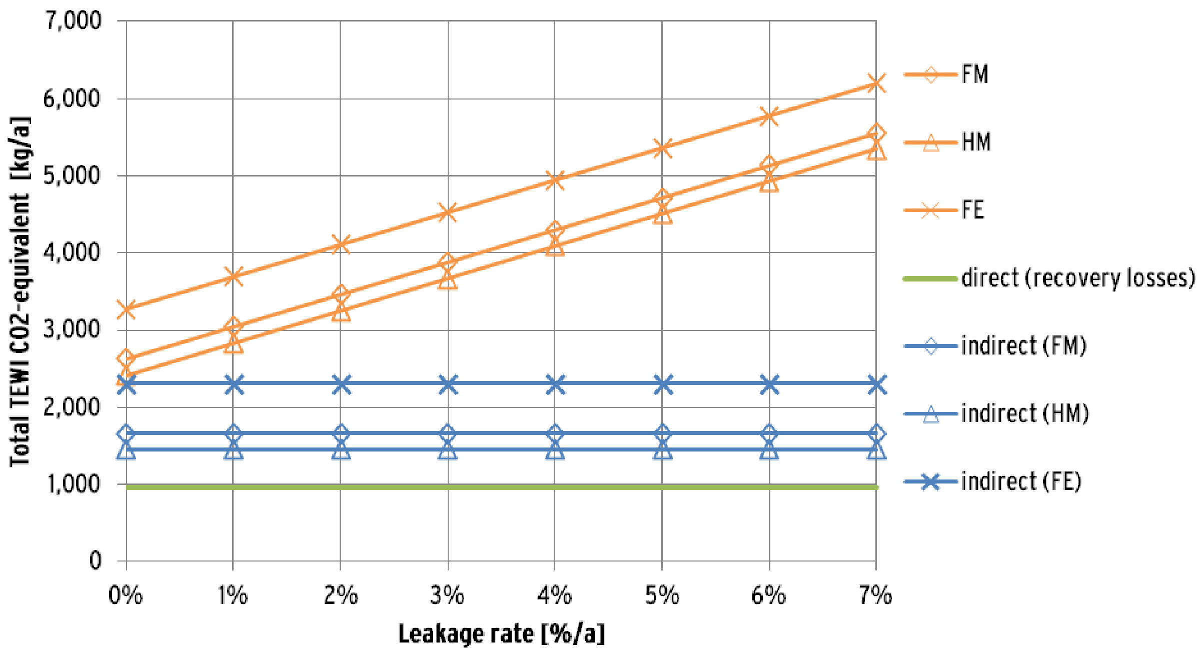


Figure 90: TEWI versus the leakage rate for VRF systems with R410A

7.3.4.3 Variant 3: Ch — Chiller operated with a compression type refrigeration plant

The chiller has a lower final energy demand than the VRF system in the present comparison. One of the reasons for this is the relatively large chilled water supply temperature (10 °C for convector systems, and 14 °C for chilled ceilings). Moreover, it is possible to reduce the electrical energy demand by employing chilled ceilings, since the ventilators for cold distribution can be avoided.

Different refrigerants (R410A, R290, R134a, R717, R718, and R1234yf) have been employed for chiller systems. Figure 91 illustrates the rough results for the seasonal energy efficiency ratio (SEER) of the refrigeration system. In order to facilitate the simulations, the constructive set-up in the numerical model is the same for all refrigerants. For instance, utilising R290 requires an internal heat exchanger for super-heating the gas prior to entering the compressor. The results are strongly oriented on the pure refrigeration circuit calculations (see Figure 42 and Figure 44, p. 65f), but with demand profiles of various system components (pumps, ventilators) comprising daily and seasonal fluctuations. Moreover, it is visible that the choice of the refrigerant has a significant influence on the energy demand of the refrigeration system. For all natural refrigerants (R290, R717, and R718) and for R134a, relatively large SEER can be obtained. Significant lower SEER can be observed for R410A, R32, and especially for R1234yf.

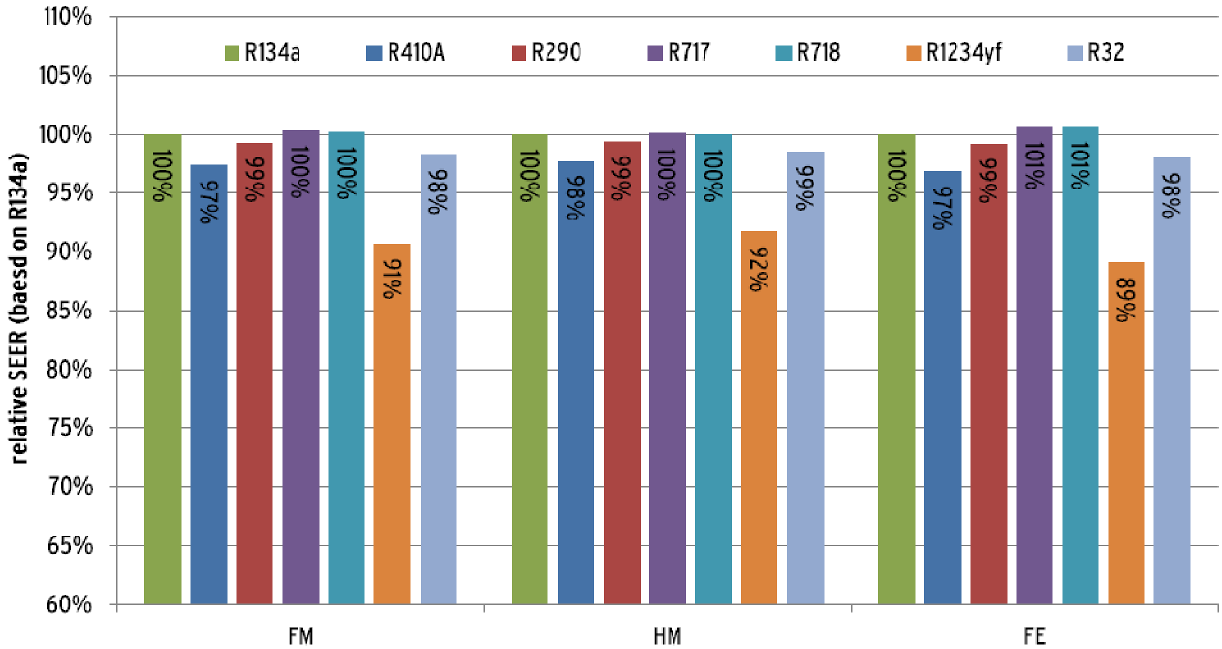


Figure 91: Calculated influence of various refrigerants on the seasonal energetic efficiency ratio of the chiller system

An additional measure for the investigation of the refrigerant influence on the system efficiency is the evaluation of the energy efficiency ratios of systems available on the market. Data sheets of various manufacturers and the database of Eurovent is utilised for this purpose. The large majority of the data can be collected for the common HFC refrigerants, such as R134a, R410A, and R407C, whereas data for the natural refrigerants R290 and R717 are rare. No data are available for the refrigerants R718, R1234yf, and R32. Figure 92 illustrates all air and water-cooled systems with a cooling power below 500 kW. Those of the values are considered for which the following conditions are valid:

- 35 °C ambient air temperature (air-cooled) and 30/35 °C cooling water temperature (water-cooled)
- 12/7 °C chilled water temperature

The energy efficiency ratios (EER) of air-cooled systems are quite independent of their nominal cooling power and are observed to be 2.8 averaged over all refrigerants. Considering water-cooled systems (see Figure 92, right-hand side), no significant trend is visible. However, the EER is far below average for very small capacities (<20 kW). The average EER is 4.5 and significantly larger than the value for air-cooled systems.

Figure 93 provides a comparison of various refrigerants by means of relative EER (based on the EER of R134a), which illustrates the results of theoretical considerations. R134a has the best results among the HFC refrigerants. However, natural refrigerants, such as R290 and R717 are much better. Compared to the results in Figure 91, the values are even better. The reasons might be:

- Small data basis for natural refrigerants
- Superior component design (sales pitch, unique selling point).

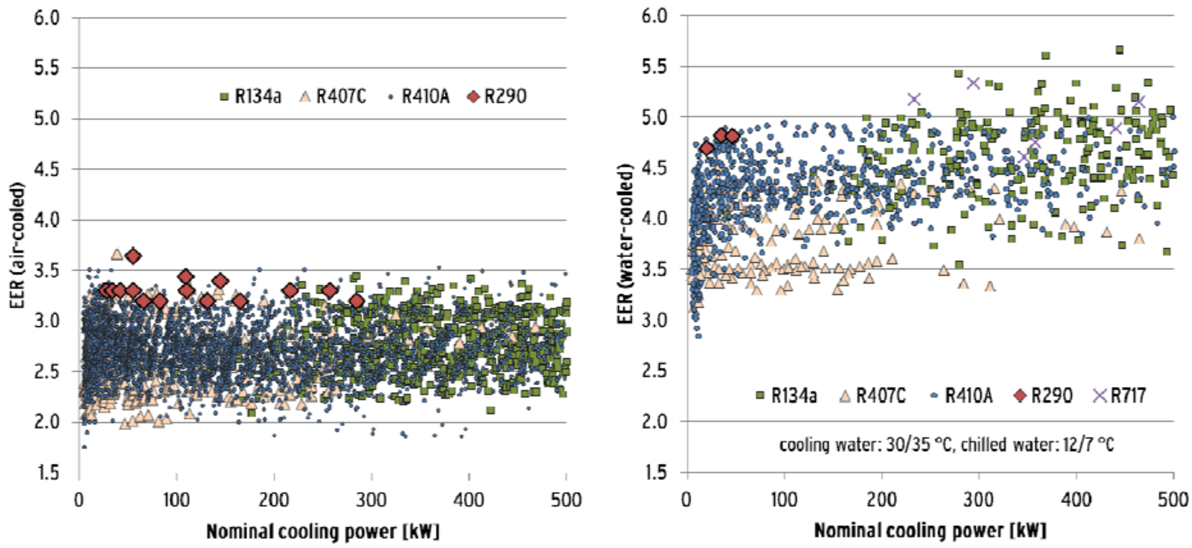


Figure 92: Market overview of energy efficiency ratios EER (at nominal cooling power) for air (left-hand side) and water-cooled (right-hand side) chillers utilising various refrigerants and nominal cooling powers (sources: Eurovent, various manufacturers)

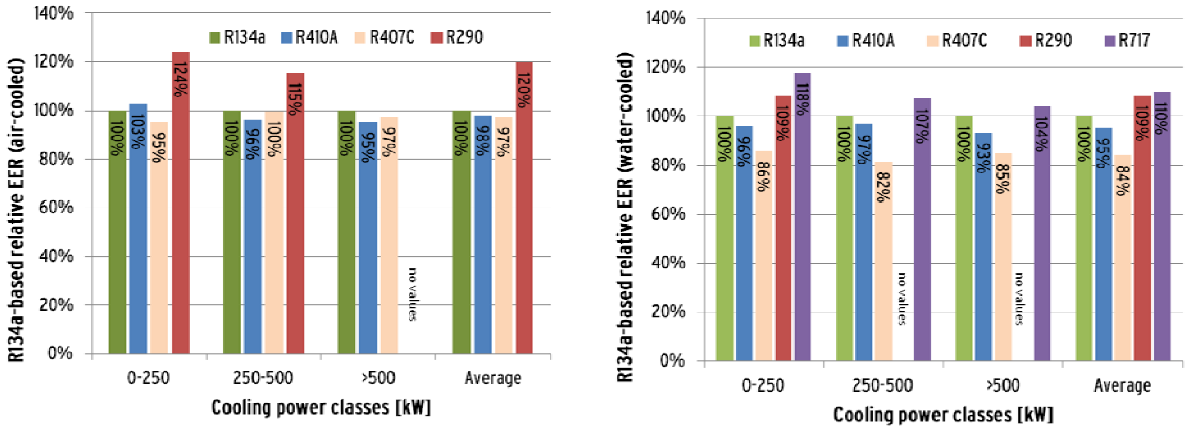


Figure 93: Relative comparison of the energy efficiency ratio (based on R134a) for various refrigerants (compare to Figure 92)

The potential influence of the refrigerant on the total TEWI is illustrated for a chiller in the climate F_M in Figure 94. The differences in the direct shares originate in the different GWP values of the refrigerants. The indirect shares have been derived from simulation calculations. This allows an evaluation of the direct TEWI share from the total TEWI. The refrigerant R410A has the largest direct TEWI share of approx. 40 %, followed by R134a with approx. 32 % and R32 with approx. 18 %. For the HFC R1234yf and the natural refrigerants R290, R717, and R718, the direct TEWI share has a very low or no influence on the total TEWI.

The influence of HFC refrigerants is always depending on the particular application and system conditions (degree of utilisation, leakage rate, etc.). The comparison illustrates in principle how large the influence of the GWP values of a particular refrigerant on the total climate impact of a chiller is (see Figure 94). This influence is especially large for systems with a small degree of utilisation (e.g., below 500 full-load hours per year). Although the direct TEWI share is smaller for chillers due to a smaller refrigerant charge compared to direct evaporation systems, the TEWI is greatly influenced by the choice of the refrigerant. The application of natural refrigerants can provide an important contribution reducing the climate impact of refrigeration systems.

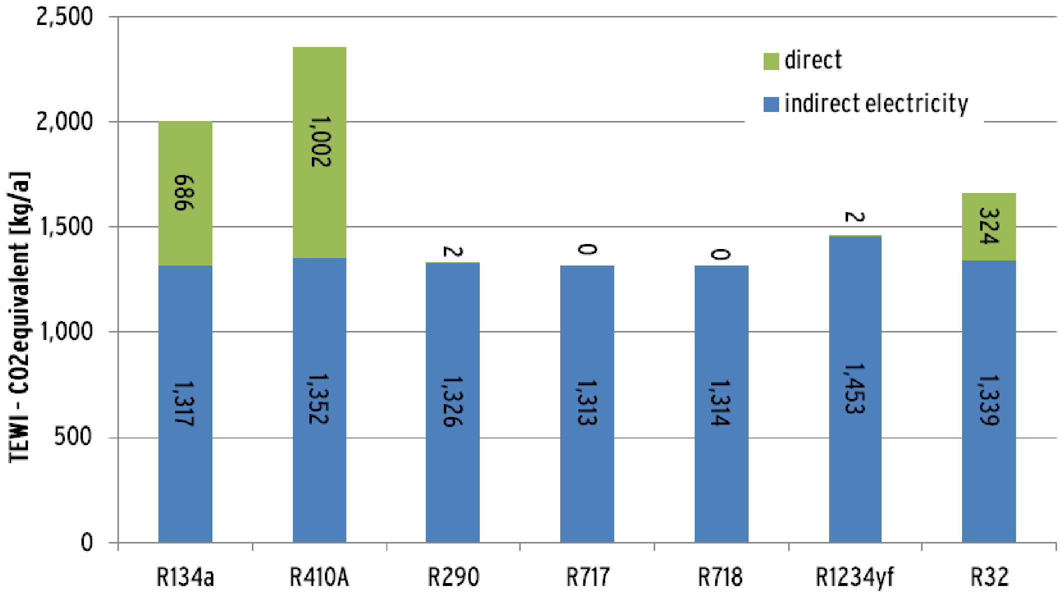


Figure 94: TEWI considerations for a chiller with various refrigerants for climate F_M

7.3.4.4 Variant 4: HVAC — Humidification, ventilation, and air conditioning with DEC systems utilising CHP waste heat

The combination of HVAC systems with DEC system provides several advantages:

- No refrigerant and hence no direct TEWI share
- Fresh air humidity can be controlled
- Large air renewal rates yield good air quality
- Efficient system-inherent heat exchanger yields large heat recovery ratio (0.8), which is especially important at low ambient temperatures

These advantages trigger the disadvantageous increased energy demand. In comparable conventional systems, much smaller air renewal rates are necessary (5.4 for DEC and 0.6 for other systems). These large air renewal rates for DEC systems are required for covering the cooling demand. It is assumed that the other systems have average demands regarding air quality. For the case of larger demands on air quality or at increased immissions into the room (e.g., in smoker areas), air renewal rates have to be adjusted (DIN-EN-13779 2005). This leads to an increase of the ventilator power and hence to an increased energy demand of the system. For such a case, a DEC system will be the better choice, since it already provides large air renewal rates. Moreover, only with this system it is possible to control the air humidity below thermal comfort thresholds by dehumidification. The extracted water amount is much larger than in comparable systems (see Figure 79, p. 106), but also due to large air renewal rates.

A great drawback of a DEC system is that large cooling loads can be covered only with a large ventilation energy effort or even not entirely. A combination of a DEC and CRP system might lead to a lower energy demand, but the TEWI of the entire system due to refrigerant leakage will be increased in this case, depending on the refrigerant.

7.4 Industrial refrigeration

7.4.1 Selected refrigeration systems

7.4.1.1 Variant 5: Ch – Chiller

The principal composition of chillers for industrial refrigeration does not differ from that for building air conditioning. The cold transfer from the refrigerant to the cooling medium is carried out with the same minimal temperature difference, independent of the refrigerant itself. The cold delivery to the customer is not modelled in detail, but considered using a thermal balance.

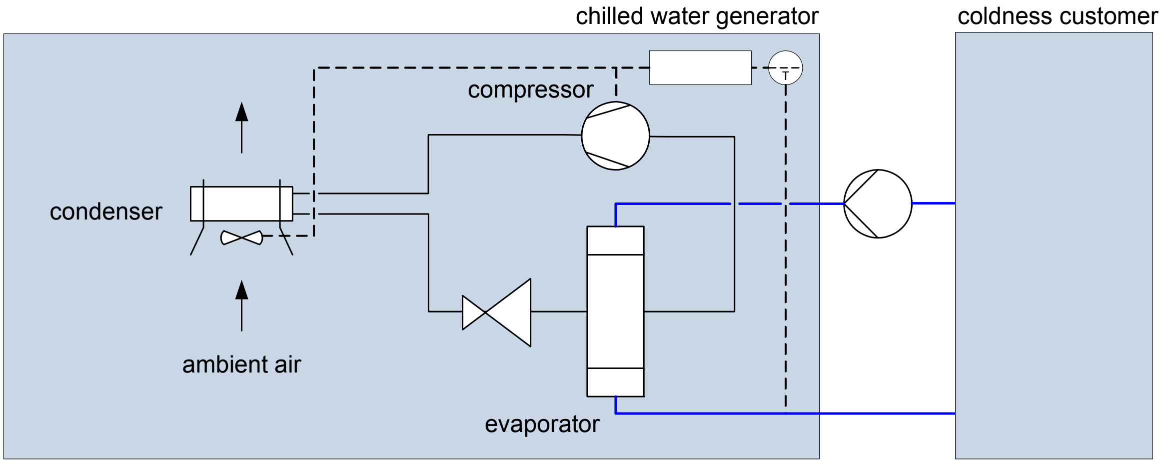


Figure 95: Schematic diagram of a chiller operated with a compression type refrigeration system

Table 23: System parameters – Variant 5 (industrial refrigeration, chiller)

Parameter	Symbol	Value	Unit
Cooling demand	$\dot{Q}_{0,N,Ind.\,Ch}$	500	kW
Evaporation temperature	$t_{0,Ind.\,Ch}$	-1	°C
Supply temperature of the cooling medium	$T_{sup,Ind.\,Ch}$	2	°C
Refrigerant	-	R410a	-
Refrigerant amount	m_{Ref}	250	kg

7.4.1.2 Variant 6: ACh – absorption type refrigeration system operated by waste heat of a CHP plant

The ACh is operated with the refrigerant-absorbent combination ammonia/water and driven by waste heat from a CHP plant. In order to obtain a constant chilled water supply temperature, the system has a chilled water storage, and, moreover, the nominal supply water temperature for the storage is slightly lower (0.5 K) than the nominal supply temperature for the customer. It can be adjusted with a reflux mixture. The waste heat of the process is rejected to ambient with a dry re-cooler with an adiabatic air pre-cooling. The adiabatic pre-cooling is active for temperatures above 16 °C.

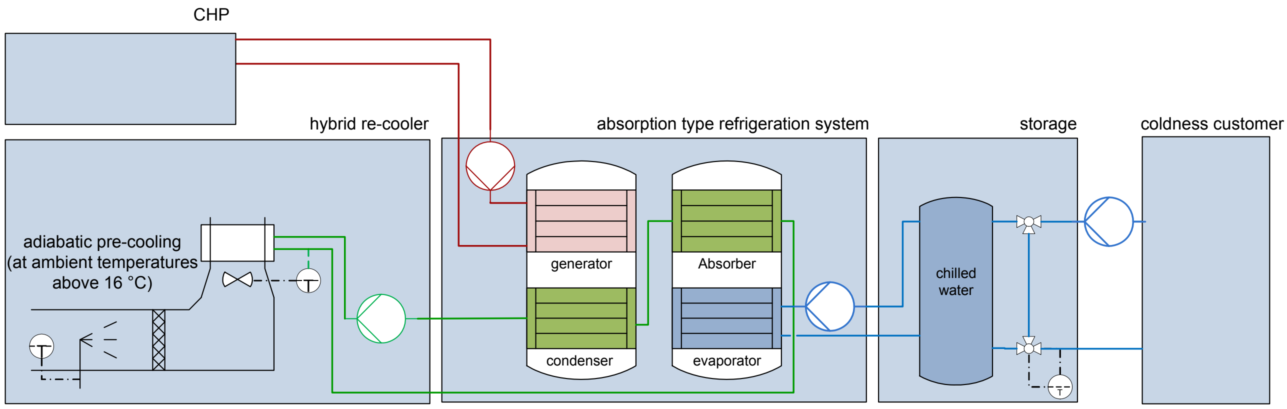


Figure 96: Schematic diagram of an absorption type refrigeration system operated with CHP waste heat

Table 24: System parameters – Variant 6 (industrial refrigeration, ACh)

Parameter	Symbol	Value	Unit
Cooling demand	$\dot{Q}_{0,N,Ind.ACh}$	500	kW
Nominal power of the re-cooler[13]	$\dot{Q}_{RC,N}$	2616	kW
Waste heat of the CHP plant	$\dot{Q}_{WH,N,CHP}$	860	kW
Volume chilled water storage	$V_{St,ChW}$	2.0	m^3
Refrigerant-absorbent combination, ACh	-	NH_3/H_2O	-

7.4.1.3 Variant 7: Direct evap. – Systems with direct evaporation

The direct evaporation system is quite similar to the VRF for building air conditioning. The only difference is that the cold transfer is not modelled explicitly. The constraint is that the demanded cooling power is provided by a pre-determined evaporation temperature.

Table 25: System parameters – Variant 7 (industrial refrigeration, direct evaporation)

Parameter	Symbol	Value	Unit
Cooling demand	$\dot{Q}_{0,N,Ind.\,DE}$	500	kW
Evaporation temperature	$t_{0,Ind.\,DE}$	2	°C
Refrigerant	-	R723, R134a	-
Refrigerant amount	m_{Ref}	350	kg

[13] Data sheet AIA re-cooler model XP120-1, for $t_{KüW}$ = 40/35 °C at t_{TK} = 25 °C, motor speed B Creating Citation...

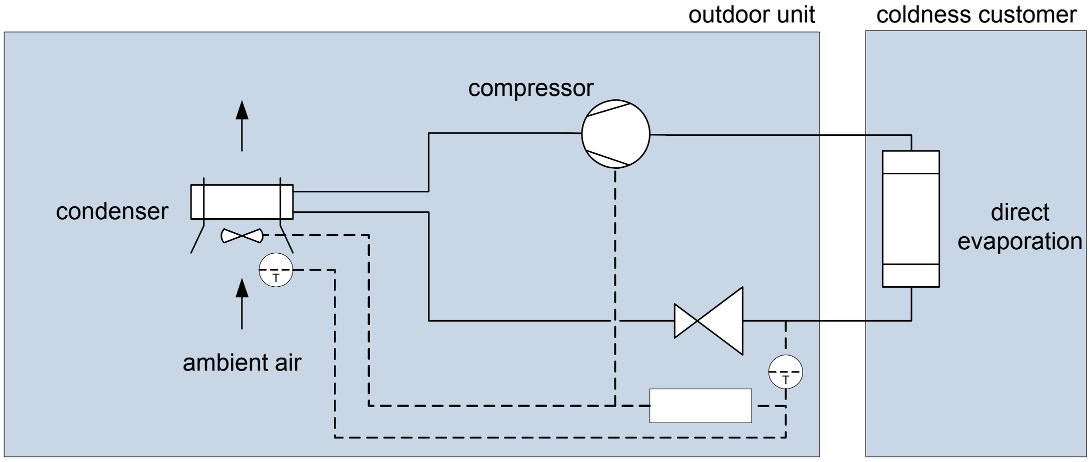


Figure 97: Schematic diagram of the simulated system with direct evaporation

7.4.2 Modelling

7.4.2.1 Model of the industrial customer

For the comparisons of the industrial refrigeration systems, a constant cooling load of 500 kW at a chilled water supply temperature of +2 °C is assumed. For direct evaporation systems, the evaporation temperature is also set to be +2 °C.

7.4.2.2 Refrigeration systems

The assumptions for the industrial refrigeration systems are similar to those for the building air conditioning (see subsection 7.3.2.2, p. 99). Further information of the modelling of particular industrial refrigeration systems are provided in sub-section 11.2.2.

7.4.3 Results

The design and control of all systems under consideration is carried out on the basis of complete demand coverage for the entire time-span. The assumption of a constant cooling demand yields large part-load differences of the refrigeration system in winter and summer. An itemisation into single systems within a cooling network in order to reduce the part-load hours can increase the efficiency significantly. Within the present study, systems are always considered single systems.

7.4.3.1 Energy demand

Figure 98 illustrates the final energy demand for electrical and thermal energy for the industrial refrigerant systems under consideration. The electrical energy demand of an absorption type refrigeration system is 220 to 240 MWh/a and roughly 40 % of the demand of direct evaporation systems. The electrical energy demand of direct evaporation systems with both R723 and R134a is virtually the same (relative difference below 1 %) for all climate scenarios. Indirect systems require approx. 18 % more energy than direct systems.

Only absorption type refrigeration systems have a heat demand, which is very large (approx. 7,500 MWh/a) compared to the electrical energy demand of other systems. The reason is the efficiency characteristic of the absorption type refrigeration plant. The resulting heat ratio does not increases steadily with decreasing re-cooling temperatures, as it is the case for compression type refrigeration systems (see Figures 105 and 106, and the discussion of the absorption type refrigeration system in subsection 7.4.4, pp. 131ff.). The application of absorption type refrigeration systems is economically and ecologically feasible only when the driving heat is waste heat otherwise rejected to the ambient (e.g. waste heat from some process).

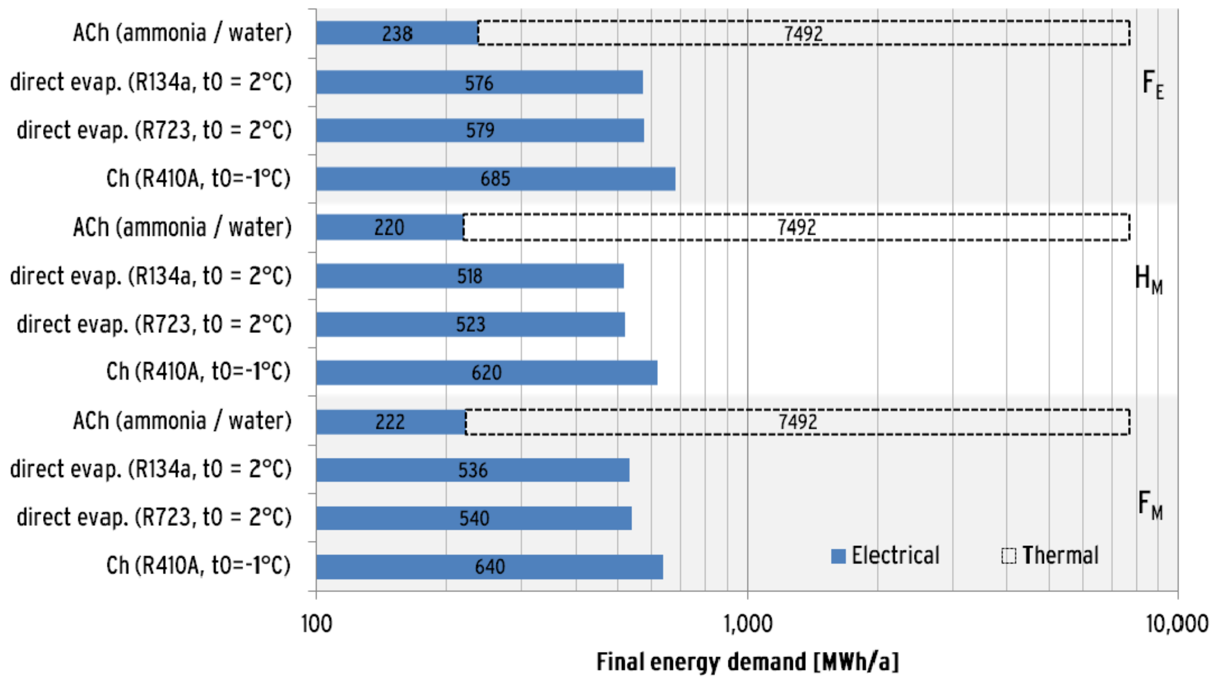


Figure 98: Electrical and thermal final energy demand of various industrial refrigeration systems at different climate scenarios (500 kW cooling power at +2 °C chilled water or evaporation temperature at the cold distribution year-round)

Figure 99 illustrates the primary energy demand of various system variants. The primary energy demand is calculated by means of factors provided by EnEV 2014, which has been developed for the building sector. It can be observed that the absorption type refrigeration systems also have a much larger total primary energy demand than CRP systems have. This leads to the conclusion that the utilisation of ACh system is superior to compression type systems only for the case of less primarily energetic costly waste heat.

Switching to free cooling at low ambient temperatures yields a reduced thermal energy demand. However, the calculations exhibit only a very limited potential for this particular method. A reduction of the thermal energy up to 5 % is accompanied with an increase of the electrical energy demand. Moreover, the process can be operated with free cooling rarely only, since ambient temperatures below -2 °C are required for cold temperatures of +2 °C. These conditions are available only 5 % of the time of a year.

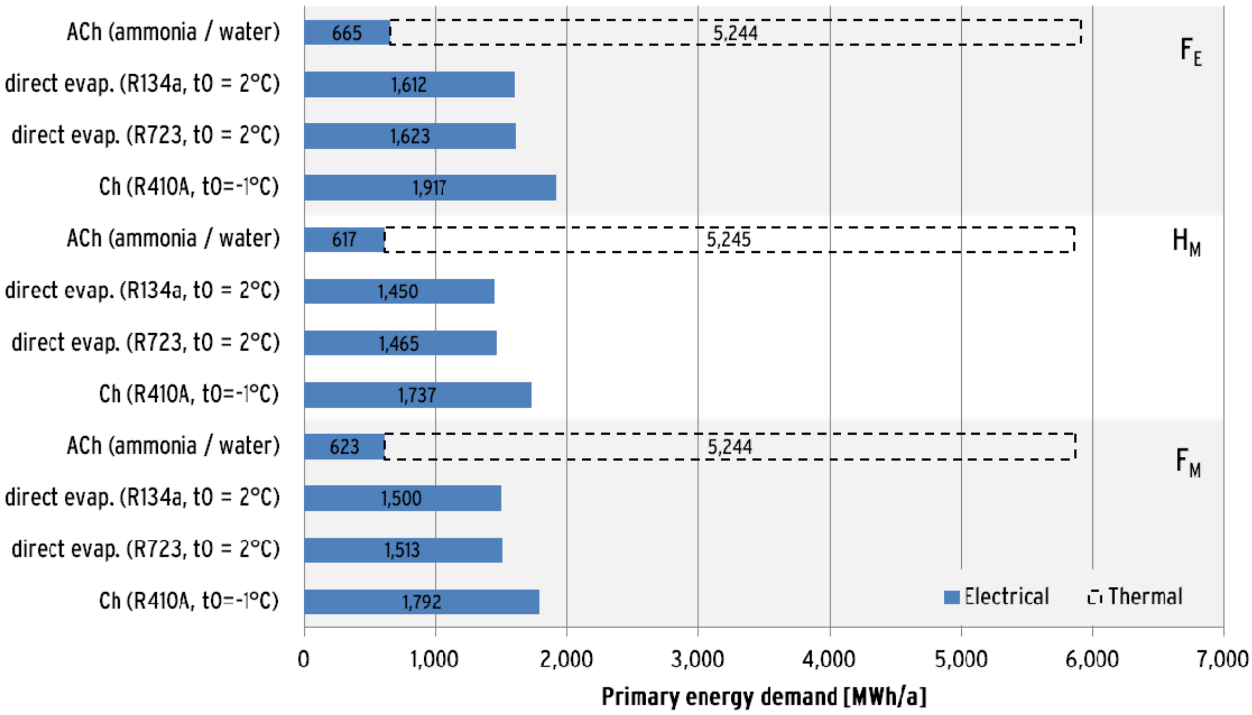


Figure 99: Primary energy demand for various industrial refrigeration systems for different climate scenarios (500 kW cooling power year-round at +2 °C chilled water/evaporation temperature at the cold distribution)

7.4.3.2 TEWI

The TEWI emissions of industrial refrigeration systems are illustrated in Figure 100. It can be observed that the direct emissions are far less important than they are for building air conditioning systems. The reason is the large number of full-load hours. Beside the leakages, the refrigerant amount is important for the direct TEWI. This has been estimated as 0.5 kg/kW and 0.7 kg/kW for chillers and direct evaporation systems, respectively, by evaluating particular systems and the literature (Wobst et al. 2003). The refrigerant amount of direct evaporation systems depends strongly upon the type of evaporator and can be significantly larger.

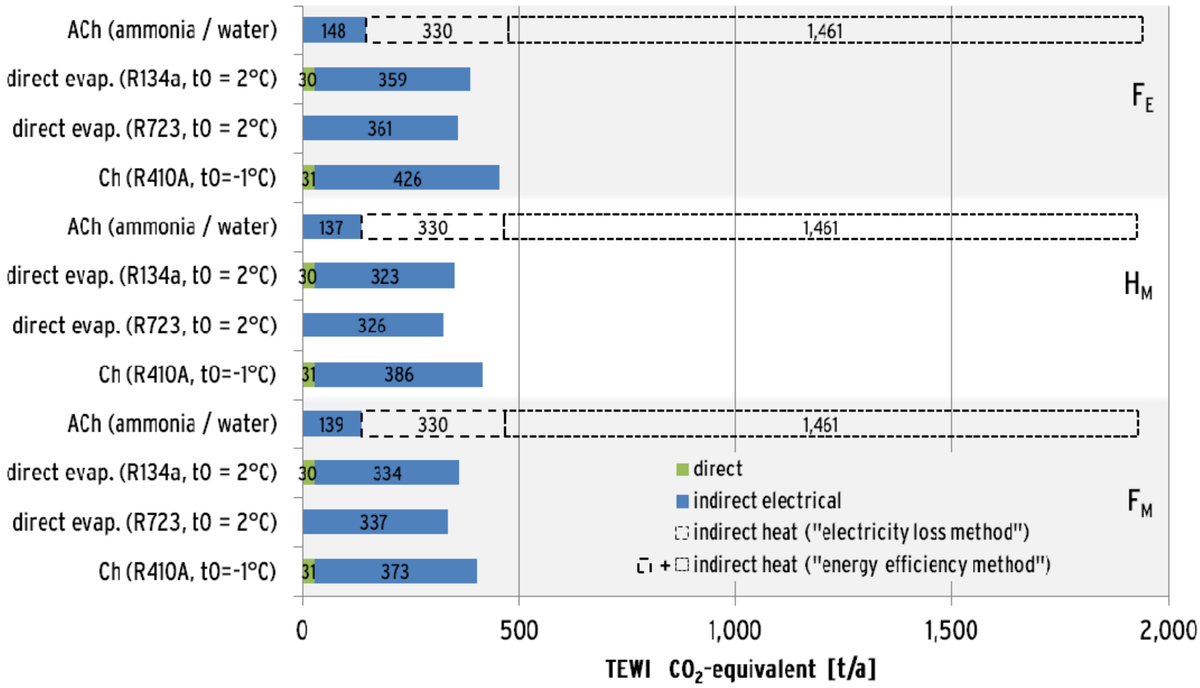


Figure 100: TEWI of various industrial refrigeration systems and different climate scenarios

With the assumptions stated above, the direct evaporation system with R723 exhibits the smallest TEWI. Utilising the fluorinated R134a in such systems has advantages over the indirect systems. However, these results depend strongly on the particular application. Major factors deteriorating the TEWI of direct evaporation systems are:

- Larger leakage rate compared to indirect systems, due to:
    1. Extensive refrigerant piping at the site
    2. Larger refrigerant charge
- Pressure drop along the suction pipes with deteriorated energy efficiency due to larger pressure ratio within the compressor
- Required power control even for lowest part-load due to missing cold storage

The heat demand of an absorption type refrigeration system leads to the largest TEWI emissions if it is covered by CHP plants and evaluated with the electricity loss or energy efficiency method. Considering the displacement mixture of natural-gas-fired CHP (see Figure 22, p. 36) or utilising waste heat, the application of ACh leads to the lowest TEWI emissions. Compared to compression type refrigeration systems, the climate protection potential of absorption type refrigeration systems could be demonstrated: the emissions are approx. 40 % below those of direct evaporation systems with R723, and approx. 33 % below those of chillers.

7.4.3.3 Costs of refrigeration systems

Process refrigeration has to be provided year-round, yielding a large degree of utilisation and, hence, an inverted ratio of investment to operation costs, compared to building air conditioning systems. It can be observed in Figure 101 that the operation costs dominate other costs. Therefore, the variation of the specific costs for electricity, water, and heat has a considerable influence on the total costs (indicated by error bars in Figure 101). Maintenance and disposal costs have almost no influence on the cost effectiveness of the systems.

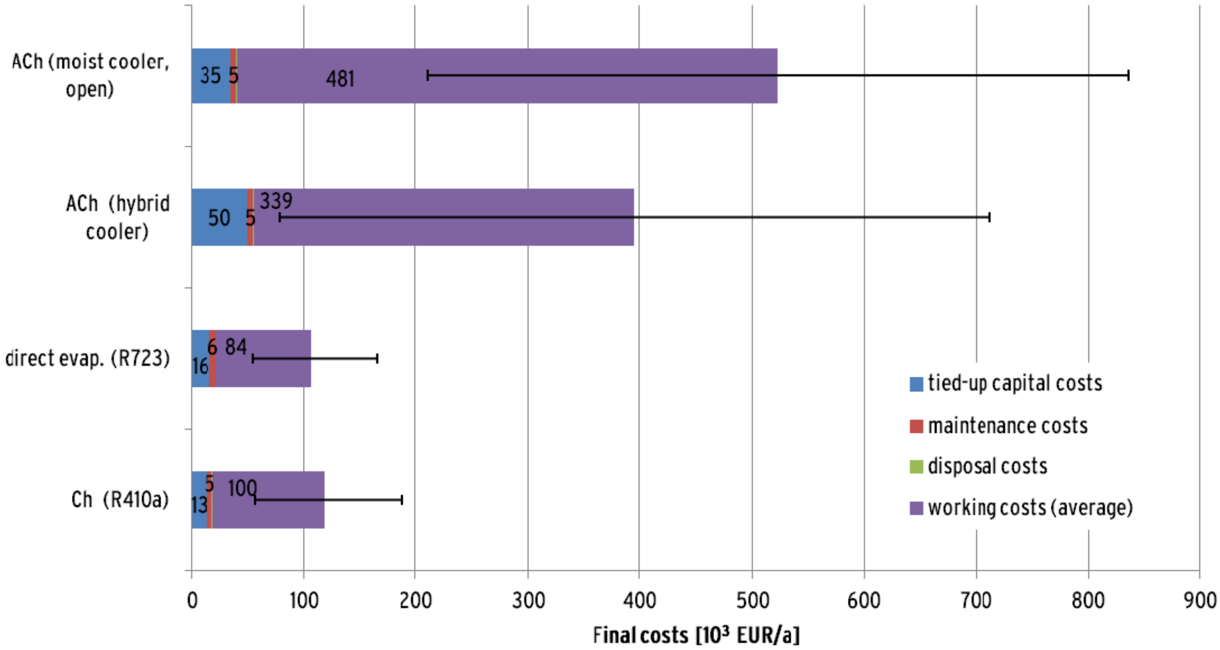


Figure 101: Total annual costs of various industrial refrigeration systems – operation costs are illustrated with their average value for all costs and climate scenarios; the variance of the operation costs – due to different cost scenarios – is indicated by error bars.

Although the investment costs have only a small share of the total costs, they are quite different for the various refrigeration systems (see Figure 102). As already observed for building air conditioning, the cooling generator of ACh systems provokes approx. twice the investment costs of CRP systems. The evaluation considers also two different re-cooling systems (hybrid and open wet cooling tower). The investment costs of the wet cooling tower are considerably smaller. Due to the larger water utilisation and the corresponding larger operation costs, a disadvantageous total costs balance arises.

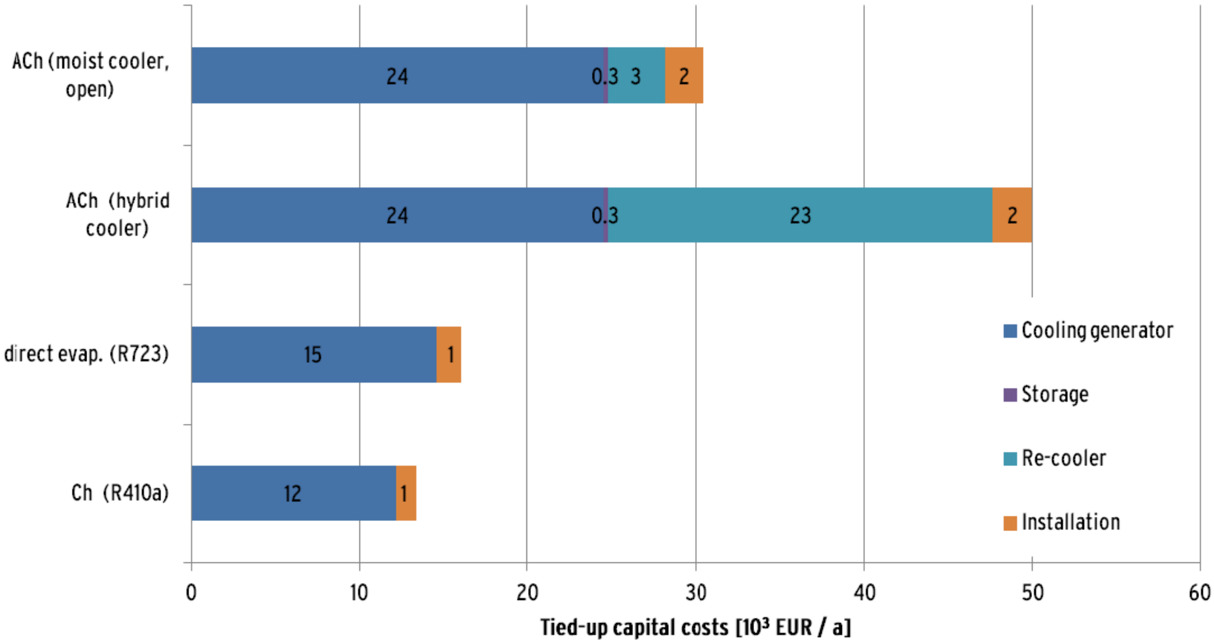


Figure 102: Composition of the investment costs of various refrigeration systems

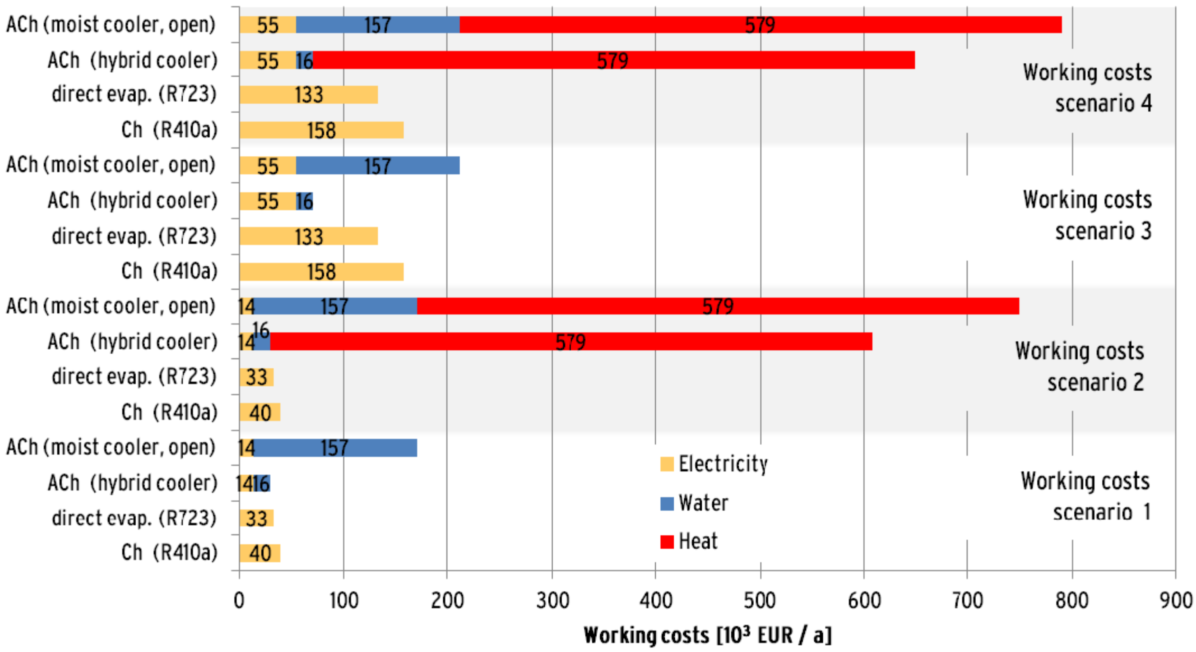


Figure 103: Composition of the operation costs for various cost scenarios (climate: FM)

illustrates the composition of the major operation cost components. At large heat supply costs (scenarios 2 and 4; see Table 16, p. 89), the ACh system is less cost efficient than comparable compression type systems. Otherwise, low or no heat supply costs (scenarios 1 and 3; see Table 16, p. 89) allow a more cost efficient operation. However, this demands for a hybrid and water-saving re-cooler. The larger investment costs of such a system are compensated by its lower operation costs.

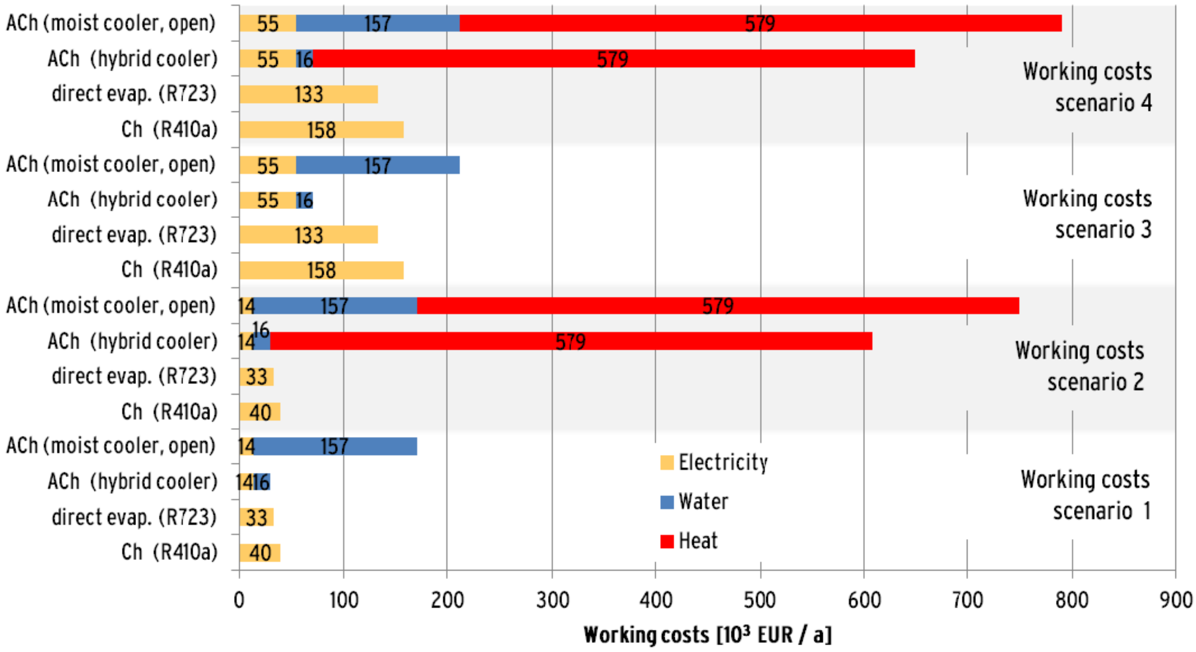


Figure 103: Composition of the operation costs for various cost scenarios (climate: FM)

7.4.4 Discussion of particular systems

7.4.4.1 Variant 5: Ch – Chiller

As already demonstrated in the evaluation of building air conditioning systems, the direct and indirect TEWI values of a refrigeration system depend strongly on the refrigerant. It has been emphasised that the degree of utilisation of the refrigeration system influences shares of direct and indirect TEWI strongly. Both observations can be made also with industrial refrigeration (see Figure 104). The indirect TEWI is relatively large due to the large degree of utilisation of the system (operation year-round). Depending on the refrigerant, the direct TEWI share is below 10 % (compared to approx. 40 % for building air conditioning; see Figure 81). Regarding the avoidance of greenhouse gas emissions, systems with a large degree of utilisation have to be carefully designed and their components chosen.

However, a direct TEWI of 10 % cannot be neglected. In our particular case (500 kW cooling power year-round), this corresponds to an emission of 30 t/a of CO_2 equivalents. A simple comparison reveals that this value corresponds to the emissions of the 51.5 MWh electrical energy production (specific CO_2 emissions for electricity production: 583 g/kWh). The utilisation of refrigerants with low GWP values and the reduction of leakage rates yield considerable savings of direct greenhouse gas emissions for industrial refrigeration.

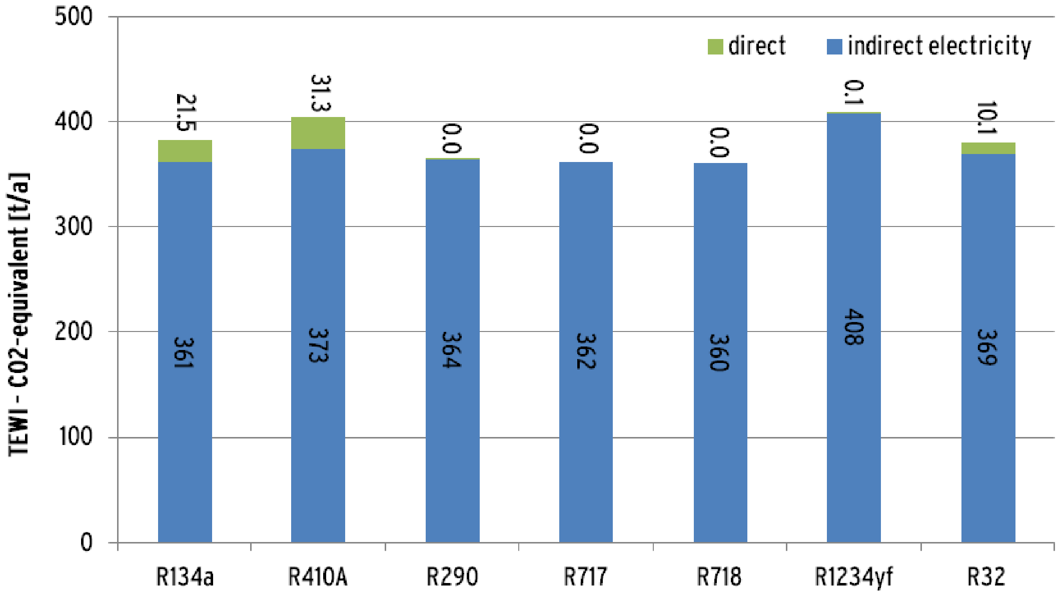


Figure 104: TEWI evaluations for a chiller with various refrigerants (climate: F_M)

7.4.4.2 Variant 6: ACh – absorption type refrigeration system operated by waste heat of a CHP plant

The large heat demand has been already discussed earlier (see subsection 7.4.3.1). The reason is the efficiency characteristics of a *single-effect* absorption type refrigeration plant. The heat ratio of cooling and heat power increases steeply with increasing driving temperature difference (F. Storkenmaier et al. 1999). The gradient of the curve becomes smaller close to the nominal operating point and finally approaches a limit (see Figure 105).

Figure 106 illustrates the heat ratio versus the cooling water supply temperature (at constant chilled and hot water supply temperatures). Below a cooling water supply temperature of 30 °C, the heat ratio increases marginally only, whereas the efficiency of compression type refrigeration systems increases linearly with decreasing ambient temperatures. The limiting factor is the minimal pressure for the expansion valve.

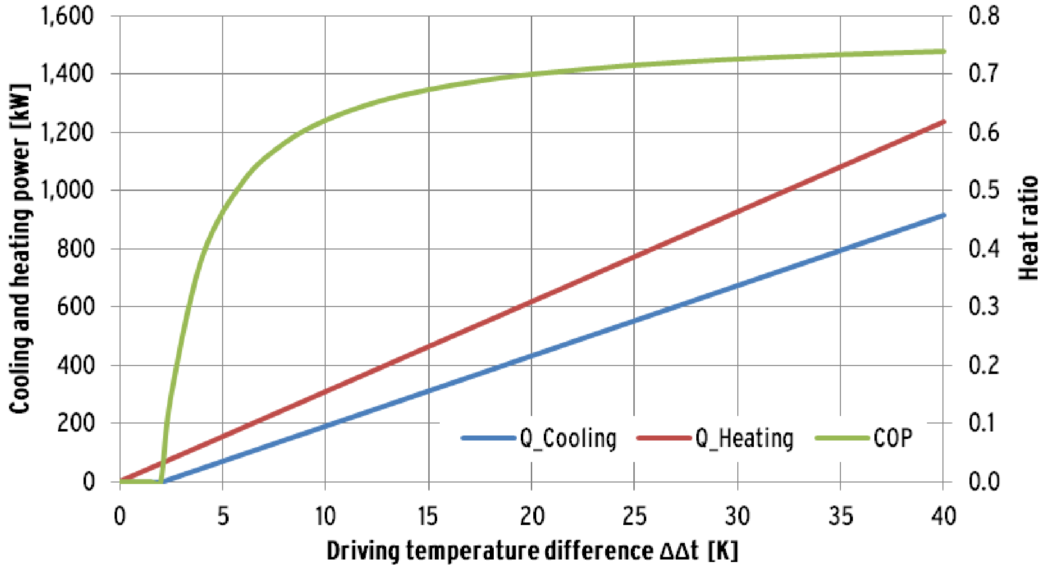


Figure 105: Schematic progress of the cooling and heating power and the heat ratio versus the total driving temperature difference

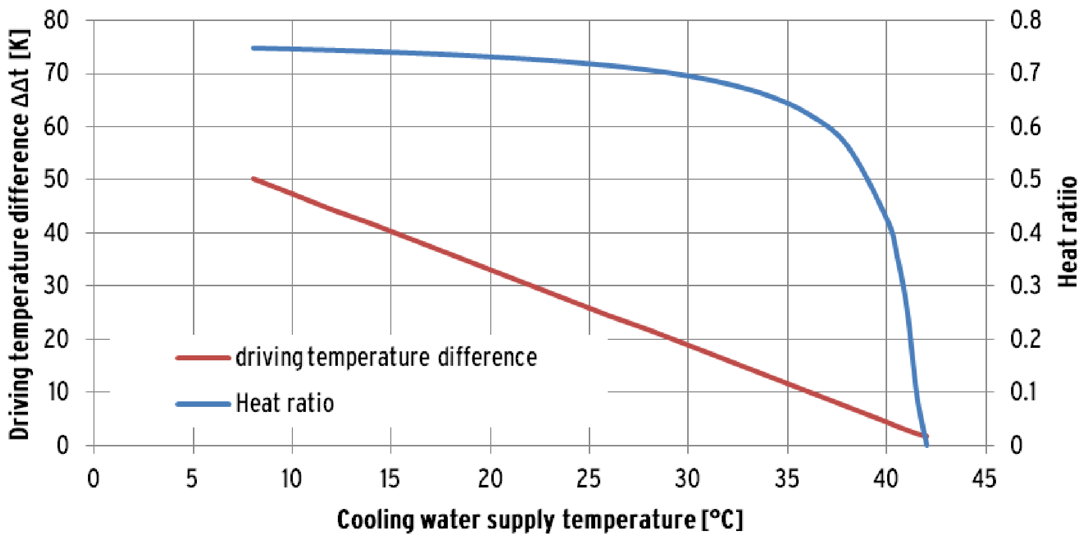


Figure 106: Schematic progress of the driving temperature difference and the heat ratio, using the example of a water/LiBr absorption type refrigeration plant ($t_{ChW,\,in}$ = 4 °C, $t_{HW,in}$ = 90 °C)

The availability of heat at higher temperatures allows reducing the heat demand of a water/LiBr *multi-effect* absorption type refrigeration system by employing a re-absorber instead of an evaporator (allowing evaporator temperatures below 0 °C). Such systems have heat ratios up to 1.1 as compared to common industrial absorption type refrigeration systems with NH_3/H_2O (0.6). The heat demand can be cut almost into halves.

7.5 Conclusions

7.5.1 Building air conditioning

The small number of full-load hours (e.g., approx. 460 h for climate scenario F_M corresponding to a degree of utilisation of 5 %) of air conditioning systems in the climate regions under consideration leads to remarkable conditions regarding the direct and indirect TEWI emissions. Due to the low electrical energy demand (due to low number of full-load hours), the indirect share is very small compared to other refrigeration applications (e.g., commercial cooling).

From this, the conclusion is drawn that for such systems a refrigerant with a low TEWI and good system tightness is especially important. The application of natural refrigerants with small GWP values in chillers is possible and already offered by some manufacturers (e.g., Frigoteam). However, VRF systems are problematical. The inflammability of refrigerants with low GWP limits the acceptance in building applications. Non-inflammable, natural high-pressure refrigerants as an alternative for R410A are rare. The application of CO_2 as a refrigerant is possible as has been proved by a market-ready VRF system (DAIKIN 2008). However, the accumulation of CO_2 within a room due to leakages at the indoor unit is considered critical. Moreover, the energy efficiency ratio remains considerably below that for conventional systems (DAIKIN 2008).

Humidification, ventilation, and air conditioning systems (HVAC) with DEC systems reach small temperature differences between fresh and waste air only. The cooling demand has to be covered by large volume flow rates. In addition, the fresh air is dehumidified in a controlled manner, leading to a large energy demand. The DEC system might be in advantage if the application requires large air renewal rates and a controlled dehumidification. A combination with a CRP system, e.g., in conjunction with chilled ceilings, demand peaks can be covered better.

7.5.2 Industrial refrigeration

Similar to building air conditioning, the results depend strongly on the boundary conditions of the demand and the evaluation. If waste heat of a sufficient temperature level (> 85 °C) is available, electrical energy can be saved with ACh systems. Is it about heat which can be utilised otherwise for heating or processes, combinations are imaginable. The advantages of particular systems can be employed: utilisation of waste heat in the summer with an absorption type refrigeration system and application of a compression type refrigeration system with its larger efficiency in the winter.

A comparison of direct and indirect evaporation systems depends strongly on the technical boundary conditions. At convenient conditions, such as constant cooling loads and short distance between the locations of cooling generation and cooling demand, a direct evaporation system is in advantage from both energetic and ecological point of view. The application of natural refrigerants shall be preferred due to much lower direct TEWI emissions.

For the case of strongly varying cooling demands or long distances between location of cooling generation and consumption, indirect evaporation systems (chillers) are in advantage. Among those, a lower amount of refrigerant and smaller frictional losses between evaporator and compressor are advantageous. Especially, the opportunity to decouple the cooling generation and demand is remarkable, considering increasing renewable energy sources and control aspects. Indirect evaporation systems can be combined with absorption type refrigeration plants very good and various modes of operation are possible (single operation, parallel operation, etc.).

Beside the low GWP values, the efficiency is an advantage of natural refrigerants. Compared to R134a, the refrigerants R290, R717, and R718 require approximately the same electrical energy demand (relative differences below 1 %), but compared to R410A, they exhibit an efficiency improvement of 3 %. In contrast, R1234yf requires approx. 10 % *more* electrical energy.

7.5.3 General statements

The energetic evaluation of various techniques for cooling generation provides a differentiated image. For instance, direct evaporation systems are worse than chillers for building air conditioning, whereas they are in advantage in industrial applications. Absorption type refrigeration systems have a relatively low electrical, but a large thermal energy demand. For the case of unused waste heat sources, this characteristic is not a disadvantage.

The calculations revealed that general statements related to the climate friendliness of various refrigeration systems are difficult. A comparison is possible only for a particular case with its system parameters and boundary conditions. However, the following can be concluded:

- The avoidance of refrigeration systems shall have highest priority during the design of buildings and industrial plants.
- For the case of cooling demand, natural refrigerants shall be preferred over synthetical ones.
- Unsophisticated system designs might lead to large TEWI values due to small energy efficiency ratios. For systems with a large degree of utilisation (large number of full-load hours), the design might have a larger influence on the total TEWI than the refrigerant.
- Contrary, in systems with a small number of full-load hours: it cannot be suggested that good energy efficiency leads to a good climate-friendliness of a particular system. In the case of CRP systems, the leakage rate has a considerable influence on the TEWI when refrigerants with a high GWP are used. Here, an increase of the efficiency of the process itself (e.g., with a chilled water storage for building air conditioning) has only limited influence. Hence, the retrieval and reduction of leakages has a large significance (especially, limits and their review in practice)
- The utilisation of unused waste heat can decrease the TEWI significantly by use of an absorption type chiller.

8 Substitution of refrigeration systems by heat-driven systems (scenarios)

The analyses in Chapter 7 have demonstrated that the application of absorption type chillers utilising waste heat can lead to TEWI emission reduction. Ab- and adsorption type refrigeration systems require mostly thermal energy for the cooling generation, yielding a lower electrical energy demand. The application of waste heat or ambient energies allows for the reduction of the primary energy demand for cooling generation. Primary energy savings and reductions of the CO_2 emissions are possible with the utilisation of heat from combined heat and power plants (CHP). The same is true for natural-gas-fired gas-steam power plants due to the displacement electricity mixture. However, for coal power stations with heat extraction it has to be investigated for a particular case (see subsection 4.2.10). Since CHP plants are usually limited by deficient heat demand, the absorption type refrigeration technology can improve the economical efficiency of such systems.

The substitution of compression type refrigeration systems by thermally driven systems is investigated within this chapter. The objective is, to provide an estimation about which shares of the cooling demand in various industrial sectors can be covered by ad- and absorption type refrigeration systems. Basis for this are the data of cooling demand and its characteristics provided in Chapter 5. Beside the data for the cooling demand, those for waste heat potentials from both industrial processes and CHP systems are relevant. Some of the data have been presented in Chapter 5 already and is extended in the present chapter.

The variations with different boundary conditions are investigated for the potential estimation:

- Utilisation of existing waste heat source for heat supply
- Utilisation of already existing or upgraded natural gas-fired CHP plants

In an ideal case, it is possible to employ unused internal waste heat at sufficient temperature level or heat from a solar thermal system for covering the cooling demand. Compared to compression type refrigeration systems, thermally driven refrigeration systems exhibit larger investment costs. Therefore, sufficiently large degrees of utilisation are required for the financial amortisation. Beside the investment costs for the absorption type refrigeration plant, further costs arise during operation, e.g., complex exhaust gas heat exchangers, cold or heat storages.

An example shall illustrate this issue. Among the dairy farms, there are many farms converting biomass into biogas, which is converted into electricity in a CHP later on. However, the cost effectiveness of such a system is reached only at large full-load hours and sufficient heat cosnumption. The problem of most of those systems is the lack of heat demand during the summer, whilst there is a cooling demand for, e.g., rapid cooling (with an explicit peak load) and storage of the milk. The designing of an absorption type refrigeration system for the rapid cooling leads to a very large system with a low degree of utilisation. Such a system is not economical at all. Moreover, it does not lead to an increased degree of utilisation of the biogas-fired CHP. Furthermore, a sufficiently large heat quantity cannot be provided by the CHP. An alternative for the present application is a small absorption type refrigeration system in conjunction with a cold storage. By these measures, it is possible to adjust the cooling distribution on the cooling demand and the cooling generation on the heat supply. However, the investment costs for the cold storage have to be considered for the economical investigation.

8.1 Potential of the waste heat utilisation for cooling generation

The temperature level of the waste heat, the cooling demand, and the temporal characteristics of both the heat source and the cooling demand are important, if waste heat is to be utilised for cooling generation. The analysis of waste heat sources in different industrial sectors requires some assumptions, which are applied on the case studies from the literature:

- Exhaust gas temperatures of furnaces (e.g., melting or annealing furnaces) are assumed to be below 300 °C. Often, the exhaust gas temperatures are higher, however, efficient burner technologies are available today with efficiency ratios up to 85 % (MIOBA 2012). So-called regenerative burners allow cooling the exhaust gases below 300 °C by means of pair-wise and discontinuous operation. Increasing the efficiency of the furnaces has, however, a larger priority than a subsequent utilisation of the exhaust gases.
- Various production processes are employed in mechanical engineering. Those of the processes offering heat sources are, e.g., techniques for metal treatment (welding, soldering, hot forging) and surface treatment (cleaning baths, electroplating, varnishing). Due to the vast number of small and medium-sized enterprises, these processes are utilised in very different manner and frequency. Hence, generalised descriptions are of limited suitability. Consequently, the degree of realisation is considered low.

The majority of the references have been already mentioned in the determination and discussion of the cooling and energy demand in subsections 5.4, 5.5, and 5.6. The case studies are presented in the so-called “Abwärmeatlas” (waste heat atlas) for the state of Saxony, which has been provided by a project of the Saxon energy agency (SEANA). The atlas provides a map including the location, kind of heat source, heat quantity, and temperatures of heat sources (SAENA 2010).

With a similar background, the Fraunhofer UMSICHT Institute investigates the implementation of a waste heat network-system for industrial parks. Objective of this project is to set up such a network system and its durable and economical operation. Hence, the waste heat cannot be utilised by the producer only, but also by all other stakeholder in the network (Frauenhofer-UMSICHT 2013).

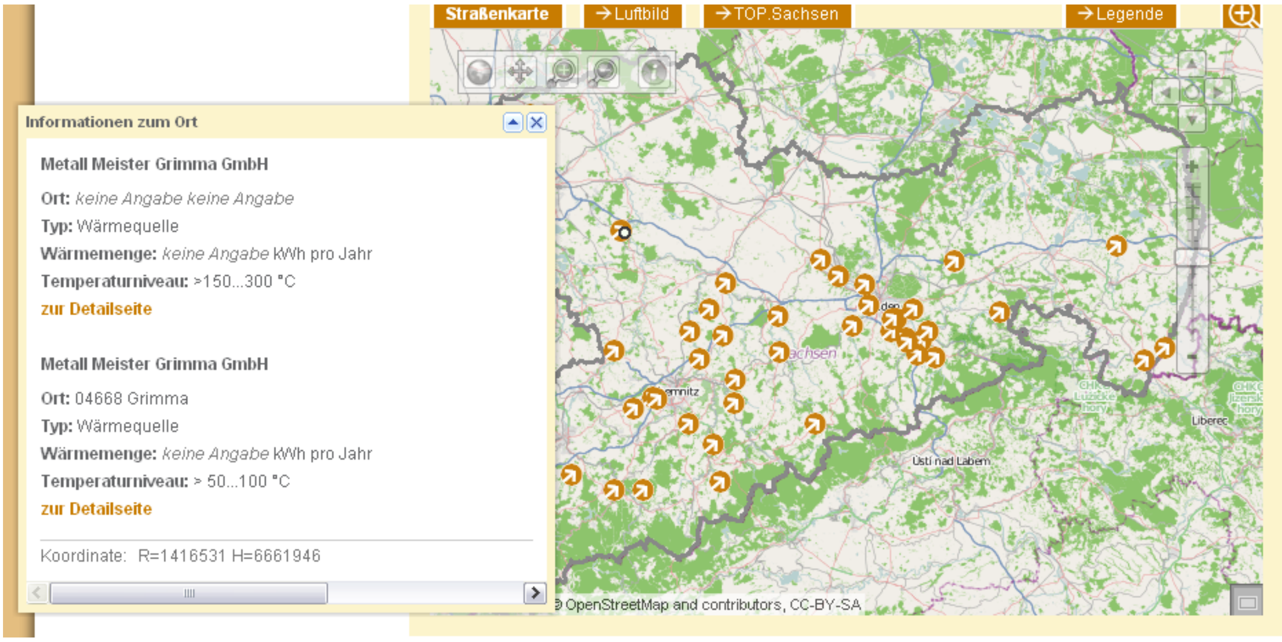


Figure 107: Detail of the SAENA GmbH waste heat atlas (right-hand side: map with locations, left-hand side: database entry for a particular entry) (SAENA 2010) www.energieportal-sachsen.de

Table 26 provides the cooling demand and waste heat offer of various industrial sectors, comprising the temporal variations and the resulting necessity of thermal storages. The required temperatures for cooling generation and the available waste heat temperatures are compared as well as the situation of the cooling demand related to the waste heat potential. Furthermore, an assessment of the required investment effort is provided. From those numbers it is possible to estimate the share of thermally driven cooling generation of the total cooling demand. The opportunities of absorption-compression cascades are considered. Moreover, Table 26 demonstrates the potential of heat-driven refrigeration systems in different industrial sectors. Different enterprise sizes and processes at various locations are also considered. The product of the percentagewise shares on the cooling demand and the opportunities for realisation is the obtainable coverage of the cooling demand by a heat-driven refrigeration systems. This value is offset against the already existing share. From these results, the additional cooling demand covering potential and the prospective electrical energy savings due to ad- and absorption type refrigeration systems can be obtained.

The estimation of the shares is carried out choosing a conservative scenario. Criteria for exclusion of heat-driven refrigeration systems are not well-engineered or not available techniques and large investment costs. The inclusion of prospective cold storage systems will increase the variable gross margin for heat-driven cooling generation.

Table 26: Evaluation of the cooling demand and waste heat source – estimation of the saved electrical energy demand by utilisation of heat-driven refrigeration systems

Industrial sector			Cooling demand		Waste heat				Evaluation and estimation					Results		
Major group	Minor group	Sub-group	Cold temperature [°C]	Cooling demand characteristics (seasonal/daily)	Waste heat sources	Ratio waste heat/cold	Waste heat temperature [°C]	Waste heat characteristics (seasonal/daily)	Requires storage	Estimated share of cooling demand	Investment costs effort	Possible degree of realisation	Product share x degree of realisation	Resulting additional cooling demand covered by ACh [GWh/a]	Share of the current CRP covered cooling demand	saving potential final electrical energy [GWh/a]
Industrial refrigeration	Mining, exploitation of pit and quarry (only stone coal)		3	constant/ constant	Machines, firedamp	<<	0	constant/ constant	no	0%	-	0%	0%	0	0%	0
	Paper and pulp industry		6	fluctuating/ constant	Machines	>	50…100	constant/ constant	no	30%	medium	30%	9%	23	9%	8
	Printing industry		20	fluctuating/ constant	Machines, paper rolls, dryer (very large cooling temperatures possible; possible extraction of large-temperature heat), after-burner heat	>	50…200	constant/ constant	no	50%	medium	30%	15%	63	10%	5
	Other chemicals industry without air and gas liquefaction		-4	constant/ constant	After-burning processes, reaction enthalpies, exhaust gases	>	50…200	constant/ constant	no	30%	low	70%	21%	411	8%	152
	Other chemicals industry only air and gas liquefaction		-190	-	Compressor waste heat, enthalpy of compressed air	>	50…500	constant/ constant	no	0%	large	0%	0%	0	0%	0
	Pharmaceutical industry		6	constant/ constant	Reaction enthalpies, exhaust gases	>	50…200	constant/ constant	no	30%	low	50%	15%	54	6%	14
	Plastics and rubber industry		6	constant/cyclic	Exhaust gas enthalpy from heat generators, heat of plastics products and machines (>200 °C difficult to utilise)	<=	50…100	constant/ cyclic	yes	20%	large	10%	2%	56	2%	28
	Building and building materials industry		-25	fluctuating/ constant	Motor waste heat, exhaust gases of combustion machines and de-centralised electricity production	<	50…100	constant/ constant	no	0%	large	0%	0%	0	0%	0
	Electric and electronics industry (also conductor board production, soldering, and semiconductor production)		5	fluctuating/ constant	Waste heat of self-produced electricity (especially semi-conductor production	<=	50…100	constant/ constant	no	80%	low	80%	64%	23	7%	6

Industrial sector			Cooling demand		Waste heat				Evaluation and estimation					Results		
Major group	Minor group	Sub-group	Cold temperature [°C]	Cooling demand characteristics (seasonal/daily)	Waste heat sources	Ratio waste heat/cold	Waste heat temperature [°C]	Waste heat characteristics (seasonal/ daily)	Requires storage	Estimated share of cooling demand	Investment costs effort	Possible degree of realisation	Product share x degree of realisation	Resulting additional cooling demand covered by ACh [GWh/a]	Share of the current CRP covered cooling demand	saving potential final electrical energy [GWh/a]
	Automotive industry		6	fluctuating/ constant	Test facilities, machine waste heat	<	<50	constant/ fluctuating	yes	20%	large	30%	6%	22	1%	5
	Mechanical engineering		6	fluctuating/ constant	Test facilities, machine waste heat	<	<50	constant/ fluctuating	yes	10%	large	30%	3%	30	3%	8
	Compressors, e.g., for compressed air generation, air desiccation (with refrigeration system)		3	constant/ constant	Compressor waste heat, enthalpy of compressed air	<=	50...100	constant/ constant	no	30%	medium until large	30%	9%	34	9%	8
	Process refrigeration (water chillers) for industrial applications		6	fluctuating/ constant	Depending on industrial sector	?	50...200	?	0	10%	?	30%	3%	-5	0%	-1
	Control cabinet cooling		20	constant/cyclic	Heat of components (>70°C)	<=	50...100	fluctuating/ constant	no	20%	large	10%	2%	8	2%	6
		> 50 employees, normal cooling	0	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	208	4%	65
		> 50 employees, freezing	-25	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	139	4%	93
		20...49 employees, normal cooling	0	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	52	4%	20
		20...49 employees, freezing	-25	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	35	4%	29
		Other normal cooling	0	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	30	4%	12
Food production	Food industry	Other freezing	-25	constant/cyclic	Exhaust gases of heat generation, plume (pastries enterprises)	<=	50...200	constant/ fluctuating	yes	30%	large	20%	6%	20	4%	18

Industrial sector			Cooling demand		Waste heat				Evaluation and estimation					Results		
Major group	Minor group	Sub-group	Cold temperature [°C]	Cooling demand characteristics (seasonal/daily)	Waste heat sources	Ratio waste heat/cold	Waste heat temperature [°C]	Waste heat characteristics (seasonal/ daily)	Requires storage	Estimated share of cooling demand	Investment costs effort	Possible degree of realisation	Product share x degree of realisation	Resulting additional cooling demand covered by ACh [GWh/a]	Share of the current CRP covered cooling demand	saving potential final electrical energy [GWh/a]
	Breweries		-7	constant/cyclic	Waste heat of fermentation process	<=	50	constant/ fluctuating	yes	30%	large	50%	15%	108	15%	41
	Dairy farms		4	constant/cyclic	Waste heat of biogas combustion	>	50...200	constant/ constant	yes	100%	large	20%	20%	210	20%	117
Building air conditioning	Commercial, trade, service		6	fluctuating/ fluctuating	Solar heat, district heat	<	50...100		possible	50%	medium	20%	10%	508	6%	175
	Industry		6	fluctuating/ fluctuating	See various industrial sectors	<=	50...200		possible	50%	medium	20%	10%	1644	9%	567
	Data centres, servers		6	fluctuating/ fluctuating	Depending on location	<	50...100		possible	20%	medium	10%	2%	67	2%	23
	Domestic		6	fluctuating/ fluctuating	Solar heat, district heat	<	50...100		possible	100%	large	5%	5%	36	5%	13
Sum														3776		1412

Figure 108 illustrates the cooling demand of different industrial sectors. Moreover, the cooling demands, which are already covered by ACh and those that can be covered by ACh utilising waste heat potentials are presented (conservative scenario). Figure 109 provides an overview of the corresponding electrical energy savings.

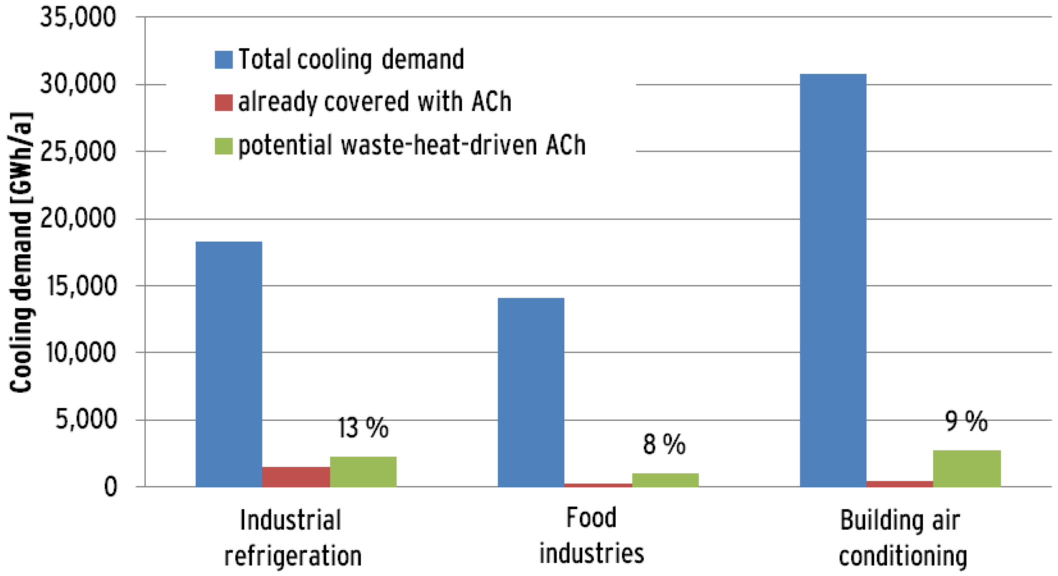


Figure 108: Cooling demand and the present and prospective shares of waste-heat-driven refrigeration technology – variant: utilisation of existing waste heat for cooling generation

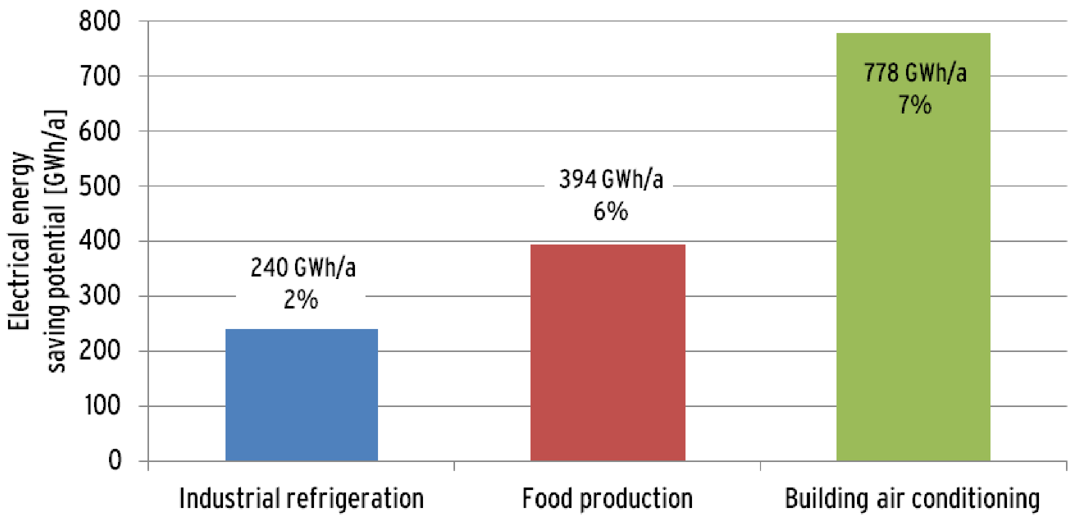


Figure 109: Obtainable savings of electrical energy in different industrial sectors (percentage-wise values based on the electrical energy demand for cooling generation in the respective sector) – variant: utilisation of existing waste heat for cooling generation; auxiliary power of ACh not considered

8.2 Potential of natural gas-driven combined heat and power plants

The application of natural gas-driven combined heat and power plants (CHP) leads to CO_2 saving whilst electricity production due to regulations regarding the feed-in prioritisation (EEG and KWKG) and so-called Merit-Order effect (see subsection 4.2.10, p. 34). The utilisation of the incidental waste heat is obligatory for claiming additional allowances ("KWK-Bonus") and feed-in priorities (KWKG 2009).

Combined heat and power plants offer two waste heat shares. Firstly, the motor heat, extracted via a cooling circuit, has the largest share with 60 to 80 %. Its temperature level is between 80 and 90 °C. Secondly, the high temperature (approx. 500 °C) exhaust gas stream has a share of approx. 20 to 40 % (see Figure 110). There are two different modes of utilisation for absorption type processes:

- Feed-in of the total waste heat at a single temperature level in a *single-effect* ACh → lower investment costs in conjunction with a lower heat ratio
- Separated feed-in of low and high temperature waste heat in a *multi-effect* ACh → larger investment costs in conjunction with a larger heat ratio

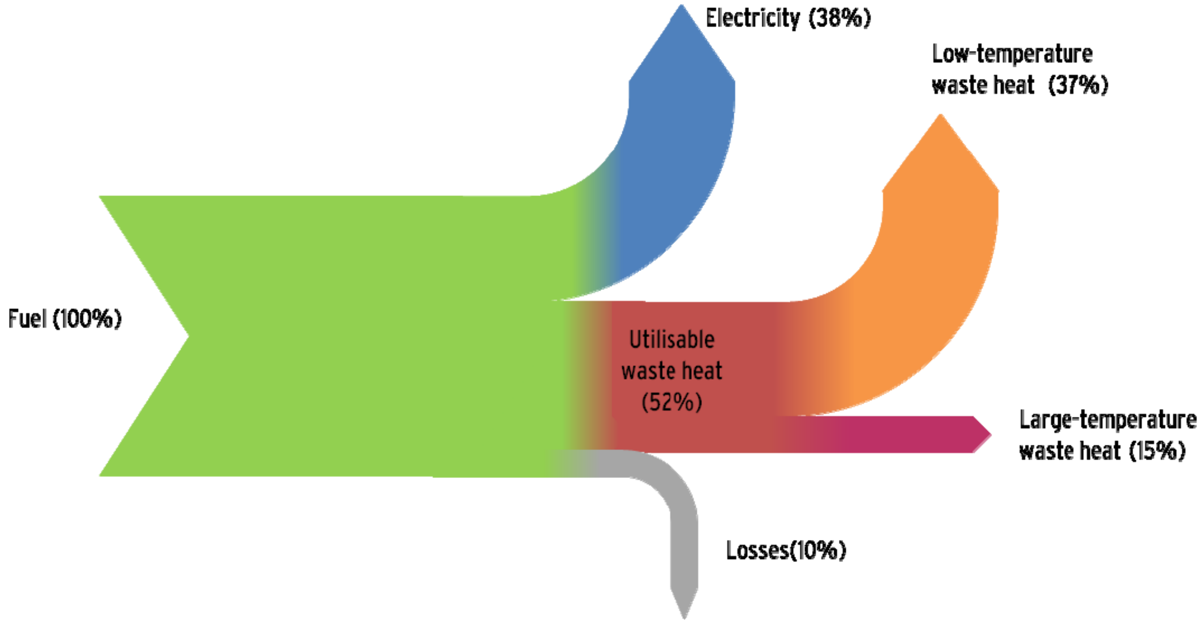


Figure 110: Typical energy flow chart of a CHP

The feed-in tariff of the generated electricity consists of three components:

- Base load price: bargained with the grid operator or average value of the base load prices of the most recent quarter (connection, taking, and remuneration obligation according to KWKG) at the electricity stock exchange EEX
- CHP add-on, depending on CHP power, proof of heat utilisation required
- Reimbursement of avoided grid utilisation fees (RAGUF)

For the case of internal electricity utilisation, the CHP add-on and the RAGUF remain; however, the base load price is not paid. The saving of the CHP operator depends on the electricity supply tariff. It shall be considered that the prices for electricity rise due to decreasing external demand at (partial) self-supply.

The amortisation of a CHP installation is coupled to a large number of full-load hours in conjunction with a waste heat utilisation (consumption of the CHP bonus). Further heat consumers are necessary for distinctive seasonal cooling demands, e.g., heating demand during the winter or process heat demand. It is to be stressed that CHP plants cover a large share of the heat demand in the industry already today (see Figure 111). On the one hand, this might be a criterion for exclusion, but on the other hand might lead to longer operation hours of already existing CHP plants. The latter one is particularly the case if the heat demand is seasonally asynchronous to the cooling demand.

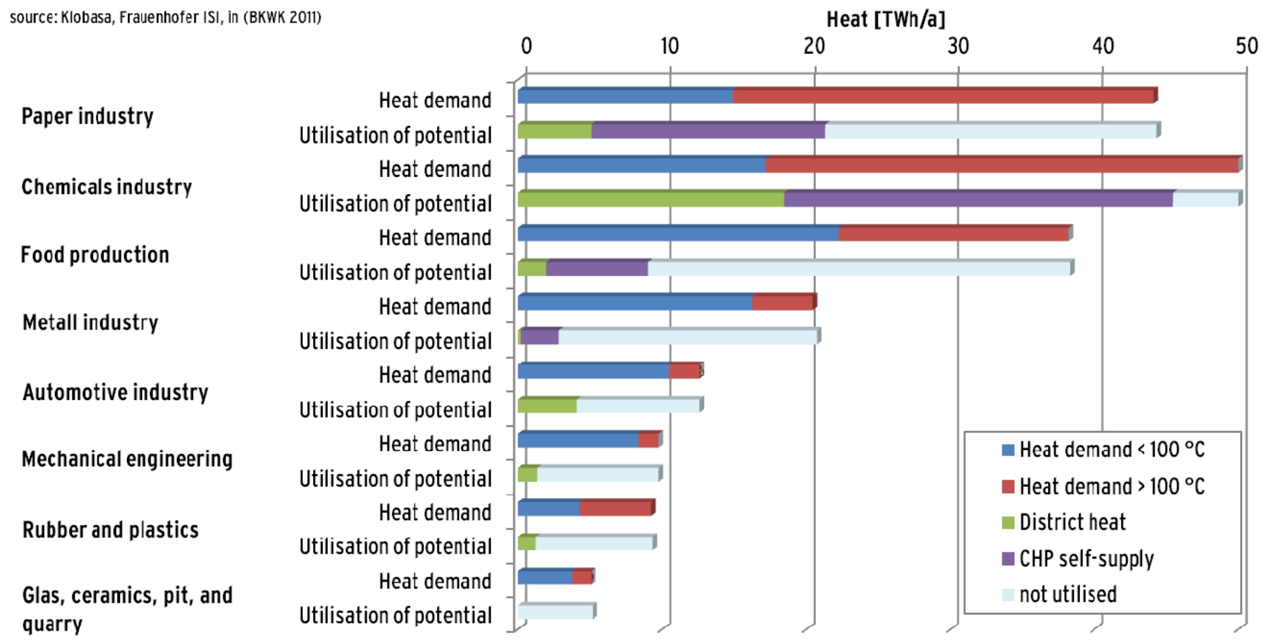


Figure 111: Current coverage by self-supply CHP plants or district heat, including not fully bailed out potentials (- BKWK 2011)

The following factors are considered for parameter estimation:

- Temperature level of the cooling demand
    1. Temperatures for deep-freezing usually demand for a cascaded system for which absorption-compression cascades are useful solutions
    2. Higher cooling temperatures lead to smaller temperature differences in the ACh, however the heat ratio does not improve significantly. In contrast, CRP systems are the more efficient the smaller the temperature difference is.
- Progression of the cooling demand – compensation by other heat demand
- Electricity demand
- Size of the enterprise: contrary to CRP, ACh are available for larger powers only (see subsection 6.3.1.3 “Market overview of ab- and adsorption type refrigeration plants“, p. 80ff. and Figure 60). The same holds for CHP.

Table 27 provides an evaluation of different industrial sectors in relation to the combined heat, power, and cooling (CHPC) installations and lists possible electrical energy savings. The evaluation is difficult, since most of the industrial sectors have a large variety of processes. A further factor of uncertainty is the current coverage of CHP systems in the particular sectors. A brochure of the inter-trade organisation (- BKWK 2011) provides an overview of utilised and to be utilised potentials of CHP in industry. It shall be mentioned that the itemisation into industrial sectors in the present study differs from that in the brochure.

Table 27: Assessment of the cooling demand in conjunction with prospective CHPC plants – estimation of the saved electricity demand by utilisation of heat-driven refrigeration systems

Industrial sector			Cooling demand			Waste heat		Assessment and estimation				Results		
Major group	Minor group	Sub-group	Required cooling temperature [°C]	Cooling demand characteristics (seasonal/ daily)	electricity demand to cooling demand	Assessment (textual)	Assessment (-- until ++)	Requires storage	Estimation of coverable share of object	Possible degree of realisation	Product of coverable share x degree of realisation	Additional cooling demand covered by ACh [GWh/a]	Share of present CRP demand	Saving potential final electrical energy [GWh/a]
Industrial refrigeration	Mining, exploitation of pit and quarry (only stone coal)		3	constant/ constant	14	Constant cooling demand, large electricity demand	++	no	100%	80%	80%	448	80%	128
	Paper and pulp industry		6	varying/ constant	248	Large CHP share already today, integration into existing systems intelligible	+	no	100%	60%	60%	151	60%	54
	Printing industry		20	varying/ constant	20	Requires large cooling water supply temperatures, lots of waste heat available, utilisation at large ambient temperatures only	o	no	50%	10%	5%	0	0%	0
	Other chemicals industry without air and gas liquefaction		-4	constant/ constant	19	Large waste heat amounts available, ACh already utilised, already large CHP share	o/+	no	60%	50%	30%	885	17%	328
	Other chemicals industry only air and gas liquefaction		-190	-	1	Refrigerant is cooling application (absorption as intermediate step only -> process optimisation)	--	no	0%	0%	0%	0	0%	0
	Pharmaceutical industry		6	constant/ constant	0	Relatively constant cooling demand, other waste heat utilisation imaginable	+	no	80%	75%	60%	438	51%	110
	Plastics and rubber industry		6	constant/ cyclic	10	Seasonally constant cooling loads, large electrical energy demand	+	yes	80%	75%	60%	1,692	60%	846
	Building and building material industry		-25	Varying/ constant	14567	Cooling of concrete can reduce the cement share in a concrete factor (process optimisation)	++	no	100%	80%	80%	1	80%	0
Industrial refrigeration	Electric and electronics industry (also conductor board production, soldering, and semiconductor production)		5	varying/ constant	0	Large share of CHPC already today, own electricity production common for security of supply	+	no	80%	90%	72%	49	15%	12
	Automotive industry		6	varying/ constant	53	Large electricity demand, relatively constant cooling demand	+	yes	60%	90%	54%	732	49%	183

Industrial sector			Cooling demand			Waste heat		Assessment and estimation				Results		
Major group	Minor group	Sub-group	Required cooling temperature [°C]	Cooling demand characteristics (seasonal/ daily)	electricity demand to cooling demand	Assessment (textual)	Assessment (-- until ++)	Requires storage	Estimation of coverable share of object	Possible degree of realisation	Product of coverable share x degree of realisation	Additional cooling demand covered by ACh [GWh/a]	Share of present CRP demand	Saving potential final electrical energy [GWh/a]
	Mechanical engineering		6	varying/ constant	47	Large electricity demand, relatively constant cooling demand	+	yes	60%	90%	54%	540	54%	135
	Compressors, e.g., for compressed air generation, air desiccation (with refrigeration system)		3	constant/ constant	149	Large electricity demand, relatively constant cooling demand, absolute powers in decentralised system usually too small for ACh	++	no	40%	80%	32%	120	32%	30
	Process refrigeration (water chillers) for industrial applications		6	varying/ constant	28	Power demand system usually too small for ACh, but large electricity demand	o	possibly	80%	24%	3%	365	21%	104
	Control cabinet cooling		20	constant/ cyclic	0	Required cooling water supply temperatures too large for beneficial ACh operation	-	no	0%	0%	0%	0	0%	0
Food production	Food industry	> 50 employees, normal cooling	0	constant/ cyclic	4	Seasonally constant cooling loads, enterprises might be too small for useful ACh operation	+	partially	50%	40%	6%	1,979	38%	618
		> 50 employees, freezing	-25	constant/ cyclic	3	Seasonally constant cooling loads, absorption-compression cascades useful	o	partially	50%	25%	6%	799	23%	532
		20…49 employees, normal cooling	0	constant/ cyclic	3	Seasonally constant cooling loads, enterprises might be too small for useful ACh operation	+	yes	50%	30%	6%	363	28%	140
		20…49 employees, freezing	-25	constant/ cyclic	2	Seasonally constant cooling loads, absorption-compression cascades useful	o	yes	50%	15%	6%	112	13%	94
		Other normal cooling	0	constant/ cyclic	1	Seasonally constant cooling loads, enterprises might be too small for useful ACh operation	+	yes	50%	30%	6%	208	28%	83
		Otherfreezing	-25	constant/ cyclic	1	Seasonally constant cooling loads, absorption-compression cascades useful	o	yes	50%	15%	6%	64	13%	58
Food produc tion	Breweries		-7	constant/ cyclic	4	There is already a heat demand at approx. 85 °C, constant cooling demand (ideal application)	++	yes	100%	80%	80%	575	80%	220

Sustainable cooling supply for building air conditioning and industry in Germany

Industrial sector			Cooling demand			Waste heat		Assessment and estimation				Results		
Major group	Minor group	Sub-group	Required cooling temperature [°C]	Cooling demand characteristics (seasonal/ daily)	electricity demand to cooling demand	Assessment (textual)	Assessment (-- until ++)	Requires storage	Estimation of coverable share of object	Possible degree of realisation	Product of coverable share x degree of realisation	Additional cooling demand covered by ACh [GWh/a]	Share of present CRP demand	Saving potential final electrical energy [GWh/a]
	Dairy farms		4	constant/ cyclic	0	CHP already discussed for waste heat, temporary cooling demand only, storage required, also barn cooling in winter, heating with biogas	+	yes	90%	20%	18%	189	18%	105
Building air conditioning	Commercial, trade, service		6	varying/ varying	47	Too short operation times, usually too small systems	-		30%	50%	15%	942	11%	325
	Industry		6	varying/ varying	33	Too short operation times for cold, possible if heat can be utilised otherwise	-	no	30%	10%	3%	415	2%	143
	Data centres, servers		6	varying/ varying	0	too short operation times, too large cooling water temperatures	-		0%	0%	0%	0	0%	0
	Domestic		6	varying/ varying	555	Too short operation times, usually too small systems	-		0%	0%	0%	0	0%	0
Sum												11,196		4,260

Figure 112 lists the cooling demand of different applications. Furthermore, the cooling demand, which is covered by ad- and absorption type chillers already today, and the share of waste heat utilisation coverage (pessimistic scenario), are presented. Moreover, Figure 113 illustrates the electrical energy savings of particular applications.

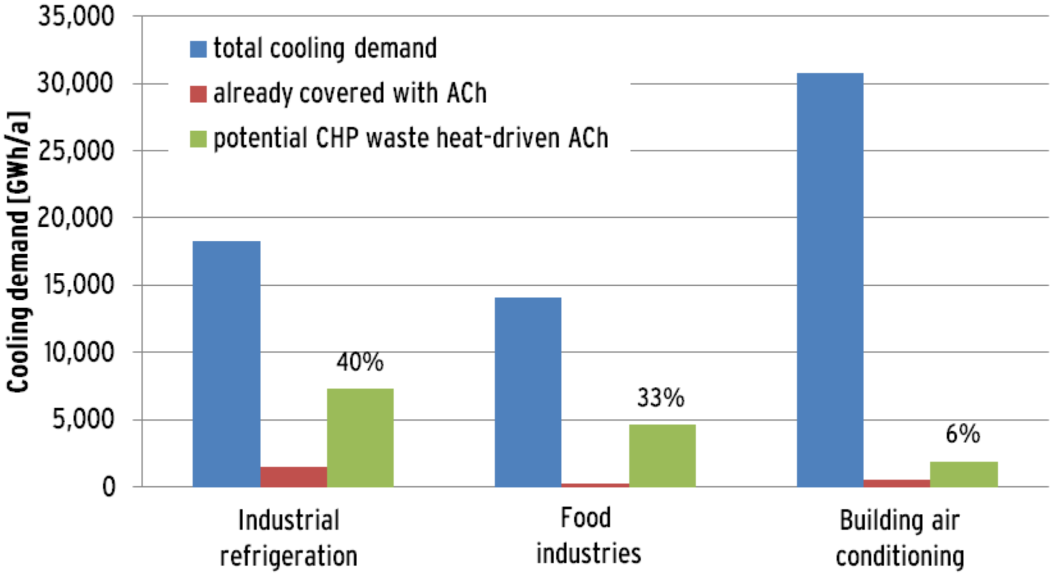


Figure 112: Cooling demand, present, and prospective waste-heat-driven refrigeration technology – potential: natural-gas-fired CHP plants

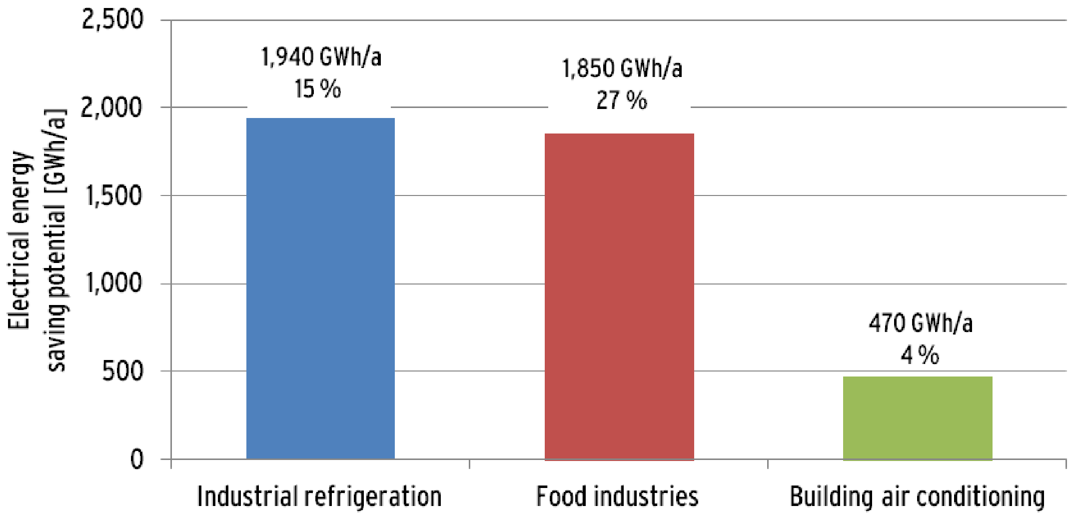


Figure 113: Obtainable electricity savings in different applications (percentages based on electricity demand of a particular sector) – potential: utilisation of natural-gas-fired CHP plants – auxiliary electrical energy demand of ACh not considered

8.3 Discussion of the potentials

The share of heat-driven systems for industrial cooling generation is approx. 9 % already today. Especially, in the chemical industry and production sites with an own energy supply (e.g., semi-conductors industry) has a large share. Figure 114 illustrates the cooling demand as well as the cover ratio by thermally driven cooling generation. The figure comprises an itemisation due to the application and the potentials already discussed. Especially, for the industrial refrigeration and food production, there exist potentials with double-digit percentages.

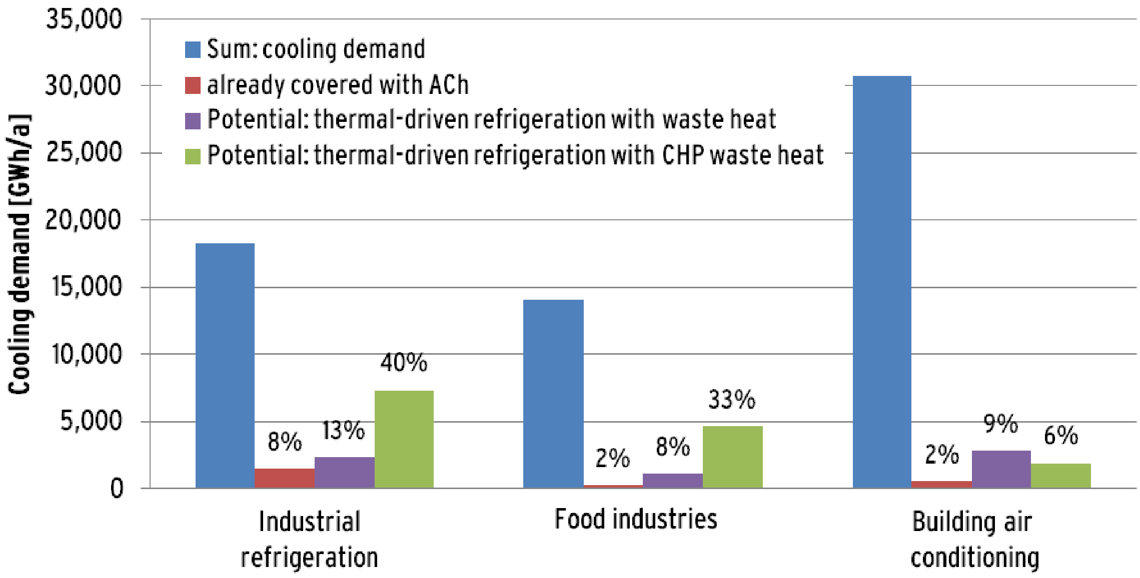


Figure 114: Display of potentials for thermally driven cooling generation

The comparison of TEWI saving potentials shall be restricted to the indirect share, which is directly related to the electrical energy saving potential and the specific TEWI assessment of the driving heat. For displaying the potential, the figures are based on the non-utilised process heat and the waste heat of CHP, considering the electricity displacement mixture. Hence, the heat is evaluated with 0 kg/kWh of CO_2. For the calculation of the electricity savings, both the prospective cover ratios (itemised by sub-sectors) and the specific electricity saving potentials (see Chapter 7) are employed. In order to facilitate the comparison, specific saving potentials for industrial refrigeration *single-effect* NH_3-H_2O ACh with desiccative cooling and indirect cold distribution compared to direct evaporation refrigeration systems with R134a are assumed 58 %. For building air conditioning, a specific saving potential of 65 % is assumed (for single-effect H_2O-LiBr ACh compared to Ch with R134a). In practice, there are larger saving potentials due to *multi-effect* absorption systems and energetically less efficient comparison systems. It shall be noted that the saving potentials consider the auxiliary electrical energy demand of the heat-driven refrigeration systems including pumps and re-cooling.

Figure 115 illustrates the electricity saving potentials. Utilising the specific emissions for electricity generation, the indirect TEWI saving potentials can be calculated as visualised in Figure 116.

The results of the electrical energy savings are partially far below the potentials based on the cooling demand. This concerns the industrial refrigeration, especially. The reason for this deviation is the influence of the gas liquefaction in the chemicals industry. Due to the low process temperatures, a low cooling demand of approx. 1,100 GWh/a has to be provided with a quite large final electrical energy demand of approx. 7,500 GWh/a (see subsection 5.4.5, p. 46ff.). An integration of an absorption type refrigeration system is not useful here.

With the application of *multi-effect* absorption type refrigeration systems and newly developed sorption systems (e.g. directly air-cooled absorption type refrigeration systems), it is possible to make further potentials for final electrical energy substitution accessible. However, they are not investigated within the present study.

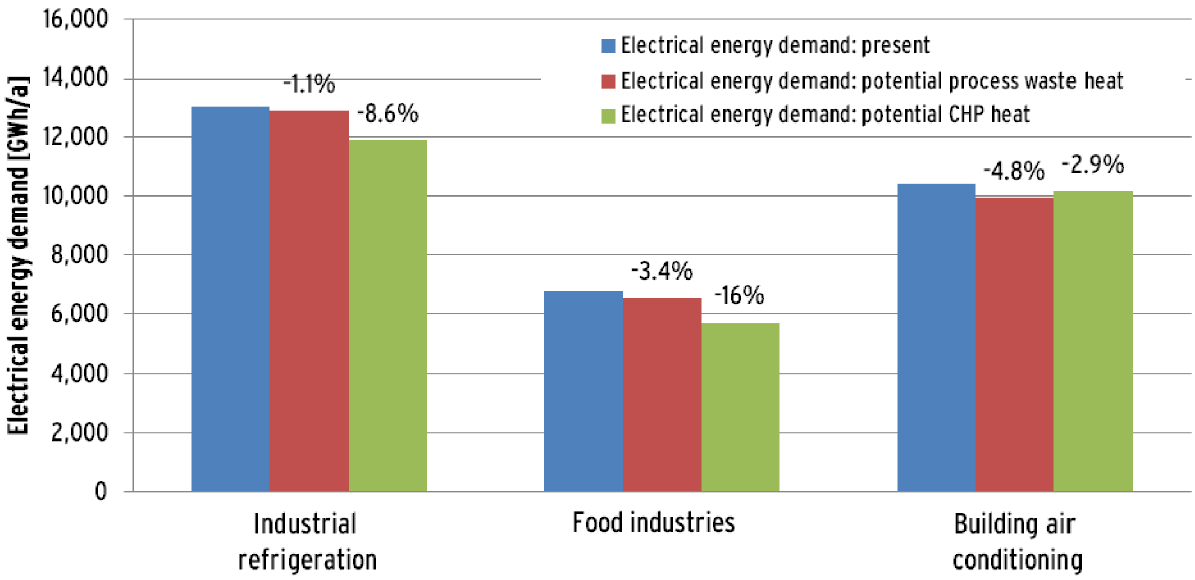


Figure 115: Electrical energy demand of the refrigeration technology in different applications, considering heat-driven cooling generation potentials (electrical auxiliary energy demand of ACh for pumps and ventilators is considered)

The resulting indirect TEWI reduction potentials are visualised in Figure 116. The sum of the reduction potential of 1.6 Mt/a CO_2 corresponds to 4 % of the indirect greenhouse gas emission of the entire stationary refrigeration technology, and to 9 % with regard to the applications under consideration.

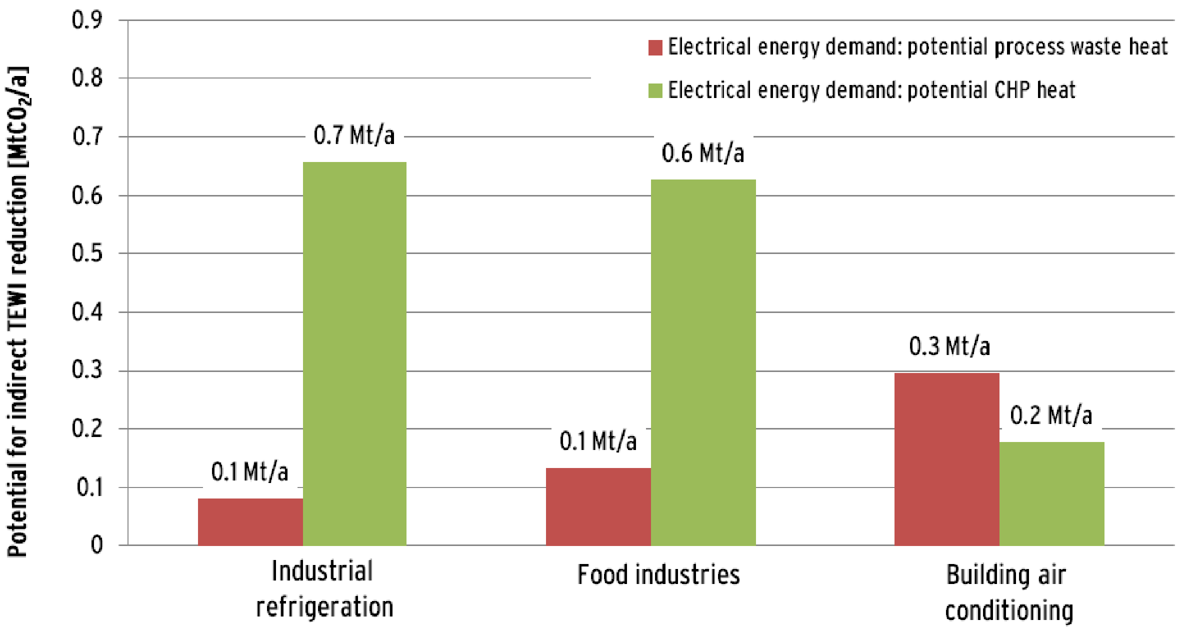


Figure 116: Indirect TEWI reduction potential of different applications, considering heat-driven cooling generation potentials (electrical auxiliary energy demand of ACh for pumps and ventilators is considered)

9 Improvement of the market situation of environment-friendly refrigeration technologies

As mentioned already in previous chapters, there are many aspects to be considered for climate-friendly refrigeration and air conditioning. Among those, the choice of a refrigerant with a low GWP value and low toxicity in conjunction with the design and installation of very efficient and well-tightened systems is crucial. Moreover, the cooling distribution system shall be suitable and be operable with relatively large supply temperatures (see, e.g., chilled ceilings, concrete core activation). Furthermore, for the case of existing heat sources, the application of ad- and absorption type refrigeration plants and processes with open-system ad- and absorption air desiccation shall be investigated.

Depending on the application and its system characteristic, the above-mentioned factors have a different importance. For instance, the tightness is very important for multi-split systems and for the assemblage of VRF systems for building air conditioning due to their large number of pipe junctions and fittings with their increased likeliness for leakages. The direct TEWI emission share originating in refrigerant leakages is relatively large due to a low number of full-load hours. Many industrial sectors, in contrast, have a cooling demand with large full-load hours, where it is more important to reduce the energy demand.

The variety of applications and the large number of different technologies lead to very different barriers regarding the market position of climate-friendly refrigeration technologies. This situation requires a closer examination of tools for overcoming these barriers. To improve the market potential, the following fields are considered separately:

- General factors and system concepts
- Refrigeration and air conditioning systems with natural refrigerants
- Ad- and absorption refrigerants

9.1 General factors and system concepts

In many market fields with a relevant energy demand, products for increasing the energy efficiency are already commercially available. These are, for instance, illumination technologies (energy-efficient lamps, LED), thermal insulation of buildings, brushless direct current motors (EC motors), etc. However, some of the climate-friendly refrigeration technologies are not entirely developed yet, leading to a system of self-presupposing dependencies (see Figure 117 for illustration).

- As long as there are no products available, prospective customers are doubtful regarding functionality, safety, and cost-effectiveness.
- Scepticism leads to a lower demand of products, which are still in development or almost launched on the market.
- Due to large costs for product development and market launch, relatively small number of items per repetition part, and low demand, there are only few products developed and launched on the market by manufacturers.

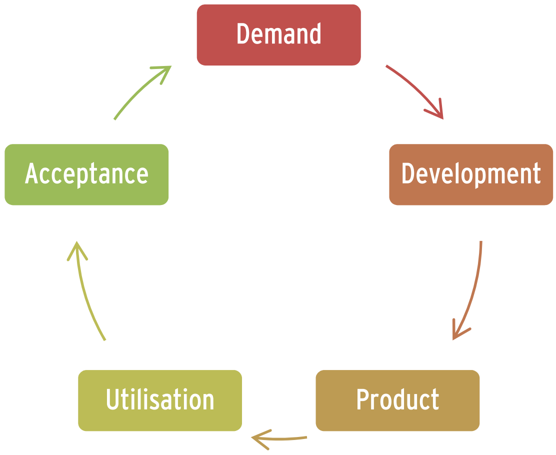


Figure 117: Self-presupposing dependencies for the application of new technologies

9.1.1 Sales sector

Very different refrigeration systems have been investigated within the present study. The presentation in section 6.1, p. 57ff. demonstrates a variety of other systems and system variants. A comparison of the systems revealed advantages of the combination of particular systems, e.g., the technological separation of dehumidification and cooling or the parallel installation of CRP and ACh. These observations imply that standard systems have usually the lowest investment costs but are not necessarily the most suitable in terms of functionality and climate-friendliness.

The majority of building services and installation enterprises restrict themselves to a major supplier, justified by:

- Knowledge within enterprise regarding the system engineering, and
- Favourable purchase conditions with larger order quantity.

Hereby, the enterprises are tied to the portfolio of the major supplier, which often does not exceed standard solutions with HFC refrigerants. Alternative solutions cannot be offered to the customer, and hence they cannot be requested.

In order to overcome this barrier, it is necessary to make plant operators and builders aware of alternatives for cooling generation and air conditioning. This knowledge shall be also available to planners and installers to be able to offer and implement superior solutions.

Recommendations

The information of plant operators and builders represents an important source of motivation for planners and installers. If specific requirements are available for contracting, these have to be realised and cannot be replaced by standard solutions. This implies that operators and builders have to be primarily informed about available technologies and their advantages. Information brochures shall be focused on this particular audience, whilst refrigeration and air conditioning technology alternatives shall be integrated better into journals for building and other relevant industries. Different alternative technologies with their particular assets and drawbacks along with their availability shall become obligatory for the education of civil engineers, architects, and energy consultants.

9.1.2 Lack of awareness

The customer usually recognises a malfunctioned heating or a bad working engine. Furthermore, consumption displays illustrate the influence of the own actions on the energy demand and energy efficiency, facilitating the awareness for the process. However, such awareness does almost not exist for refrigeration systems. Technical problems and installation deficits, which, for instance, yield to leakages, become visible not before remarkable incidents occur. These might be, e.g., a strong increase of room temperatures or the fouling of reefer cargo in cold storage rooms. A separate recording of the net and final energy demand of refrigeration technology is uncommon.

Measurement techniques for these observations are available in a broad range, such as measurement devices for electrical energy demand, filling levels, pressure, and temperature. Cold meters are available for indirect systems (e.g., chillers) analogue to heat meters. For direct evaporation systems, it is necessary to measure the refrigerant mass flow rate and the specific enthalpies. Standard systems are presently not available, but some systems are in development (Tzscheutschler 2011). Systems without dry-expansion evaporation are difficult (e.g., refrigerant circuit systems), since the enthalpy of a two-phase flow has to be determined at the inlet and outlet. There are not any sensors available for the direct measurement of such quantities. For classical dry-expansion evaporation systems, it is possible to estimate the mass flow rate by virtue of the pressures and the compressor map. The required pressure values are usually available in larger systems, since they are utilised for control.

Indirect leakage detection via process quantities is possible, too. Hereby, leakages can be detected much easier and, more important, prior to serious impacts (e.g., loss of cooling power, shut-off systems by pressure control device).

Recommendations

The objective shall be the sensitisation of refrigeration system operators. Therefore, financial promoters for the application of automatically evaluated cold meters and separate energy meters shall be considered in conjunction with indirect leakage detection systems. Hereby, promoting the renewal of systems with a proved efficiency increase might promote the utilisation of installed measurement devices. Instead of financial support, another stimulus for leakage detection systems might be taxes on refrigerants depending on the GWP (see subsection 9.2.5). For such a case, the system operator has the opportunity to install an almost hermetic system or a system with many detachable joints but with an effective leakage detection system.

Moreover, the development of controlling tools for the automatic emission of error messages by SMS or e-mail shall be promoted.

9.1.3 Legal requirements

EU directive on the energy performance of buildings (EPBD) — energy saving regulations (EnEV)

The EU directive on the energy performance of buildings (EPBD) fixes requirements of an energetic minimum standard for buildings. These requirements have been implemented into national law through the Energy Saving Ordinance (“Energieeinsparverordnung”, EnEV).

The present Energy Saving Ordinance for buildings (EnEV 2009) provides structural and system engineering requirements on buildings, which are heated and/or cooled consuming energy. Moreover, requirements on the heating, cooling, air handling, hot water, and illumination systems for buildings are defined. In order to facilitate the ordinance, the EnEV provides a method for balancing the final energy entering and leaving the building with its primary energetic evaluation. Initial point for the building rating is the calculation of the primary energy demand of a reference building with corresponding reference technologies. This particular primary energy demand is considered the upper limit for the real building including the systems engineering. The primary energy demand of the real building is calculated according to (DIN-V-18599-1 2012) or – for existing buildings – by means of the measured energy consumption.

Additional requirements, which arise from the EnEV for air conditioning and refrigeration systems (EnEV 2009), comprise the following:

- For ventilators/HVAC systems only with design volume flow rates above 4,000 m^3/h
- Automatically regulated control units for systems with concerted room air humidity control with different set values for (de-)humidification and with measured fresh and exhaust air humidity as set point
- Load or time-depending airflow for central ventilation systems for fresh air volume flow rates above 9 $m^3/(h \cdot m^2)$. Air volume flows for complying with safety and health requirements are excluded
- Calculation procedure for the heat load of cold distribution and chilled water pipes as well as armatures
- Heat recovery systems (at least classified as H3 according to DIN EN 13053) with a volume flow rate above 4,000 m^3/h

The amended EPBD 2010 demands that new buildings have to be nearly zero energy buildings from 2020 (2019 for public buildings) on (clause 9). Moreover, the remaining low energy demand shall be covered by renewable energy sources, which shall be located close to the building. Some of the requirements will be incorporated into the amended EnEV 2014 already, which will enter into force as from 1.1.2016 (Tuschinski 2013). Major changes are:

- The annual primary energy demand of the reference building has to be multiplied by 0.75.
- The heat protection of the building envelope will be increased.
- The non-renewable share of electrical energy shall be evaluated primary energetic with 1.8.
- For CHP-heated buildings, DIN V 18599 procedure B has to be employed, and contrary to the standard, the supplied electrical energy from CHP plants the value 2.3 has to be employed.

Eco-Design directive for refrigeration and air conditioning systems

The Eco-Design directive serves an environmentally responsible design of energy-driven products within the European Union. It is implemented by the EuP directive 2005/32/EG (energy-using products – environmentally responsible design of energy-driven products) and the ErP directive 2009/125/EG (energy-related products – extension by products relevant for energy demand, such as windows, insulation materials, etc.) (Siede 2011). The implementation into German law has been carried out with the so-called "Energiebetriebene-Produkte-Gesetz" (EBPG) as from 27.02.2008.

The EuP and ErP directives consider the entire energy demand of a product lifecycle from production over consumption during operation to its disposal. The directives require further product-specific regulations, which entered into force as soon as the directives had been published. For the present range of applications, the following regulations are relevant:

- ENTR Lot 1: Cooling and refrigeration systems, including refrigeration systems for industrial processes, central refrigeration systems, cooling chambers, cooling rooms, blast chiller, and condensers
- ENTR Lot 6: Air conditioning and ventilation systems
- ENER Lot 10: Room air conditioning systems up to 12 kW cooling power and ventilators
- ENER Lot 11: Pumps, ventilators, and electric motors
- ENER Lot 12: Commercial cooling and refrigeration systems
- ENER Lot 13: Domestic cooling and refrigeration devices

The regulations for room air conditioning systems and ventilators (regulation (EG) No. 206/2012), circulation pumps (regulation (EG) No. 641/2009), and ventilators (regulation (EG) No. 327/2011) have already entered into force. All the others are presently in the draft phase. A first draft of the European Council for energy-efficient economics has been already provided for the ENTR Lot 1 (ECEEE 2011). This draft is concerned with chillers offered as complete sets:

- Allowed minimal performance numbers and seasonal energy efficiency ratios of air-cooled condensers; calculation of the seasonal energy efficiency ratio according to EN 14511
- Reduction of the energy demand by 2 to 5% from 2014 and by 12 to 16% from 2017 on

Requirements are formulated for the maximal thermal transmission coefficient of cooling chambers. Moreover, rules arise for the product documentation.

It needs to be mentioned that the draft provides reference temperatures for the heat rejection along with the performance numbers and seasonal energy efficiency ratios, but no temperatures for the heat supply. Hence, an evaluation of the data is difficult. The direct TEWI emissions are assigned to be 3 to 6% in preliminary studies and therefore not considered in the requirements.

9.1.4 Investment marketing – energy saving contracting

Individual solutions, complex systems, compression type refrigeration systems with natural refrigerants, and ad- and absorption type refrigeration systems require generally larger investment costs (see discussions in subsections 9.2.5, pp. 172ff. and 9.3.3, pp. 179ff.). The majority of the cost originates in additional material expenses (complex systems) and additional effort for design and assembly compared to serial products.

Due to the financial and economic crisis, the risks of common investments on the capital market have risen, whilst both the return and expected return have diminished. For instance, considering German government bonds, the return rate decreased continuously from 8 % to 2 % from 1995 to 2013 (TradingEconomics, 2013).This implies for institutional investors that money is available at the capital market at very good conditions due to the low key interest rate (Fankhauser 2012).

Contrary to the development of shares or government bonds, energy prices increase and it is expected that this trend will continue due to increasing world population, degree of industrialisation, and living standard (Leipziger Institut für Energie GmbH 2012; Kafsack 2011; EC 2011). Investments into measures for energy efficiency are an interesting alternative to common offers at the capital market. There are already some service providers with various concepts, e.g., as pure investors (Fankhauser 2012) or providers serving as energy suppliers (Tejero 2012). There are further models conceivable for the case of operation cost savings due to climate-friendly refrigeration technologies. However, initial installations for air conditioning are difficult to estimate, since there is no experience for new buildings and their utilisation available. Otherwise, the expansion of CHP plants in conjunction with ACh or the substitution of conventional chillers by highly efficient chillers utilising natural refrigerants appears to be interesting.

Recommendations

It shall be stressed that the elucidation of financial service providers regarding refrigeration and air conditioning is important; e.g., at information events. A good example is the platform Energy Efficiency in Industrial Processes (EEIP, www.ee-ip.org). It efficiently connects developers and manufacturers of efficiency technologies with their customers and corresponding service providers (e.g., of the financial sector).

It is imaginable to invite such service providers to fairs (e.g., ISH, Chillventa) and conferences (e.g., DKV meetings). An inclusion into talks offered there is greatly appreciated. Investors and service provider, working in these sectors, shall be motivated to publish in journals of refrigeration and air conditioning technologies.

9.1.5 Pilot projects — public procurement

Solutions, such as concrete core activation or sorptive dehumidification exhibit a relatively large investment effort and require an early inclusion into building design. Short-term design phases and the desired short payback periods in the industry are also responsible for the exclusion of new solutions. With a concerted promotion of climate-friendly air conditioning systems for new public buildings, the application and dissemination of alternative solutions can be increased. In conjunction with a programme for the monitoring of such systems, the obtained results of public projects can be provided to interested customers. A comparable open handling with such experiences cannot be expected from the industry.

Recommendations

It is recommended to introduce a compulsory evaluation of climate-friendly refrigeration systems for new public buildings. In order to facilitate a detailed analysis and documentation of the experiences, sufficient

measurement technology shall be considered in the design stage already and implemented later on.

9.2 Refrigeration and air conditioning systems with natural refrigerants

The application of natural refrigerants for domestic refrigeration is a standard for over 20 years. For instance, in all fridges and freezers offered within Europe, the refrigerant R600a (isobutane) is employed. Moreover, also in industrial plug-in refrigeration systems (deep freezing chests, beverage chillers) the application of R290 (propane) and R600a is the standard yet.

The market review of components for natural refrigerants exhibited that many components of several manufacturers are offered already today. However, the scale of production is below those for HFC refrigerants (see subsection 6.2.3, pp. 69ff.)

The barriers and promoters are itemised as follows:

- Safety-related barriers and limitations by law
- Incentives for the application of natural refrigerants – limitation of the HFC refrigerant utilisation
- Application-technical barriers
- Technical barriers
- Economical barriers
- Public procurement

There are several factors for many of the above-mentioned categories.

9.2.1 Safety-related barriers and limitations by law

Products being produced or offered in Europe have to comply the respective regulations of the European Union. For the refrigeration and air conditioning systems, these are the machinery directive, low voltage directive, electro-magnetic compatibility directive, and the pressure equipment directive. The implementation into national law has been carried out with the so-called "Produktsicherheitsgesetz" (BMJ 2011).

Due to legal requirements and product liability, the manufacturers generally refer to harmonised European standards. For refrigeration and air conditioning systems, the (DIN-EN-378-1 2011) is in the foreground. This particular standard regulates the itemisation of refrigerants, systems, and placement conditions. Based on this information, the maximum refrigerant charge can be calculated. Eighteen different variants for the placement conditions are considered, including direct and indirect systems. All of those variants are further itemised into refrigerant categories (A1, A2, A3, B1, and B2).

Table 28: Itemisation of refrigerants, systems, and placement conditions in DIN EN 378

Category	Group
Refrigerant	According to toxicity (A and B) and inflammability (1…3) (see Table 6, p. 39)
System	**direct** (evaporator has direct contact to medium to be chilled) **indirect** (evaporator chills medium, which transfers cold to heat exchanger, which has contact to medium to be

	chilled, in a closed circuit
Placement condition	**Class A –** General placement area (location, where any person without any safety precautions may have access to or is allowed to sleep) – e.g., hotels, restaurants, sales areas, public buildings **Class B –** Controlled placement area (rooms, parts of a building, or buildings, where the number of people is limited and at least some them are familiar with the safety precautions) – e.g. office buildings **Class C –** Placement area, with authorised people access only (those people have to be familiar with the security precautions)

There are no general limitations regarding the location of permanently closed refrigeration circuit systems with refrigerants of the classes A2 and A3 (hydrocarbons, e.g., R290) and a maximal refrigerant charge of 150 g. However, even very compact air conditioning systems (e.g., facade-integrated systems) are hardly realisable with this small amount of refrigerant.

Direct systems for general placement areas – mono- und multi-split systems

There is a maximum refrigerant charge under the following requirements:

- for all placement areas,
- direct air conditioning systems with outdoor units (compressor, condenser, refrigerant receiver),
- operated with refrigerants of classes A2 and A3, and with
- refrigerant charge above 4 m^3 x LEL (LEL: lower explosion limit in kg/m^3).

The maximum charge size can be calculated with:

$$m_{max} = 2.5\ LEL^{\frac{5}{4}}\ h_0\ \sqrt{A}.$$

Herein, h_0 is the installation height of the device, and A is the floor area. For a given refrigerant charge, it is possible to calculate the minimum floor area as:

$$A_{min} = \left(\frac{m}{2.5\ LEL^{\frac{5}{4}}\ \ h_0}\right)^2.$$

Table 29 provides an overview of all safety-related data of inflammable natural refrigerants. Supplementary, Table 30 provides data for the installation heights for particular kinds of placement.

Typical mono-split units have refrigerant charges between 200 g and 1 kg (see Figure 118). For instance, mono-split systems with 400 g R290 can be employed for rooms with a floor area larger than 28 m^2. Multi-split or VRF systems have a much larger refrigerant charge (sometimes more than 40 kg) in their entire refrigerant circuit yielding very large minimal floor areas. Hence, the application of flammable refrigerants in multi-split systems is ruled out.

The total charge size of class A1 refrigerants (e.g., R744, carbon dioxide) is limited to the *practical limit x room volume*. For R744 and a room volume of 50 m^3 (i.e., 20 m^2 floor area, 2.5 m ceiling height), the maximum charge size is 5 kg. For multi-split systems and the very common refrigerant R410A, there is an analogue limit of 22 kg. There are no limits for office buildings (B2).

Table 29: Data of inflammability/explosiveness of various refrigerants of the safety class A3, reference: DIN EN 378, Safety data sheets of refrigerants

Refrigerant	R170	R1150	R290	R1270	R600	R600a	R32	R152a
Trade name	Ethane	Ethene, ethylene	Propane	Propene, propylene	n-Butane	Isobutane	CH_2F_2	CH_3-CHF_2
LFL (lower explosion limit acc. DIN EN 378)	0.038 kg/m^3	0.036 kg/m^3	0.038 kg/m^3	0.047 kg/m^3	0.048 kg/m^3	0.038 kg/m^3	0.307 kg/m^3	0.130 kg/m^3
Lower expl. limit	2.4 vol%	2.4 vol%	1.7 vol%	1.8 vol%	1.4 vol%	1.5 vol%	14 vol%	approx. 4 vol%
Upper expl. limit	16 vol%	34 vol%	9.5 vol%	11 vol%	8.5 vol%	8.5 vol%	31 vol%	approx. 17 vol%
Ignition temperature	515 °C	425 °C	470 °C	455 °C	365 °C	460 °C	648 °C	455 °C
Min. Ignition energy	0.25 mJ	0.082 mJ	0.25 mJ	0.18 mJ	0.25 mJ	0.18 mJ	–	–

Table 30: Installation height depending on the kind of placement, reference: DIN EN 378

Kind of placement	Installation height h_0
Placement on the floor	0.6 m
Wall-mounted	1.8 m
Window-mounted	1.0 m
Ceiling-mounted	2.2 m

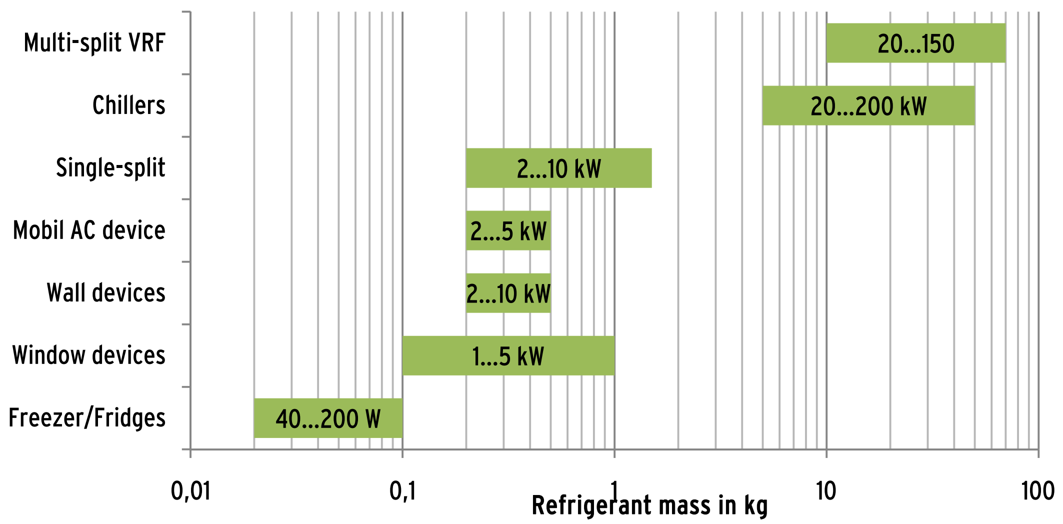


Figure 118: Typical refrigerant charges and power ranges of air conditioning systems (Daniel Colbourne 2012), complemented with data for fridges and freezers

The total refrigerant charge of class B2 refrigerants (e.g., R717, ammonia) is limited for direct systems in public buildings to the *practical limit x room volume*. This yields a limit of 17.5 g R717 in a 50 m^3 room, whereas the limit is 25 kg for office buildings (class B). Hence, the installation of multi-split systems with R717 in office buildings is allowed to some extend. Advantage of R717 is its good perceptibility at very low and relatively safe concentrations.

Table 31: Practical limits of various natural refrigerants according to DIN EN 378

Refrigerant	R1150	RE170	R290	R1270	R600a	R717	R718	R723	R744
Trade name	Ethene, Ethylene	Dimethyl ether, DME	Propane	Propene, Propylene	Isobutane	Ammonia	Water	Ammonia + DME	CO_2
Practical limit	0.007 kg/m³	0.013 kg/m³	0.008 kg/m³	0.008 kg/m³	0.011 kg/m³	0.00035 kg/m³	-	No information	0.1 kg/m³

Indirect systems with outdoor placement or in the machine room

The placement of all system components with refrigerant outdoor or in a machine room is the most convenient case regarding the allowed refrigerant charge. A tight casing with mechanical ventilation can be considered as machine room. However, it has to be ensured that leaking refrigerant cannot enter nearby rooms, staircases or yards, but has to be extracted safely.

Presuming that authorised people only have access to the machine rooms or the outdoor placement, DIN EN 378 does not impose any limitations.

Summary of safety-related barriers

Based on the DIN EN 378, which is the safety standard to be applied, Table 32 summarises the opportunities for the application of natural refrigerants for various applications. DIN EN 378 does not limit the outdoor placement of indirect systems, which is very common for air conditioning chillers. For direct systems, it is required to consider the placement classes A and B. In placement areas of class B, to which office rooms are explicitly assigned by DIN EN 378, the application of multi-split systems with R744 (CO_2) is unproblematic and with R717 (NH_3) or R723 (NH_3+DME) it is possible but with limitations. Installation of single-split systems with inflammable refrigerants is limited, especially with more than 150 g refrigerant only in accordance with the room volume. At this point, the study of the Tianjin Fire Research Institute (China) shall be mentioned, which will be briefly presented later on.

For industrial refrigeration systems, there are some limitations for all refrigerants, except for R718 (water). For R744 (CO_2) these are the fewest, whilst they are the most for inflammable refrigerants of class A3. By means of combinations of systems, it is possible to reduce the limitations. For instance, for low temperature refrigeration it is convenient to have a cascade of R290 in the large temperature stage and R744 with direct evaporation in the low temperature stage.

Table 32: Opportunities for the application of natural refrigerants for various cases according to DIN EN 378

	Direct systems for air conditioning		Indirect systems for refrigeration or air conditioning	Direct systems for industrial refrigeration
Refrigerant class	Mono-Split	Multi-Split	Chiller outdoor-placed	
Inflammable refrigerant A3 (R290, R1270 etc.)	Possible with limitations	Not possible	Possible without limitations	Possible with strong limitations
Inflammable and toxic refrigerant	Possible in offices (class B)	Possible with strong limitations in public buildings (class B)	Possible without limitations	Possible with limitations

B2 (R717, R723)	Not possible in public buildings (class A)	Not possible in public buildings (class A)		
Carbon dioxide A1 (R744)	Possible in offices (class B)	Possible in offices (class B)	Possible without limitations	Possible with low limitations
	Possible with low limitations in public buildings	Possible with limitations in public buildings		
Water A1 (R718)	Possible without limitations	Possible without limitations	Possible without limitations even in public areas (class A)	Possible without limitations

Study on the hazard potential of mono-split air conditioning systems with R290

The Tianjin Fire Research Institute (TFRI, China) has published a study on the hazard potential of mono-split systems with propane as refrigerant. The authors carried out numerical and experimental investigations of the transient propane distribution after a leakage in conjunction with various outdoor units in high-rise buildings.

The study showed that, although the refrigerant charge in mono-split systems is larger than that of domestic refrigeration systems, the propane concentration after a leakage is almost equally large (see Figure 119). Due to the larger density of the hydrocarbons isobutane and propane, they accumulate at the floor. Contrary to fridges, where the leakage occurs close to the floor, the air conditioning indoor device leakages take place close to the ceiling. Due to the ventilator, the refrigerant is distributed in the entire room.

Explosion incidents of combined fridges and freezer take place with a frequency of $6x10^{-8}$/a, whilst the acceptable threshold for public areas is 10^{-5}/a (Daniel Colbourne 2012). However, the frequency of explosion incidents of mono-split units has been observed to be $5x10^{-10}$, which is two orders of magnitude below combined fridges and freezers. Hence, it can be stated that the hazard potential of a mono-split air conditioning system with propane is lower than that of common domestic combined fridges and freezers.

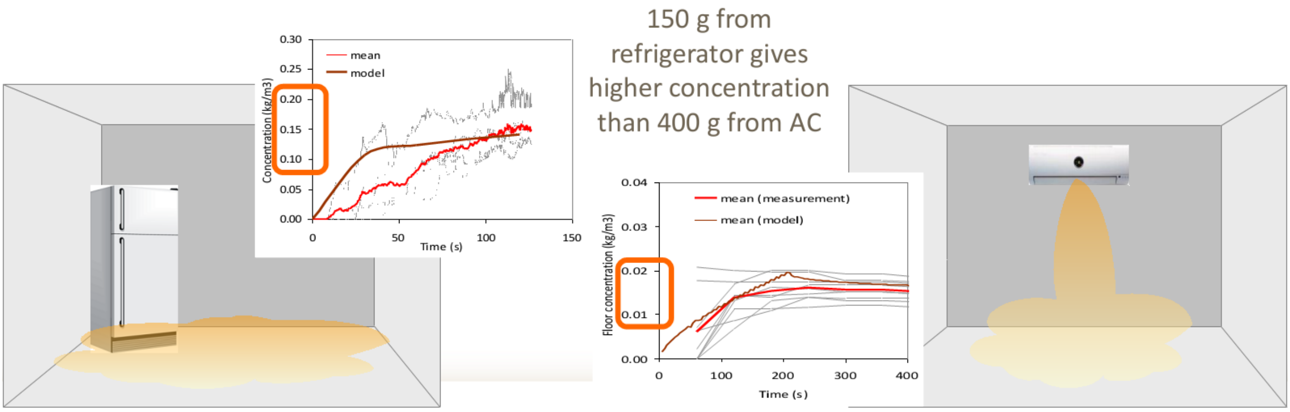


Figure 119: Refrigerant distribution in the room after leakage; comparison of a combined fridge/freezer and a single-split air conditioning device; reference: Daniel Colbourne 2012 – results of the study of the TFRI

Summary

The existing standards allow the application of natural refrigerants in a broad range. For air conditioning applications, it is possible to employ them in indirect systems for almost all cases.

Recommendations

The results of the study (Daniel Colbourne 2012) shall be verified in further investigations. An adaption of the DIN EN 378 according to the new observations is recommended.

9.2.2 Incentives for the application of natural refrigerants – limitations of the application of HFC refrigerants

Legal regulations can yield an incentive for the application of natural refrigerants by taxation, limitation, and prohibition of HFC refrigerants. In some European countries, experiences with such incentives have been made already.

Emission-justified taxes of greenhouse gases

For the enforcement of the objectives for the reduction of greenhouse gas emissions defined in the Kyoto protocol, different countries utilise various measures. One of those measures is the trade with emission certificates. The idea behind is to reduce the emissions with the least economic costs, i.e., the emissions of those processes shall be reduced, where the reduction requires the least costs. Greenhouse gases have to be considered according to their GWP value.

The taxation of refrigerants according to their GWP values picks up the idea to reduce emissions by economic means. Hereby, incentives are made to minimise the refrigerant charge, increase the system tightness and their control, and choice of small GWP value refrigerants.

Table 33 lists taxes for HFCs including their definitions. For comparison purposes, the reference prices for the refrigerants R134a and R404A are between 10…16 €/kg in Germany. Due to the taxation, the prices increase between 40 and 220 % for R134a and between 120 and 620 % for R404A. A deposit on HFC, which creates an incentive for appropriate disposal, is applied in Norway only.

Table 33: Taxation of refrigerants of different, references: (Honeywell 2012; Rhiemeier, Harnisch & Kauffeld 2008; Strogies & Gniffke 2013)

Country	Costs per CO_2 equivalent	Example for R134a	Definitions/comments
Australia	approx. 18.25 €/$tCO_{2,eq}$ (23 AU$/$tCO_{2,eq}$)	approx. 26.10 €/kg_{R134a}	Annual price rise of 5% is fixed
Denmark	approx. 20.00 €/$tCO_{2,eq}$ (150 DKK/$tCO_{2,eq}$)	approx. 28.60 €/kg_{R134a}	To be paid at import, except for systems already filled (e.g., fridges)
Norway	approx. 24.00 €/$tCO_{2,eq}$ (190.50 NK/$tCO_{2,eq}$)	approx. 34.30 €/kg_{R134a}	To be paid at import (no production in Norway), tax implemented as deposit → incentive for suitable disposal
Poland	approx. 4.70 €/$tCO_{2,eq}$	approx. 6.70 €/kg_{R134a}	Final decision in parliament still open
Slovenia	approx. 12.50 €/$tCO_{2,eq}$	approx. 17.90 €/kg_{R134a}	Initial filling considered with 5%, service fillings considered with 100%
Sweden	approx. 22 €/$tCO_{2,eq}$	approx. 31.50 €/kg_{R134a}	Final decision in parliament still open

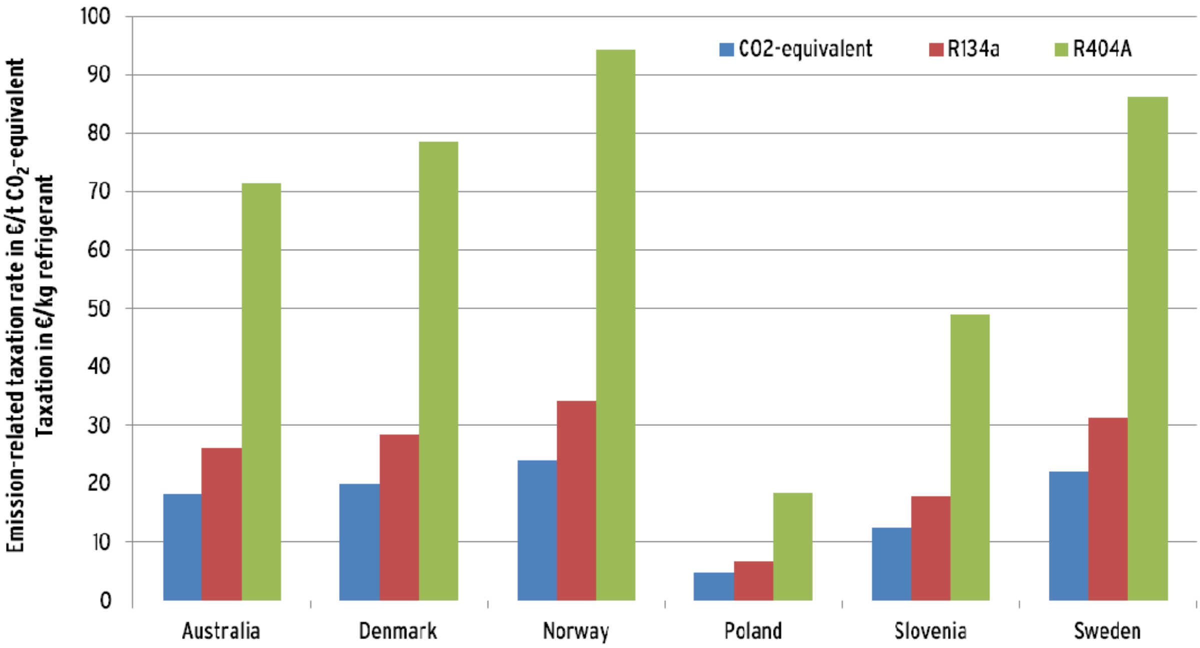


Figure 120: Tax rates based on CO_2 equivalent, R134a, and R404A

Limitations and prohibition of the application of refrigerants with high GWP values

In some European countries national regulations exist, which exceed the European Regulation No 842/2006 on certain fluorinated greenhouse gases and which limit the use of PFCs and HFCs. Most of those regulations entered into force before the F-gas regulation. The decision of the European Council to limit the national regulations' validity until 2012 has been revoked, which allows applying the national regulations furthermore. On overview of those regulations is provided in Table 34.

Table 34: Overview of HFC prohibition in European Countries

Country	Prohibition
Denmark	HFC prohibition for systems with more than 10 kg refrigerant from 1.1.2007 – is allowed to be continued after 2012 according to EU decision (hydrocarbons21-2012)
Austria	Partial HFC prohibition since 2002 with various exceptions (e.g., 20 kg in stationary systems and 100 kg in systems with complex pipe system are allowed)
Sweden	Limit of 30...40 kg HFC
Switzerland	Mandatory approval for HFC application in new systems, extensions, and modifications. Requirement for approval is a lack of natural refrigerant alternatives. For systems with more than 3 kg: notification requirement, service check book, and tightness check

References: (ChemRRV-Schweiz 2006; Hydrocarbons21.com 2012; Rhiemeier, Harnisch, Ters, et al. 2008; WKO 2007)

The effectiveness of the measures with regard to a reduced use of HFC refrigerants can be observed in the field of deep freezing applications. Systems applying R744 are standard for those applications in Scandinavia and Switzerland today (Brouwers 2009). The key influencing factors are the above mentioned GWP-related taxation and further legal restrictions of HFC applications and refrigerant charges.

On behalf of EPEE SKM Enviros has investigated the effectiveness of different scenarios for the reduction of HFC refrigerants (Gluckman et al. 2012). Four different scenarios for the progressive reduction of F-gases in Europe are itemised by their effectiveness in different applications of refrigeration technology (see Table 35). The results of this study are important for the evaluation of the obtainable emission reductions arising from the new F-gas regulation.

Table 35: Scenarios for the HFC phase down within the EPEE study (Gluckman et al. 2012)

Scenario	Explanation	Comments
A	Low impact (base case; all scenarios are compared with this case for the economical analysis)	Scenario A is a conservative point of view on the changes of refrigerant utilisation. It reflects the prospective utilisation of HFCs based on the present legal requirements (especially F-gas regulation of 2006).
B	Average impact	Scenario B introduces the reduction of HFC utilisation in new systems and an improvement of the leakage level by entire implementation of the F-gas regulation.
C	Strong impact	Compared to scenario B, this one assumes (i) a larger degree of very low GWP alternatives, (ii) an earlier application of medium GWP alternatives for new systems in order to avoid new large GWP systems, and (iii) refill a part of existing systems with large GWP refrigerants with alternatives, as long it is feasible.
D	Strongest impact	This scenario exceeds C and assumes the widespread utilisation of A2L (easily inflammable) refrigerants in stationary air conditioning and industrial refrigeration systems from as 2020.

Table 36 summarises the gross emission reduction compared to scenario A for Europe in 2030. The largest share amounts for the commercial refrigeration. The study provides detailed information for the sectors under consideration. Whilst scenario C yields large savings compared to scenario B, scenario D has almost no effect for industrial refrigeration. For stationary air conditioning systems, the largest savings can be obtained with scenario B, whilst C and D do not contribute significantly. Scenario D does not offer any further savings compared to C for chillers.

Table 36: Reduction of gross emissions (Mt_{CO2}) compared to scenario A for Europe in 2030 (Gluckman et al. 2012)

Industrial sector \ Scenario	B	C	D
Domestic refrigeration	0.1	0.1	0.1
Commercial refrigeration	24.2	34.6	34.6
Transport refrigeration	0.9	1.4	1.4
Industrial refrigeration	2.7	5.2	5.4
Stationary refrigeration and heat pumps	14.5	15.4	16.9
Chillers and heat pumps (water)	5.0	5.8	5.8
Mobile air conditioning	2.3	2.5	2.5
Sum	49.6	64.8	66.6
Sum for the study-relevant sectors	22.2	26.4	28.1

Table 37 illustrates the specific costs for emission reduction, compared to scenario A. It is obvious that, economically, scenario C is the most suitable for the industrial refrigeration sector as it leads to negative costs compared to scenario A. The same trend can be observed for chillers, where negative costs are calculated for scenario B. Stationary air conditioning systems exhibit relatively large costs, whereby scenarios B and C are comparable, whereas scenario D exhibits extraordinarily high costs.

Table 37: Costs of emission reduction (€/t_{CO2}) compared to scenario A for Europe in 2030 (Gluckman et al. 2012)

Industrial sector \ Scenario	B	C	D
Domestic refrigeration	-119	-95	-95
Commercial refrigeration	15	23	23
Transport refrigeration	5	-11	-11
Industrial refrigeration	10	-1	16
Stationary refrigeration and heat pumps	24	27	45
Chillers and heat pumps (water)	-7	4	4
Mobile air conditioning	7	11	11
Sum	15	19	25
Sum for the study-relevant sectors	15	16	31

New European Regulation on F-gases

In 2014, the European Parliament and the Council adopted a new regulation on F-gases. The new regulation is one element to reduce non-CO_2 emissions from industry by 70 % by 2030, as planned. The new regulation is intended to provide a share of the reduction, since it comprises many requirements with strong effect on the refrigeration and air conditioning technologies. These requirements are:

- From 2020 prohibition of service for refrigeration systems containing refrigerants with a GWP ≥ 2,500 (e.g., R404A, R507, R422D, R23) and with a refrigerant charge that correlates to at least 5 t CO_2 equivalent.
- Introduction of gradually decreasing limits for bulk cargo HFCs marketed in the European Union from 2016. Until 2030 HFCs will be reduced to 21% of the average between 2009 and 2012 in terms of CO_2 equivalents.
- Prohibition of refrigerants with a GWP ≥ 2,500 in hermetically sealed commercial cooling and refrigeration devices as of 2020. From 2022, refrigerants with a GWP < 150 will be allowed only.
- Prohibition of refrigerants with a GWP > 150 in domestic fridges/freezers from 2015 and movable room air conditioning equipment from 2020.
- HFCs in pre-charged refrigeration, air conditioning, and heat pump systems must be accounted for within the quota system as of 2017.

The temporal development of the prohibitions is illustrated in Figure 121.

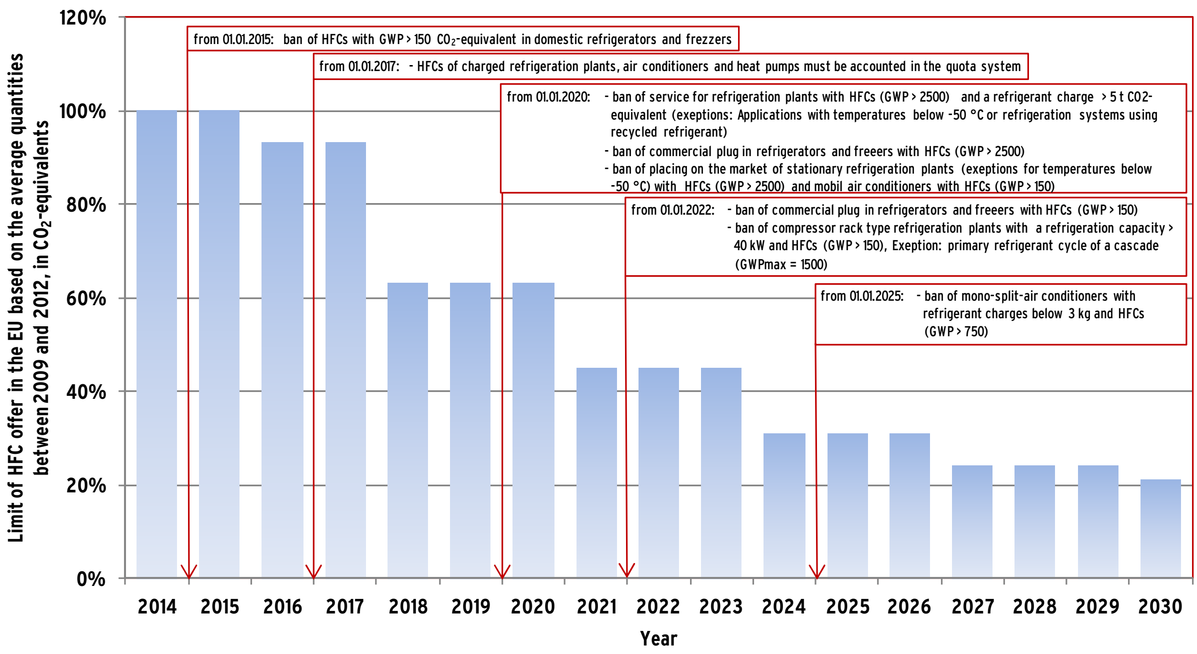


Figure 121: HFC phase down scheme (measured in CO_2 equivalents) according to Regulation (EU) No 517/2014 on fluorinated greenhouse gases

The limitation of HFCs in domestic refrigeration systems affects a limited number of products only. The European manufacturers changed to halogen-free hydrocarbons, primarily R600a, almost twenty years ago. For commercial fridges and freezers mostly R134a (small devices, chilled bars), R404A (chest freezer, beverage chillers), R744 (beverage chillers), and R290 (chest freezer, chilled bars) are utilised. From those examples, it becomes obvious that the change from fluorinated to natural refrigerants is driven by beverage producers. It is expected that HFCs will be entirely substituted in this particular sector in the following years.

The service prohibition for refrigerants with a GWP > 2,500 affects mostly systems with R404A (GWP=3,922) and R507A (GWP=3,985). Both refrigerants were considered to be substitutes for R22 and R502 (Bitzer-Kühlmaschinenbau 2012). These refrigerants are mostly utilised in chillers and in industrial systems. Hereby, 5 t CO_2 equivalent corresponds to a charge size of 1.25 kg R404A or R507A. Since this refrigerant charge is exceeded in almost all chillers and industrial refrigeration systems, the new regulation implies a prohibition of those refrigerants.

However, the chosen limit of GWP=2,500 leaves various refrigerants untouched. Among those are R134a (GWP=1,430), R407C (GWP=1,774), and R410A (GWP=2,088). Especially in multi-split systems, in which R410A is the standard refrigerant and charge sizes of 20-100 kg are common, a large total potential remains untouched. For instance, the CO_2 equivalent charge size here is 40-200 t.

The primary incentive for the change to natural refrigerants is the future limitation to market HFCs, which is supposed to lead to a strong increase of prices for HFC refrigerants.

Recommendations

The implementation of the new European F-gas regulation shall be supported. Additional measures such as an HFC taxation may be considered. The legal limitations of HFC utilisation in Switzerland, Denmark, or Norway, and the GWP-based taxation serve as examples.

9.2.3 Application-technical barriers

Experts interviews – manufacturers of end devices

Based on studies at the ILK Dresden (Müller et al. 2013) and of Shecco (Shecco 2012), information of experts interviews are provided here. Supplementary to them, opinions of refrigerant system manufacturers and specialised companies, which have been gathered in personal interviews, will also be provided.

The study of the ILK Dresden was concerned with natural refrigerants in heat pumps, which have the same transition process from chemical to natural refrigerants and are therefore comparable (Müller et al. 2013). In this study, manufacturers of domestic heat pumps have been interviewed regarding their motivation and barriers for the application of natural refrigerants. The responses could be classified into four categories (see also Figure 122):

- Manufacturers who do not plan to produce heat pumps with natural refrigerants
- Manufacturers who might produce heat pumps with natural refrigerants in the future
- Manufacturers who offer heat pumps with natural refrigerants
- Manufacturers who do not offer heat pumps with natural refrigerants any more

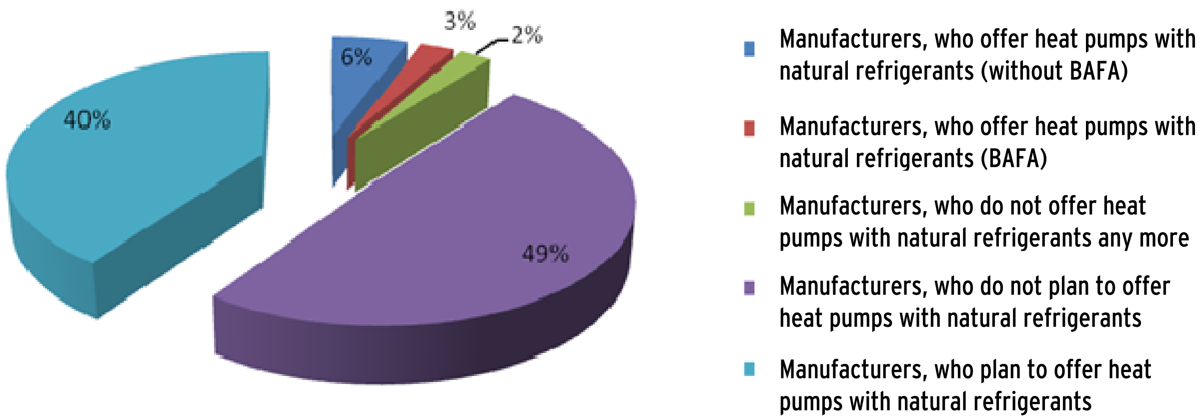


Figure 122: Heat pump manufacturers and their products with natural refrigerants (Müller et al. 2013)

Table 38: Statements of heat pump manufacturers concerning the application of natural refrigerants

Manufacturer class	Statements
Manufacturers, who do not plan to produce heat pumps with natural refrigerants	The following major reasons have been mentioned: • No understanding of necessity (low TEWI values also with hermetic HFC systems at large energy efficiency obtainable) • Disadvantages of natural refrigerants (inflammable, toxic, large pressures) • Reserve of customers and installation enterprises • Large development effort • Small enterprises with limited portfolio and small capacities for R&D • Risk of reduction of market shares due to (more expensive) heat pumps with natural refrigerant Problems with the component procuration for natural refrigerants have hardly been mentioned.
Manufacturers, who might produce heat pumps with natural refrigerants in the future	The following barriers (with decreasing relevance) for the development of heat pumps with natural refrigerant have been mentioned by the manufacturers: • Cheaper components • Assistance with safety-related considerations • Broadening of the portfolio of components • Amendment of the safety standards • Better financial support for customers • Better financial support for manufacturers • Assistance with technical design The lack of pressure from politics has been mentioned, e.g., by taxes on HFCs. Beside natural refrigerants, the developments regarding other low-GWP refrigerants are considered by the manufacturers.
Manufacturers, who offer heat pumps with natural refrigerants	The following desires and requests for improving the market situation of heat pumps with natural refrigerants have been mentioned: • Better financial support for customers of such a heat pump (e.g. 500 €/heat pump) • Amendment of safety standards • Assistance with technical consultation • Broadening of the portfolio of components, cheaper components
Manufacturers, who do not offer heat pumps with natural refrigerants any more	The following reason have been mentioned to take heat pumps with natural refrigerants off the market: • Technical problems with compressor (unexplainable failures) • Long delivery times for components • Difficult to find partners for service (especially for small enterprises) However, many satisfied customers have been mentioned.

Experts interviews — Customer of end devices

The major subject of the study (Shecco 2012) were the customers of end devices. Among those, there are mostly enterprises of trade and food production, and restaurants operating globally.

The following missing incentives have been reported:

- Legal requirements of the phase-down of HFCs similar to the Montreal protocol for CHFCs
- Incorporation of the Eco-Design directive
- Taxation of HFCs

Moreover, the following general barriers are recognised:

- Availability of trained service personnel
- Availability of systems with a competitive price

Further recommendations have been raised:

- Introduction of hybrid refrigeration systems as a first step (e.g., CO_2-R134a cascades)
- Training of refrigeration system engineers and crafts man on systems with natural refrigerants, certification of systems with natural refrigerants
- Development of low-priced compact systems (repetition parts with large lot sizes)
- Training of developers, planners, service personnel, and customers

Expert interview — Refrigeration system enterprises and refrigeration system engineers

In the course of an information event of a large German refrigeration system enterprise on the proposal of the new EU F-gase regulation in March 2013, a good overview of the prospects of refrigeration system enterprises could be obtained by personal interviews and by questions related to the talks. Two requirements have been mentioned related to the implementation of the new F-gas regulation:

- Training with special emphasis on specific properties of natural refrigerants: Contrary to HFCs, whose handling is quite similar, different natural refrigerants have to be handled quite differently. Therefore, it has to be distinguished between pure hydrocarbons, ammonia, and carbon dioxide.
- The required infrastructure and tools in service cars for the work on systems with chemical and natural refrigerants is considered a special problem. The service cars are fully filled with equipment, tools, and refrigerant storages already today. Acquisition, storage, and handling of additional equipment are a problem to be considered.

Recommendations

The following recommendations can be extracted from the experts interviews:

- Legal regulations for the HCF phase-down
- Legal requirements for the taxation according to GWP (see pp. 172ff.)
- Consideration of direct TEWI emissions in the Eco-Design regulations ENTR Lot 1 and Lot 6 (for air conditioning systems < 12 kW (ENTR Lot 10), these are considered with a SEER bonus for systems a refrigerant of GWP<150 (EC, 2012b))
- Training of refrigeration technicians (skilled workers) shall be extended by natural refrigerants
- Support of the development of refrigeration systems series with high quantities (e.g., mono-split air conditioning systems, chillers with small capacity)
- Support of the development/pilot projects of large systems, if necessary as a combination of HFC and natural refrigerant

9.2.4 Technical barriers

There are many technical barriers for the application of natural refrigerants for refrigeration and air conditioning systems due to their specific properties. A major part is concerned with the safety aspects of explosive, inflammable, or toxic refrigerants. Resulting requirements are defined by DIN EN 378. Table 39 lists groups of natural refrigerants with their properties and the corresponding technical requirements. These requirements can be covered by technical measures, but with larger effort and hence costs. However, there is a further demand for research, e.g., for the utilisation of the expansion work of R744 systems, but also for the safety-related evaluation of R723.

Table 39: Natural refrigerants and their properties to be considered

Refrigerant/Refrigerant group	Material property	Technological requirements
Hydrocarbons (e.g., R290, R1270, R601, RE170)	explosive	Explosion-save design Save discharge of possible refrigerant leakage streams Gas detectors for security switch-off
Carbon dioxide (R744)	Very low evaporation temperature →large pressures in the systems (> 100 bar)	Material strength
	Low critical temperature (approx. 31 °C)	Transcritical operation incl. pressure control, gas cooler, and additional circuits for obtaining acceptable COPs (internal heat exchangers, expansion machines, or ejectors)
Ammonia (R717)	Toxicity	Gas detectors
	Inflammability	System for leakage stream solution into water
	Strong solubility	No copper and brass materials applicable
Water (R718)	Small saturation vapour pressures (9 mbar at 6 °C)	Large pipe cross sections Large turbo compressors Decarburisation-free materials

Recommendations

There is a lot of research necessary for increasing the market shares of systems with natural refrigerants. Some approaches will be presented in the following:

With respect to reduced heating and cooling loads, facade-integrated, combined HVAC systems with R290 are an interesting approach. For the case of leakages, an additional airflow to outdoor can be initiated preventing an explosive mixture indoor. By means of a complete production of the refrigeration circuit by the manufacturer, it is possible to avoid detachable joints and to obtain a hermetic circuit, which is the standard for domestic refrigeration systems.

The reduction of the refrigerant charge is an important step for the application of inflammable, natural refrigerants. There are approaches available, which are mainly concerned with the heat exchangers. For instance, air-cooled evaporators made from MPE (multi port extrusion; it is problematic to obtain an equally distributed refrigerant flow in the microchannels) and asymmetric plate phase change heat exchangers.

9.2.5 Economical barriers

Refrigeration and air conditioning systems with HFCs are standard solutions today. Systems with natural refrigerants are considered specialised solutions in many industrial sectors due to their low market share. Some of the natural refrigerants require additional safety systems, such as gas detectors, extractors, or casings, leading to larger investment costs compared to a similar HFC refrigeration system. According to a German manufacturer of R290 refrigeration systems, the additional costs amount to approx. 30 to 60 %.

Due to some effects reducing the operation costs, the amortisation of the additional costs is possible:

- Reduced maintenance costs due to lower effort for tightness check
- Reduced energy costs due to prospective higher system efficiency (especially R717, R723, according to manufacturers also for R290, see p. 136)
- Reduced refrigerant costs for natural refrigerants
- Other advantages due to specific refrigerant properties

Tightness control

The objective of the F-gas regulation is the reduction of fluorinated greenhouse gas emissions. As a measure, operators of such refrigeration systems are obliged to carry out tightness checks frequently and to document them. The frequency of checks depends on the refrigerant charge (see Table 40).

Table 40: Requirements for the tightness checks according to the F-gas Regulation (EC No 842/2006)

Refrigerant charge	Requirements
3…30 kg (>6 kg for hermetic systems)	Annual checks
30…300 kg	Biannual checks, or annual for the case of an existing leakage detection system
>300 kg	Quarterly checks, or biannual for the case of an existing leakage detection system

According to article 3, there are both direct and indirect measurement techniques allowed (see EC Regulation 1516/2007) for the tightness check. It is obliged to check parts of the system, where leakages are most likely. The measurement techniques differ in their accuracy of detection by orders of magnitudes. Whilst direct methods are difficult due to constructional limitations, indirect methods are only applicable for cases of reliable indicators of leakages. The measurement technique comprises the analysis of one of the following parameters: pressure, temperatures, compressor flow, liquid filling level, and refilling quantity.

It shall be stressed that even in systems with frequent tightness checks, total refrigerant losses due to accidents cannot be ruled out.

A very good tightness of the refrigeration system and an optimal refrigerant charge are important for the efficient and economic system operation. Due to these reasons, frequent tightness checks are suggested. Moreover, the good tightness of the systems is important for inflammable or toxic natural refrigerants. Due to the imprecise formulations in the EC Regulations No 842/2006 and 1516/2007 with their resulting freedom of interpretation, marginally larger or even no increased maintenance costs for natural refrigerant systems are expected compared to HFC systems.

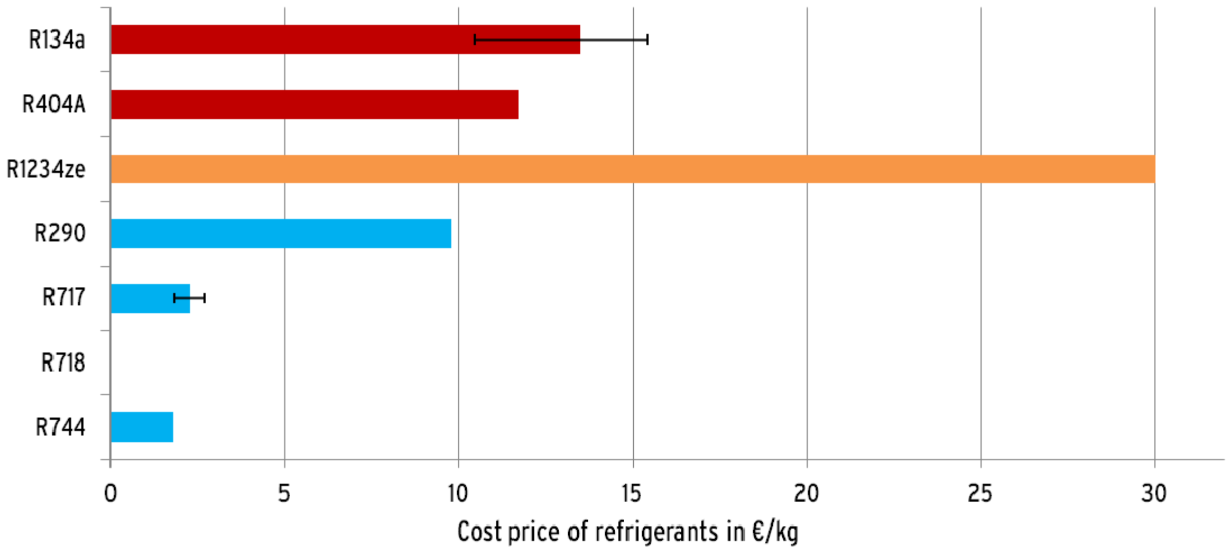
Saving of energy costs due to a larger system efficiency

The results of the theoretical comparison of refrigerants (subsection 6.2.1, pp. 60ff) and the simulations of various systems showed that the refrigerants R717 and R723 yielded a better efficiency compared to R134a in similar systems. Compared to R404A and R410A, which have a larger volumetric cooling power than R134a, larger efficiency ratios are provides by refrigerants R290 and R1270. However, for R718 and R744 it is required to modify the system in order to obtain the energy efficiency ratios of R134a. Among those modifications, there is a double-staged compression with intermediate cooling for R718 and an internal heat exchanger, an expansion machine or an ejector for R744.

Significant energy cost savings are expected only for many full-load hours due to the slightly larger energy efficiency ratios. However, operators are more and more open-minded towards natural refrigerants and efficient operation modes. The latter one can be achieved with an optimised control scheme.

Refrigerant costs

In order to evaluate the reduction of costs for refrigerants, those relevant in Germany are presented in Figure 123. The specific costs of R717 and R744 are smaller than those for common HFCs by one order of magnitude, whilst for R718 there are several orders of magnitude. However, R290 has low cost advantages only. Compared to the HFC R1234ze, the costs of R290 are approx. 30 %.



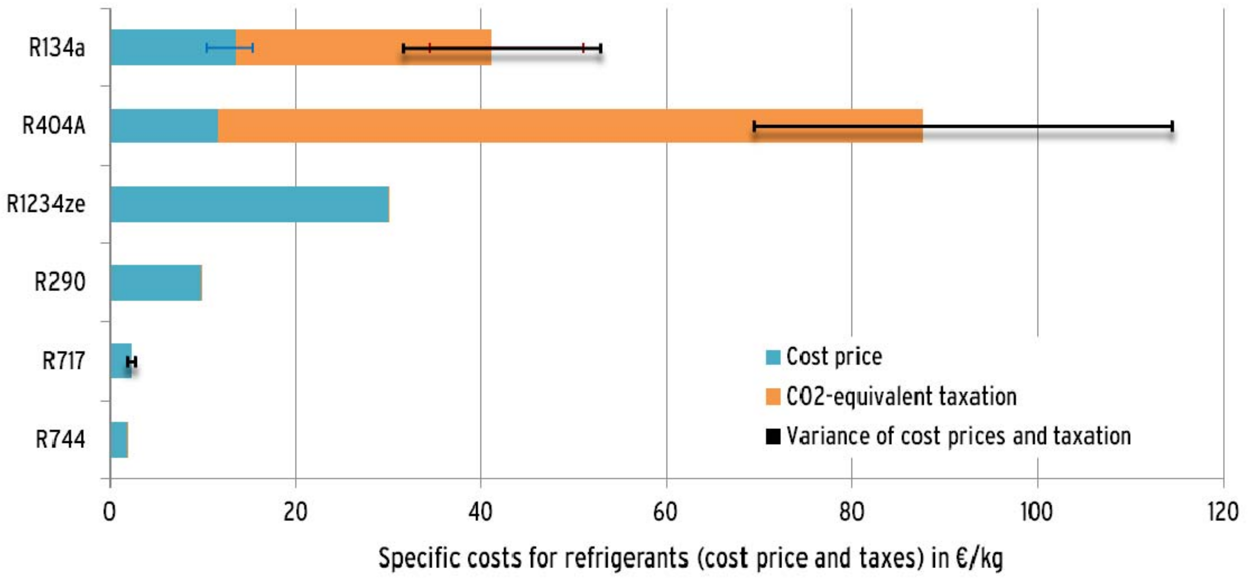
The cost prices have been enquired from three wholesale enterprises by ILK Dresden between January and December 2012.

Figure 123: Cost prices for refrigerants

Based on Figure 118, air conditioning units with a power below 10 kW are allowed to have 1.5 kg refrigerant. Substituting R404A with R290 yields a cost reduction potential of 3 €. Larger savings (approx. 500 €) can be obtained when substituting R134a with R717 in a chiller with 50 kg refrigerant. However, the cost advantage of natural refrigerants compared to HFCs is small compared to the investment costs. Due to the presently large costs for unsaturated HFCs (e.g., R1234ze), larger saving potentials arise, whereby a sufficiently large economical advantage cannot be expected here.

Figure 124 illustrates the total specific costs of refrigerants, including their individual CO_2 equivalent taxation. The tax rates are calculated as a mean value of the tax rates in those countries having legal requirements already. Furthermore, the variance of the cost prices (blue error bars), tax rates (red error bars), and the overall variance of the total price (black error bars)is indicated.

Due to the GWP-related taxation, the costs for R404A increase by factor 6, for R134a by factor 2 to 3. Therefore, cost advantages of natural refrigerants arise. For instance, considering an air conditioning unit with a refrigerant charge of 1.5 kg, the saving is approx. 100 €. Especially for serial products, this is a strong incentive. Moreover, for a chiller with 50 kg of R134a saving potential is approx. 1,250 €, which is not negligible.



The cost prices have been enquired from three wholesale enterprises by the ILK Dresden between January and December 2012. The CO_2 equivalent taxation has been calculated according to the legal requirements presented in subsection 9.2.2.

Figure 124: Specific refrigerant costs comprising cost prices and CO_2 equivalent taxation

Economical advantages originating in the specific refrigerant properties

Economical advantages due to the specific properties arise mainly for R744. Given the large pressure and its resulting large volumetric cooling power, small pipe cross-sections and compressor volume are realisable. There are special cost advantages for direct evaporation systems with long pipes (e.g., multi-split systems, industrial refrigeration systems).

Present developments of R718 systems allow storing cold in terms of a pumpable water/ice slurry. For the case of a direct evaporation ice production (the refrigerant is the same as the storage material), large energy efficiency ratios can be obtained also due to the relatively large evaporation temperatures. These advantages can be utilised for cases with relatively low cold temperatures of approx. -2 to 6 °C. Charging the storage over night allows employing advantageous condensation conditions and possible energy excesses in the electrical power supply system.

Support for investment costs of commercial refrigeration systems

The German Ministry for the Environment, Nature Conservation, and Nuclear Safety has approved a guideline for supporting commercial refrigeration systems as of 1st January 2009 (BMU & BAFA 2009)[14]. Herewith, measures and systems are supported, which lead to electrical energy savings of more than 35 % of the present energy demand. The basic properties and restrictions for the support are provided as follows:

[14] A revised version of the guideline entered into force 1st January 2014 (see http://www.bafa.de/bafa/de/energie/kaelteanlagen/index.html; in German).

- **Measures on existing installations** are supported with the requirement of an **energy demand reduction of more than 35 %**. The **electrical energy demand** has to exceed **150,000 kWh/a** before beginning the measure. The **support** of the net investment costs is **15 %**. A **bonus** of **additional 10 % is awarded for systems with R717, R744, or non-halogenated refrigerants** (i.e., all presently common natural refrigerants).
- **New installations** are supported **only** if they are **operated with natural refrigerants**. The overall efficiency has to be proven by the manufacturer or an independent service provider. The estimated electrical energy demand has to be larger than 100,000 kWh/a and the annual costs for electrical energy (including power price) exceeding 10,000 €/a. The **support** of the net investment costs is **25 %.**
- For the case refrigeration system **waste heat utilisation** (cooling, condensation) within the enterprise, e.g., for tap water heating, an **additional bonus of 10 %** of the net investment costs is possible.
- Further requirements are concerned with energy meters, the maintenance, and insulation materials for cooling rooms (no utilisation of green-house gases)

Since the support is based on the net investment costs of the entire refrigeration system (including cold distribution and fan coils), the additional costs for systems with natural refrigerants mentioned above can be covered completely.

The minimum requirements of new installations demand for large powers depending on the field of application. Chillers for air conditioning with approx. 500 full-load hours per year require a nominal cooling power of 200 kW, and industrial refrigeration systems require approx. 40 kW at a degree of capacity utilisation of approx. 30 %. A support of smaller systems might improve the development of chillers with natural refrigerants for air conditioning.

The guideline demand for a separate energy meter is positive, since it provides information about the cooling generation efficiency.

Summary

A major drawback of refrigeration and air conditioning systems with natural refrigerants are the relatively large investment costs. These originate from safety-related requirements, specific refrigerant properties, and the lack of series production. The present economical advantages are considered small and are not sufficient for the financial amortisation of the investment. However, due to the investment support for larger industrial refrigeration systems from 2009, an important measure for generating a larger market for systems with natural refrigerants. The temporally limited extension to smaller systems is advisable in order to increase the offers of series products with natural refrigerants on the market.

Recommendations

The investment support for new and existing systems of small power with natural refrigerants should be extended. A proof of the energy efficiency analogue to the heat pump support shall be included.

GWP-related taxation of refrigerants as a transparent measure as an additional incentive for larger system tightness and the utilisation of natural refrigerants

Intensification of the required tightness checks of systems with HFC refrigerants

9.3 Ad- and absorption type refrigeration systems

9.3.1 Safety-related barriers

It has to be distinguished between ammonia/water AbCh, water/lithium bromide AbCh, and AdCh. The regulations in DIN EN 378 concerned with ammonia as refrigerant are presented in subsection 9.2.1 at pp. 157ff. Lithium bromide is classified as water hazard class 2, which does not lead to barriers for water/lithium bromide AbCh. The sorbents silica gel and zeolite do not have any hazard potential for humans and the environment. Hence, there are no barriers of any kind for adsorption type refrigeration systems.

9.3.2 Application-related barriers

Temperature-dependent system behaviour — availability of driving heat

Absorption type refrigeration systems have a system behaviour, which depends strongly on the temperature conditions of external media. Therefore, the cooling generation has demands for the re-cooling and the heat supply. There are multiple limitations:

- Heat sources have to have sufficient temperature levels and heat powers.
- The heat supply has to be available for the estimated timespan of ACh utilisation. Discontinuities during a day are coverable by heat storages. However, discontinuities over a longer timespan, and uncertainties related to the prospective development of the cooling demand or the heat supply are a strong limitation.

Considering the waste heat of CHP motors, which can be extracted easily, the temperature level is between 85 and 90 °C. In order to obtain large heat ratios for the building air conditioning it is advantageous to have chilled water temperatures between 10 and 14 °C. However, this mode of operation requires a specialised cold distribution within the building.

Recommendation

The design of air conditioning systems with larger supply temperatures is an important requirement for the ecological and economical operation of ACh. Since larger temperatures have also advantages for CRP systems, the dissemination of systems such as core activation or chilled ceilings is necessary. Especially for public buildings, new technologies can act as an example. The incorporation of new information into education (studies of building services) is required.

Space requirements

Since ad- and absorption type refrigeration systems have a higher number of heat exchangers, and due to the utilisation of water as refrigerant (AdCh, water/lithium bromide AbCh) larger cross sections are necessary, and larger volumes are required compared to CRP systems. Moreover, there is a demand for larger re-cooling units and additional hydraulics for the hot water pipes.

Recommendations

The reduction of the heat exchanger construction volume should be supported: e.g., by new coating technologies of the sorbent on the heat exchanger surface for adsorption and surface structures for absorption.

The reduction of additional system effort can be realised by directly air-cooled ACh or smart control of the re-cooling. There is research on this field with a large potential. A reduced system effort yields a reduced additional electrical energy demand.

Efficiency and limits of application

There are clear limits of the heat ratios obtainable for ammonia/water, water/lithium bromide, and water/lithium chloride systems. Their maximum heat ratios are 0.6, 0.7, and 0.7, respectively. The limits for the refrigerant-absorbent combinations H_2O/LiBr and H_2O /LiCl are due to:

- Water as refrigerant (chilled water supply temperature > 0°C)
- Space of solubility of water/salt mixture (crystallisation limits → limitations of temperature difference between chilled, cooling, and heating water)

A circuit in which the condenser is substituted by an absorber and the evaporator by a desorber, allows for chilled water temperatures <0°C (Richter 2008; Richter et al. 2011). Beside the well-known refrigerant-absorbent combinations, there are ionic liquids which are considered promising sorbents for absorption type refrigeration systems and heat pumps (Richter 2008; Richter et al. 2011). Ionic liquids are organic, at ambient conditions liquid salts with negligible vapour pressure. There is a vast number of possible combinations (Merck 2005). First investigations with an absorption type refrigeration system with water as a refrigerant and 1-Ethyl-3-Methylimidazolium-Ethylsulfate as an absorbent show promising results, which are, however, still behind those of H_2O/LiBr. The reasons are, e.g., the much larger viscosity and interfacial tension (Radspieler & Christian Schweigler 2010).

Beside new refrigerant-absorbent combinations, multi-staged and multi-effect systems provide the opportunity for adjustments according to the ambient temperature conditions and towards a larger energy yield.

Recommendations

Support of research and development on the fields of:

a) **Absorption type refrigeration systems with ionic liquids, and**

b) **Multi-staged and multi-effective absorption type refrigeration systems.**

Lack of information

There is lots of information available about chillers and compressors in data sheets or web pages of manufacturers. This information comprises the system characteristics (cooling power, energy efficiency ratio) depending on the ambient conditions (ambient temperature, required cooling temperature) and is provided by means of diagrams, tables, and sometimes in terms of polynomials.

However, information about the ad- and absorption type system behaviour aside the set point is difficult to find, if ever. Both the strong dependence on the ambient conditions and the lack of information leads to a limiting uncertainty. Moreover, experiences of operators are difficult to find for other operators or builders.

There are two reasons for the lack of information:

- Larger demand for measurements due to:
    1. Three rather than two fluid flows
    2. Larger thermal inertia compared to CRP system (longer set times until reaching steady-state)
- More complex physical relation, which can be approximated by polynomials only insufficiently

Moreover, the set points of the system are individually defined by manufacturers. Hence, a comparison is always very limited.

The model of the characteristic equations provides the opportunity to calculate the behaviour of systems based on few measurement points (Christian Schweigler et al. 2003; Felix Ziegler 1998). Although there are differences for identically constructed systems (Neumann et al. 2011), this approach provides important information for system engineers and operators. More information by the manufacturers about the system behaviour at different temperatures is desirable. An example for a suitable visualisation is the so-called power nomogram (Figure 125).

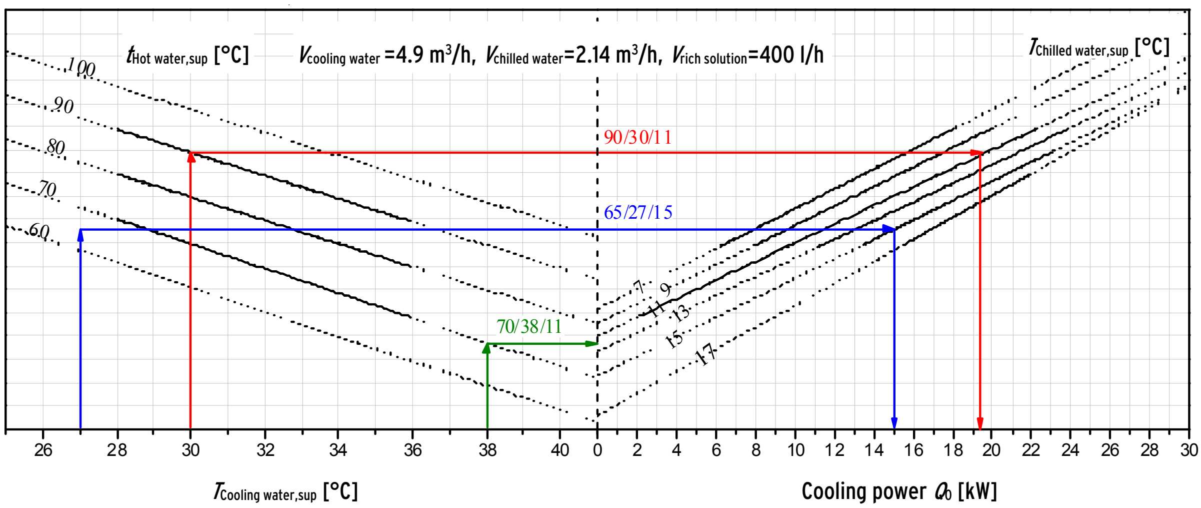


Figure 125: Power nomogram of water/LiBr absorption type refrigeration system (Safarik et al. 2010)

9.3.3 Economical barriers

A large barrier is the higher investment costs compared to CRP systems of the same power. The larger costs originate from:

- Larger material demand due to two additional heat exchangers (absorber, generator), and an additional dephlegmator (for ammonia/water systems only)

- The re-cooling requires a 2-3 times larger cooling power than CRP systems, with implications for the re-cooler, cooling water pipes, and cooling water pump.
- Additional re-cooler usually required. There are only few developments of directly air-cooled ACh available.
- Additional pipe system for hot water or steam
- Required infrastructure for the connection of waste heat/connection to district heat
- Depending on the heat supply → backup systems

The following investment advantages are usually not compensating the above-mentioned additional costs:

- No mechanical compressor and its power electronics (e.g., frequency converter)
- Lower electrical connection power (share: power price)

The following points are important for the economical amortisation (for details see subsection 7.4.3.3):

- Low heat generation costs
- Minimal electrical energy demand for pump (see electrical EER)
- Optimised mode of operation

The last point comprises the interrelation of heat supply, cooling demand, re-cooling, and possible alternatives for cooling generation.

9.4 Summary of recommendations

9.4.1 Legal requirements

a) The reduction of the direct TEWI emission by limitation of HFC application is possible for many applications. The new F-gas regulation comprises effective measures for the reduction of the direct TEWI emissions. For an even stronger national regulation, additional measures could be implemented.

b) Measures for the reduction of indirect TEWI emissions are mainly covered by the Eco-Design directive on European level. The ordinances being important for the refrigeration and air conditioning technology (ENTR Lots 1 and 6) are presently under amendment. The resulting minimum requirements are concerning the energy efficiency and seasonal energy efficiency ratio, and the design of control units for the air handling technology.

c) Intensification of tightness checks of HFC systems: A variety of allowed methods is not able to detect leakages early. Moreover, the application of indirect leakage detection systems (by measurement of system parameters and their analysis) shall be mandatory if such a system is available.

d) Implementation of a GWP-related taxation of refrigerants: Taxes above 20 €/t CO_2 equivalent yield a useful incentive. The present price for certificates for CO_2 emissions is approx. 4.20 €/t CO_2 equivalent (as of March 2013). With an implementation of the taxes as a deposit, efforts for increasing the system tightness and an entire, correct disposal are honoured.

e) Implementation of an obligatory record of cooling and driving powers of refrigeration systems with a large capacity: Proof of the obtained energy efficiency ratios and seasonal energy efficiency ratios. Moreover, these data are the basis for design and evaluation of measures for saving energy.

9.4.2 Support of research — major points

a) Hybrid refrigeration cascades of absorption and compression stages with the objective of ACh operation in the optimal set point and the utilisation of the CRP advantages at small temperature differences.

b) Compact air conditioning systems with natural refrigerants (especially R290)

c) Compression cascades with natural refrigerants in high- and low-temperature stages

d) Leakage detection system based on process data and pre-calculation of expected states of operation depending on ambient conditions

e) Support of developments of refrigerant charge reduction

f) Implementation of free cooling in refrigeration systems

g) Refrigeration systems utilising the available technical work of R744 during expansion

h) Cold storage systems for decoupling energy demand, supply and utilising good ambient conditions for cooling generation

i) Field tests for actual leakages, their frequency of occurrence and causes

j) Monitoring of refrigeration systems with HFC and natural refrigerants and comparison of the energy efficiency in real system operation

k) Absorption type refrigeration systems with new refrigerant-absorbent combinations based on ionic liquids

l) Multi-staged and multi-effect absorption type refrigeration systems

m) Reduction of construction size of thermally driven refrigeration systems, e.g., by new shapes of heat exchangers

n) Open and decentralised sorption processes for simple implementation into rooms, also without connection to air ducts

9.4.3 Support of the cost effectiveness

o) Exploit the legal opportunities in order to monetarily reflect the environmental aspects of HFCs for the operator (see legal requirements for intensification of tightness checks and GWP-related taxation of refrigerants)

p) Support of the investment into systems with natural refrigerants; Large-scale systems shall have a monitoring system by legal requirement.

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11 Appendix

11.1 Leakage rates

Further details on leakage rates from the literature are presented in addition to those values provided already in the TEWI calculation in subsection 7.2.2.

11.1.1 Field test at commercial refrigeration systems

The “Forschungsrat Kältetechnik” has investigated commercial refrigeration systems regarding their tightness and leakages (Birndt 1999). 64 commercial refrigeration systems in the states Hesse and Saxony have been investigated in 1999. The systems had refrigerant charges between 0.7 and 360 kg. All systems have been installed between 1990 and 1999 or have been changed from PFC to HFC refrigerants. For the leakage tests, halogen leakage search devices and selective gas detectors have been applied.

Table 41 provides the averaged leakage rates itemised by refrigerant charge, whereas

illustrates the leakage rates for particular systems. It can be observed that the results differ tremendously among the system (logarithmic scale for the leakage rate). For instance, approx. a third of the systems do not have any detectable leakages, whereas some systems have a total refrigerant loss within one year. The particular systems are itemised according to time-dependent leakage classes (“FKT-Stufenplan”, ChemKlimaSchutzV, see subsection 11.1.2) in Table 43.

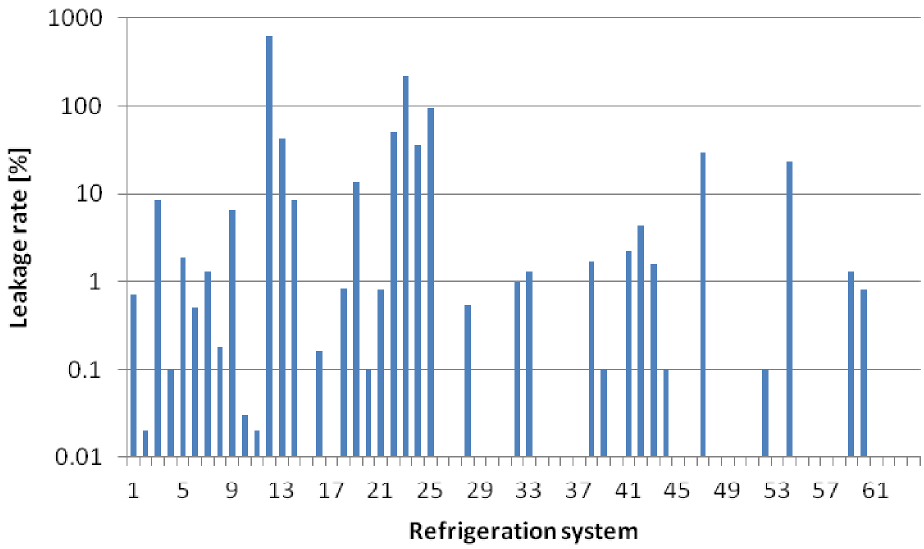

Figure 126: Leakage rates of particular systems

Table 41: Average leakage rates classified by refrigerant charge

Mass	<10 kg	10...100kg	>100kg
Number	25	18	11
Averaged leakage rate	18.7 %	38.5 %	1.3 %

Table 42: Classification of the results (Birndt 1999) according to time-dependent leakage classes ("FKT-Stufenplan", ChemKlimaSchutzV)

	Complying with stage 1			Comply with stage 2			Comply with stage 3		
Mass	<10kg	10…100kg	>100kg	<10kg	10…100kg	>100kg	<10kg	10…100kg	>100kg
Yes	19	14	10	19	13	10	18	13	8
No	6	4	1	6	5	1	7	5	3

11.1.2 Time-dependent leakage classes ("FKT-Stufenplan")

Due to the study of (Birndt 1999), the "Forschungsrat Kältetechnik" has proposed a time-dependent leakage classes plan for refrigeration systems assembled from components (assembly systems) at the installation site (Kruse & Wobst 2004). For refrigeration systems assembled at the manufacturer's site (e.g., chillers), there are stronger tightness requirements. The values being proposed have been incorporated in the German ordinance "Chemikalien-Klimaschutzverordnung" (Chemicals Climate Protection Ordinance) (ChemKlimaschutzV 2007) which was approved by the German national parliament in 2007.

Table 43: Acceptable leakage rates according to the "FKT-Stufenplan"; incorporated into ChemKlimaSchutzV (ChemKlimaschutzV 2007; Kruse & Wobst 2004).

Stage	Time limit	Refrigerant charge [kg]	Leakage rate [%] Assembly systems	Leakage rate [%] Chillers
1	by 06/2005	< 10	10	
		10 … 100	8	
		> 100	4	
2	by 06/2008	< 10	6	
		10 … 100	4	2
		> 100	2	
3	by 06/2018	< 10	3	
		10 … 100	2	
		> 100	1	

11.2 Details for modelling refrigeration systems

11.2.1 Refrigeration systems for building air conditioning

Table 44: Additional information for modelling of multi-split VRF systems

Component	Value/assumption
• Outdoor unit	
Refrigerant	• R410a
Condenser	• Minimal temperature difference (cooling air and refrigerant): 4 K • Sub-cooling of the refrigerant: 5 K • Ventilator: $P_{el,nominal}$ = 625 W, variable volume flow rate
• Indoor unit	
Evaporator/ Convector	• Evaporation temperature: 6 °C • Minimal temperature difference (cooling air and refrigerant): 4 K • Assumed superheat of the refrigerant: 7 K • Preferably realistic change of state of air during convection, with weighted mean values from two extreme models: - Model 1: change of state along a straight line between supply state and dew point of t_0 - Model 2: after solely sensible cooling until dew point: dehumidification along dew line - weighting of both dehumidification models with a TRNSYS convector model • $\varsigma_{air,\ nominal}$ = 5000 m³/h, $P_{el,\ nominal}$ = 410 W, constant volume flow rate • All indoor devices are comprised to a single large on (single-zone model)
Refrigerant pipes	• Pressure and temperature variations due to frictional losses in gas suction pipe considered only (length: 50 m, diameter: 33.7 mm); negligible losses in liquid pipe
• Part-load behaviour	
Compressor	• Constant compressor power below 25 % of nominal cooling power • Part-load behaviour validated according to DIN V 18599-7
Ventilator of condenser	• Volume flow rate of condenser ventilator directly proportional to cooling power, but constant below 50 % of nominal cooling power • Part-load behaviour validated according to DIN V 18599-7
• Control	
-	• Setting of room temperature to maximum value by control of refrigerant mass flow rate (for cooling cases)

Table 45: Additional information for modelling of chillers (with/without storage, convector/chilled ceilings)

Component	Value/assumption
• Chiller	
Refrigerant	• System with storage: R410a • System without storage: R410a, R290, R134a, R717, R718, and R1234yf
Compressor	• for R718 double-staged compression with internal intermediate cooling
Condenser	• for system with storage minimal condensation temperature 15 °C (avoiding unrealistic liquefaction temperatures during the night whilst charging the storage at low ambient temperatures) • Ventilator: $P_{el, nominal}$ = 625 W, variable volume flow rate
Evaporator	• Nominal chilled water supply temperature ($t_{VL,KWS}$): 14 °C for chilled ceilings, 10 °C for convectors (without storage or shunted out storage), 6 °C (charging storage) • Variable return temperature • Evaporation temperature: $t_{sup,Ch}$ – 2 K • Superheat of the refrigerant: 0 K
• Chilled water storage	
Storage	• Stratified storage, outside location • Volume: priority variant (from comparison) 7.5 m^3 • Losses: 0.8 W/m^{2}K (insulation: approx. 50 mm mineral wool)
Pump	• $\mu_{water, nominal}$ = 6000 kg/h, $P_{el, nominal}$ = 150 W, delivery height: 5 m (constant volume flow rate)
• Indoor unit (chilled water circuit)	
Convector	Air change of state with TRNSYS convector model (Type 52b, (TRNSYS 2004)) • $\varsigma_{air, nominal}$ = 5000 m^3/h, $P_{el, nominal}$ = 410 W, constant volume flow rate • All indoor devices are comprised to a single large on (single-zone model)
Chilled water pipes	• Pressure drop in chilled water circuit neglected (influence on cooling effectiveness negligible, electrical energy demand for driving flow by pump considered) • Heat supply by pump considered via pump energy efficiency ratio • Cooling losses through pipes neglected (assumptions: internal laying, good insulation)
Pump	• $\mu_{water,nominal}$ = 6000 kg/h, $P_{el,nominal}$ = 550 W, delivery height: 20 m, variable volume flow rate (identical for systems chilled ceiling/convectors)
• Part-load behaviour	
Compressor	• Multi-staged compressor network for entire office building, simplified by continuous part-load of 0...100 % • Part-load behaviour validated according to DIN V 18599-7
Ventilator of condenser	• Analogous to multi-split VRF systems
• Control	
System without storage	• Setting of room temperature to maximum value by control of refrigerant and chilled water mass flow rates (for cooling cases) • Supply temperature for chilled ceilings adjusted to dew temperature in order to avoid condensation

Component	Value/assumption
System with storage	• Setting of the room temperature analogue to systems without storage (control of cooling medium mass flow rate and adjustment of cooling generation power/utilisation of the storage) • Control with three modes: - Mode 1: charging storage (until 2 K below nominal supply temperature, in the morning prior to office cooling) - Mode 2: discharging storage (for better storage utilisation supply temperature 4 K above nominal supply temperature) - Mode 3: shunting out storage (e.g., if storage is empty.)

Table 46: Additional information for modelling a heating, ventilation, and air conditioning system with DEC system

Component	Value/assumption
• Regenerator unit	
Pump for concentrated aqueous solution	$P_{el,nominal}$ = 0,8 kW, for both system sizes under consideration (SS1/SS2), constant volume flow rate (menerga 2010)
Regenerator	Specific power demand for regeneration: approx. 1.1 $kW_{thermal}$/(kg water/h), at hot water conditions (see CHP)
Ventilator for regeneration air	$P_{el,nominal}$ = 0.8 kW (SS1) and 1.0 kW (SS2), constant volume flow rate (menerga 2010)
Pump for diluted aqueous solution	$P_{el,nominal}$ = 0.8 kW, for both system sizes under consideration (SS1/SS2), constant volume flow rate (menerga 2010)
• Cooling and desiccation unit	
Evaporation cooling	• pump: $P_{el,nominal}$ = 0.6 kW, constant volume flow rate (SG1/SG2), constant volume flow rate • water demand: 8…10 kg/h per (1000 m^3_{air}/h), assumption: 9 kg/h per (1000 m^3_{air}/h)
Absorber	• For the case of dehumidification: dehumidification of fresh air until nominal humidity (assumption: dehumidification power always sufficient) • Nominal humidity of fresh air: 11.0 or 9.0 g water/kg air, depending on variant
Supply and exhaust ventilators	$P_{el, nominal}$ = 5.5 kW, $\varsigma_{air, nominal}$ = 6100 m^3/h (SS1) and $P_{el, nominal}$ = 7.2 kW, $\varsigma_{air,nominal}$ = 8300 m^3/h (SS2), variable power (menerga 2010)
• Building	
	• No system components considered in the air conditioned room
• Part-load behaviour	
Regenerator	• Thermal power demand depending on dehumidification demand (see above) • Electrical power demand of pumps and regenerator ventilator proportional to dehumidification power, assuming cyclic regeneration with surge drum
Supply and exhaust ventilators	• Volume flow rate depending on cooling demand adjustable (considering minimal air renewal rate of 0.4)
Pump for evaporative cooling	• Water volume flow rate directly proportional to cooling power
• Control	

Component	Value/assumption
-	• Setting the room temperature to fixed maximum value by controlling air volume flow rates of fresh and exhaust air ventilators (for cooling case) • Following operation states (OS) are defined: - OS 0: system completely deactivated (night, weekend) - OS 1: system is operated with heat recovery at room temperatures below 22 °C (at increasing temperatures) and 20 °C (at decreasing temperatures), air renewal rate 0.4, heat recovery ratio 0.8 - OS 2: free cooling at room temperatures above 22 °C (at increasing temperatures) until max. room temperature and 20 °C at decreasing temperatures, air renewal rate 0.4 - OS 3: indirect evaporative cooling, maximum temperature 22 °C, air volume flow rate 448 m^3/h (air renewal rate 0.4) until nominal volume flow rate of fresh and exhaust air ventilator → control via room temperature by PID controller - OS 4: as OS 3, but with dehumidification if humidity in office is above 11.5 g water/kg air, desiccation of air until 11 g water/kg air (second variant 9.5 g water/kg air to 9 g water/kg air)
• CHP	
	• Supply/return temperatures of hot water for regenerator is assumed to be suitable for heating demand of regeneration (70/60 °C)
• Balance	
Energy demand	• Thermal energy demand of regenerator and water demand for adiabatic evaporative cooling is balanced

11.2.2 Systems for industrial refrigeration

Table 47: Additional information for modelling a chiller for industrial refrigeration

Component	Value/assumption
• Chiller	
Refrigerant	• R410a, R290, R134a, R717, and R1234yf
Compressor	• for R718 double-stage compression with internal intermediate cooling
Condenser	• minimal temperature difference between cooling air and refrigerant: 4 K • for systems with storage: minimal condensation temperature 5 °C (for R718) and 15 °C (for other refrigerants) • Ventilator: $P_{el,\ nominal}$ = 12.5 kW, variable power
Evaporator	• Evaporator temperature: -1 °C • Superheat of refrigerant: 0 K
• Control	
-	• Adjustment of compressor and ventilator power to ambient temperature for generation of a constant cooling power

Table 48: Additional information for modelling an absorption type refrigeration system for industrial refrigeration

Component	Value/assumption
• Re-cooler	
Ventilator	Ventilator: $P_{el,nominal}$ = 97.6 kW, $\varsigma_{air,\ nominal}$ = 200 m^3/s, variable volume flow rate, derived from (AIA 2008)
Adiabatic pre-cooling	• At ambient temperatures above 16 °C

Component	**Value/assumption**
Pump	• $P_{el, nominal}$ = 0.25 kW, delivery height: 20 m, constant volume flow rate
• Absorption type refrigeration system	
refrigerant-abs. combination	• Ammonia/water
Characteristic values	• Heat ration: 0.58 • Driving temperature difference: 11.8 K • (data sheets, empirical values)
Temperatures and volume flow rates	• Hot water: temperatures: 90/83 °C, volume flow rate: 105.5 m^3/h • Cooling water: temperatures: 34/39 °C, volume flow rate: 234.7 m^3/h • Chilled water: temperatures: 1.5/6.5 °C, volume flow rate: 86 m^3/h • (chilled water temperature 0.5 K below nominal supply temperature for cold consumer (+2 °C) for compensating slight variations)
Pump	• Assumed power for constant volume flow rate: - hot water pump: $P_{el, nominal}$ = 4.8 kW, delivery height: 10 m - cooling water pump: $P_{el, nominal}$ = 10.7 kW, delivery height: 10 m - chilled water pump: $P_{el, nominal}$ = 2.0 kW, delivery height: 2 m
Pipes	• Electrical energy effort for overcoming pressure drop considered • Heat entry from pump considered via pump energy efficiency ratio • "cold losses" through pipe neglected (assumption: short and well insulated pipes)
• Storage	
-	• For compensation of fluctuations in control (e.g., for starting the adiabatic pre-cooling in re-cooler) • Volume: 2 m^2 • "cold losses" negligible (since < 0.05 % of cooling power)
• Control	
-	• 2 control variants: - Variant 1: constant cooling water temperature by controlling the volume flow rate of the re-cool ventilator depending on ambient temperature → constant chilled water temperature - Variant 2: variable cooling water temperature depending on the ambient temperature (minimal allowed cooling water temperature 6.5 °C), below ambient temperatures of 2.5 °C → free cooling of chilled water via re-cooler • Slight differences of nominal temperature compensated by storage and controlled return flow mixture (between storage and customer)
• Balance	
-	• Beside balance of electrical energy also thermal energy consumption and water consumption

Table 49: Additional information for modelling a direct evaporation system for industrial refrigeration

Component	**Value/assumption**
• Chiller	
Refrigerant	• R723, and R134a
Condenser	• Minimal condensation temperature: 15 °C
Evaporator	• Evaporator temperature: 2 °C • Superheat of refrigerant: 0 K
• Control	
-	• Analogue to chiller for industrial refrigeration

Table 50 : Listing of investment, maintenance, and disposal costs of particular systems for building air conditioning (for VRF and chiller systems, office floor area 400 m^2, nominal cooling power 25 kW, except for DEC system: 57.9 kW, including VAT)

	Unit	VRF	Reference/Comment	Chiller			Reference/Comment
				Convector	Convector + storage	Chilled ceiling	
Investment costs							
Outdoor unit	EUR/kW(cold)	316	Cost function of [Wiemken], individual tender	240	240	240	Cost function of [Wiemken]
Indoor unit, chilled ceiling	EUR/m^2(build)	31	Individual tender	31	31	87	Individual tender, [dietrich2011]
Cold circuit	EUR/m^2(build)	29	Individual tender	29	29	29	Analogous to VRF
Collector	EUR/m^2(coll.)	-		-	-	-	
Storage	EUR/m^2	-		-	830	-	Cost function of [Wiemken]
Re-cooler	EUR/kW(rc.)	-	Integrated into outdoor unit	-	-	-	Integrated into outdoor unit
Installation device	EUR/m^2(build)	2.35	Individual tender	2.35	2.35	2.35	10 % of price for device
Installation circuit	EUR/m^2(build)	21	Individual tender	21	21	21	Analogous to VRF
Total investment costs	EUR/m^2	40,362	See above and [Kunicic2005]	37,655	43,878	60,981	See above and [Kunicic2005]
lifetime	a	15	[VDI 2067, DIN EN 13779:2004, DIN EN 15459]	15	15	15	[VDI 2067, DIN EN 13779:2004, DIN EN 15459]
Market interest rate (without inflation)	%	6	Internet research	6	6	6	Internet research
Annuities	**EUR/a**	**4,156**	[VDI 2067]	**3,877**	**4,518**	**6,279**	[VDI 2067]
Annual maintenance costs							
Related to investment costs	%/a	4.00	[DIN EN 15459], individual tender	4.00	4.00	-	[DIN EN 15459], individual tender
Absolute value	**EUR/a**	**1,614**		**1,506**	**1,755**	**1,506**	
Disposal costs							
Related to investment costs	%	2	Experience value of manufacturer	-	-	-	
Absolute value	**EUR/a**	**54**		**54**	**54**	**54**	Analogous to VRF

Table 51: Listing of investment, maintenance, and disposal costs of particular systems for building air conditioning (for ACh and DEC systems, office floor area 400 m², nominal cooling power 25 kW, except for DEC system: 57.9 kW, including VAT)

	Unit	ACh FW (A_{col} = 90 m²) Chilled ceiling	Reference/Comment	DEC	Reference/Comment
Investment costs					
Outdoor unit	EUR/kW(cold)	525	Cost function of [Wiemken]	1,314	Individual tender
Indoor unit, chilled ceiling	EUR/m²(build)	87	Individual tender, [dietrich2011]	-	Included into outdoor unit
Cold circuit	EUR/m²(build)	29	Analogous to VRF	62	Estimation manufacturer
Collector	EUR/m²(coll.)	229		-	
Storage	EUR/m²	947	Cost function of [Wiemken]	-	
Re-cooler	EUR/kW(rc.)	42	Cost function of [Wiemken]	-	
Installation device	EUR/m²(build)	3.3	10 % of device price	19.0	10 % of device price
Installation circuit	EUR/m²(build)	21	Analogous to VRF	33.3	35 % of cold circuit costs
Total investment costs	EUR/m²	95,752		108,415	
Lifetime	a	18	[VDI 2067]	15	[VDI 2067]
Market interest rate (without inflation)	%	6	Internet research	6	Internet research
Annuities	**EUR/a**	**8,843**	[VDI 2067]	**11,163**	[VDI 2067]
Annual maintenance costs					
Related to investment costs	%/a	-	Absolutely as VRF	-	Absolutely as VRF
Absolute value	**EUR/a**	**1,614**		**1,614**	
Disposal costs					
Related to investment costs	%	-		-	
Absolute value	**EUR/a**	**54**	Analogous to VRF	**54**	Analogous to VRF

Table 52: Listing of investment, maintenance, and disposal costs of particular systems for industrial refrigeration (nominal cooling power 500 kW, including VAT)

	Unit	KWS	Reference/Comment	Direct evaporator	Reference/Comment	ACh	Reference/Comment
Investment costs							
Device costs	EUR/kW(cold)	195	Incl. re-cooler, cost function of [Wiemken]	234	Incl. re-cooler, 20 % more expansive than chiller analogous to office cooling	530	Cost function of [Wiemken]
Storage	EUR/m²	-		-		1,546	Cost function of [Wiemken]
Re-cooler	EUR/kW	-		-		177/26	Individual tender (hybrid cooling tower/open wet cooling tower)
Installation	EUR/kW	20	10 % investment costs (analogous to DEC)	23	Analogous to chiller	50	10 % investment costs (analogous to DEC)
Total investment costs	EUR/m²	129,853		155,824		541,669	
Lifetime	a	15	[VDI 2067, DIN EN 13779:2004, DIN EN 15459]	15	Analogous to chiller	18	[VDI 2067, DIN EN 13779:2004, DIN EN 15459]
Market interest rate (without inflation)	%	6	Internet research	6	Internet research	6	Internet research
Annuities	**EUR/a**	**13,370**	[VDI 2067]	**16,044**	[VDI 2067]	**50,027**	[VDI 2067]
Annual maintenance costs							
Based in investment costs	%/a	4	[DIN EN 15459], indiv. tender	4	Analogous to chiller	-	Absolutely as VRF
Absolute value	**EUR/a**	**5,194**		**6,233**		**5,194**	
Disposal costs							
Based in investment costs	%	2	Experience value of manufacturer	2	Experience value of manufacturer	2	Experience value of manufacturer
Absolute value	**EUR/a**	**173**		**208**		**602**	