Method and guidelines to establish System Efficiency Index during field measurements on air conditioning and heat pump systems

Anna-Lena Lane, Jessica Benson, Lina Eriksson, Per Fahlén, Roger Nordman, Caroline Haglund Stignor, Klas Berglöf[1], Guy Hundy[2]

SP Technical Research Institute of Sweden

Resurseffektiva kyl- och värmepumpssystem

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Anna-Lena Lane, Jessica Benson, Lina Eriksson, Per Fahlén, Roger Nordman, Caroline Haglund Stignor, Klas Berglöf[1], Guy Hundy[2]

1 Klas Berglöf, working at ClimaCheck AB
2 Guy Hundy, working at IOR

Preface

This project has been led by SP Technical Research Institute of Sweden. The work has been done with the following project partners: Klas Berglöf at Climacheck, Sweden, Guy Hundy at IOR, England, Holger Kühl at Külanalyze Germany and Anthony Wootton, Ambient Control, England.

The project started in Waterloo Ville in April 2012 with an education on the ClimaCheck equipment and a kick off meeting for the participants. The work has continued with regular web meetings during project period and individual work done by SP and the project partners.

Thanks to the Swedish Energy Agency and Effsys+ for the funding and thanks to all participants for good collaboration.

Abstract

Rising energy prices create incentives and interest for property owners to influence their energy costs. In order to minimize costs, there is both an interest to maintain the original efficiency, but also to determine if the performance can be improved. It is therefore of interest to compare internal data with similar applications in order to find more profitable solutions.

The aim of the project have been to develop a methodology for evaluation of short term measurements for heat pump and air conditioning systems, used for reliable and energy optimization. This have been done by evaluation of the key performance indicator System Efficiency Index, SEI, earlier defined in previous works done by IOR in England and VDMA in Germany. [4-6]

The key performance indicator System Efficiency Index, SEI, describes the efficiency for a heat pump or air-conditioning system according to the best possible action for the case. Values for COP, including used and delivered power for the process, and temperatures to define the theoretical best action according to Carnot COP have to be measured.

Typically, COP is used as a key performance indicator for heat pump and air conditioning systems. The purpose of SEI and COP is different and they supplement each other. COP has a weakness as comparator as it is strongly dependent on operating conditions.

SEI answers the question how **efficient** the process is in a measured point. The measured value can be compared with values for other conditions. In this way SEI is a general indicator. The difference tells about the performance in the measured point according to ideal performance, other measured points or dimensioning data. It shows the potential for optimization and the quality of COP.

A measurement methodology to measure SEI for cooling and heating have been developed with system boundaries at four levels. For liquid/liquid units at system boundary one, including the refrigerant process, laboratory and field measurements have been evaluated. The result shows that SEI is a useful indicator for analysis of heat pump and air conditioning systems. A scale for identification of good performance in the system have been proposed, based on the measurement results.

Further development of the methodology will provide evaluation of more system types and establish the method and the scale.

Key words: Key indicator, System efficiency index, SEI, Field measurement, heat pump, air-conditioning

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Sammanfattning

Med ökande energipriser finns ett intresse hos fastighetsägare, att påverka sina energikostnader. Genom att underhålla och förbättra prestanda på anläggningen kan energikostnaden minska. Man behöver därför kunna jämföra prestanda för värmepumpar och luftkonditioneringsaggregat med liknande anläggningar för att hitta förbättringsmöjligheter.

Syftet med projektet har varit att utveckla en metod för utvärdering av korttidsmätningar på värmepumpar och luftkonditioneringsanläggningar, för att energioptimera och förlänga livslängden för anläggningarna. Detta har gjorts genom att utveckla nyckeltalet System Effektivitet Index, SEI, som tidigare definierats av IOR i England och VDMA i Tyskland. [4-6]

Nyckeltalet System Effektivitets Index, SEI, beskriver effektiviteten för en värmepump eller luftkonditioneringsanläggning jämfört med bästa möjliga prestanda för det aktuella driftfallet. SEI bestäms med mätvärden för COP, där levererad och använd effekt till processen ingår, samt temperaturer för att bestämma bästa teoretiska prestanda enligt Carnot COP.

Ett vanligt förekommande nyckeltal för värmepumpar och luftkonditionering är COP. Syftet med SEI och COP är olika och de kompletterar varandra. Svagheten med COP som jämförelsetal är att det är mycket temperaturberoende.

SEI svarar på frågan hur effektiv en process är i en mätpunkt. Mätvärdet kan jämföras med värden för andra temperaturer. På detta sätt är SEI ett allmänt nyckeltal. SEI beskriver prestanda för en mätpunkt jämfört med bästa möjliga prestanda. Mätpunkter kan därför jämföras med varandra och med dimensionerande data. Det visar kvaliteten på COP och möjligheter till förbättringar.

En metod för att mäta SEI för värme och kyla, med fyra olika nivåer på systemgränser har tagits fram i projektet. För vätska/vätska aggregat har lab- och fältmätningar utvärderats med systemgräns ett, där själva kylprocessen ingår. Resultatet av utvärderingen visar att SEI är ett användbart nyckeltal för att analysera värmepumpar och luftkonditioneringsanläggningar. Baserat på mätningarna har en skala för vad som är bra prestanda förslagits.

Vidare arbete med utvärdering av mätningar fler anläggningar och systemtyper skulle befästa metoden och skalan.

Contents

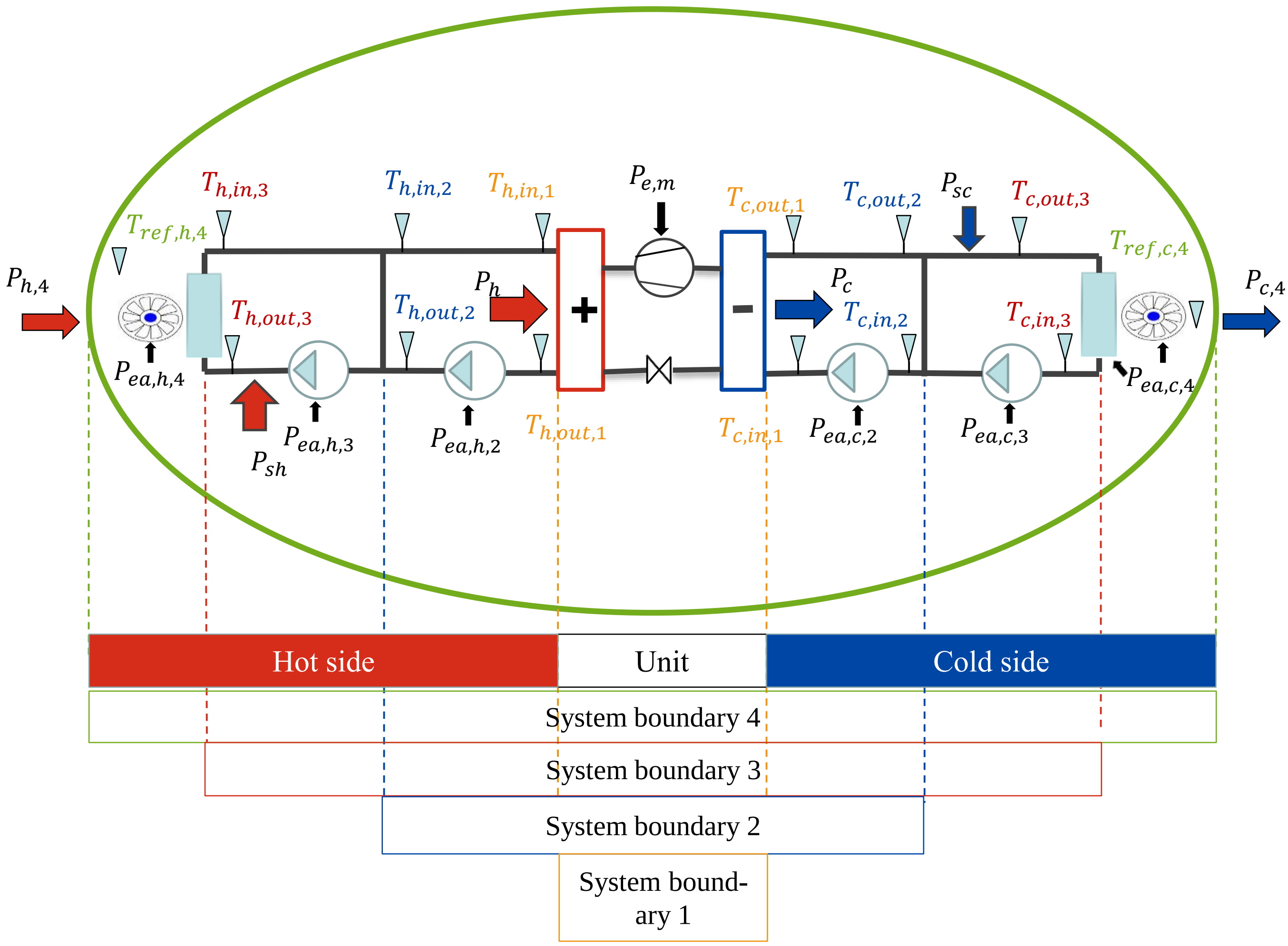

Figure 1 System boundaries and nomenclature

Nomenclature
Designations for quantities

Goodness numbers and efficiencies

$SEI_{i,j,(k)}$	System Efficiency Index. Index i indicates the mode of operation (h for heating, c for cooling), index j indicates the system boundary (1-4) on the cold side or the overall system boundary, index k can be used to indicate a different system boundary on the warm side.	[-]
COP_i	Coefficient of performance. Index i indicates the mode of operation (h for heating, c for cooling).	[-]
$COP_{i,j,(k)}$	Coefficient of systems performance. Index i indicates the mode of operation (h for heating, c for cooling), index j indicates the system boundary (1-4) on the cold side or the overall system boundary, index k can be used to indicate a different system boundary on the warm side.	[-]
η_n	Sub efficiencies index n (1-6)	[-]
SPF_Hn	Seasonal Performance Factor according to system boundary n=1-3, defined in the SEPEMO project	[-]

Temperature, pressure and flow

T_1, t_1	Temperature on the hot side of a Carnot cycle.	[K], [°C]
T_2, t_2	Temperature on the cold side of a Carnot cycle.	[K], [°C]
T_c, t_c	Condensing temperature	[K], [°C]
T_e, t_e	Evaporating temperature	[K], [°C]
$T_{ref,h,j}$, $t_{ref,h,j}$	Reference temperature for the hot side at system boundary *j. Measured or calculated value. (Also denoted with TsecW)*	[K], [°C]
$T_{ref,c,j}$, $t_{ref,c,j}$	Reference temperature for the cold side at system boundary *j. Measured or calculated value. (Also denoted with TsecC)*	[K], [°C]
$T_{h,in,j}$, $t_{h,in,j}$	Temperature of warm *(h)* heat transfer media (secondary warm *(h)* fluid) at the inlet to system boundary *j*.	[K], [°C]
$T_{h,out,j}$, $t_{h,out,j}$	Temperature warm *(h)* heat transfer media (secondary warm *(h)* fluid) at the outlet to system boundary *j*.	[K], [°C]
$T_{c,in,j}$, $t_{c,in,j}$	Temperature of cold *(c)* heat transfer media (secondary cold *(c)* fluid) at the inlet to system boundary *j*.	[K], [°C]
$T_{c,out,j}$, $t_{c,out,j}$	Temperature of cold *(c)* heat transfer media (secondary cold *(c)* fluid) at the outlet to system boundary *j*.	[K], [°C]
p	Pressure	[Pa]
$q_{v,i}$	volume flow rate for cold secondary fluid (*i=c)* and for warm secondary fluid *(i=h)*	[m^3/s]
$q_{m,R}$	Refrigerant mass flow rate	[kg/s]

Electric power and work

$P_{m,C}$	Mechanical power added to a Carnot cycle.	[W]
$P_{e,m,C}$	Electric power added to a Carnot cycle.	[W]
P_{com}	Shaft power to the compressor.	[W]
$P_{e,m}$	Electric power to the compressor motor.	[W]
$P_{ea,h,j}$	Auxiliary electric power used for fluid transport (pump, fan or other electric auxiliary) of the warm heat transfer media *(h)*, included in the *(j)* system boundary (1-4).	[W]
$P_{ea,c,j}$	Auxiliary electric power used for fluid transport (pump, fan or other electric auxiliary) of the cold heat transfer media *(c)*, included in the *(j)* system boundary (1-4).	[W]
$P_{e,sh}$	Power used to produce and distribute supplementary heating	[W]
$P_{e,sc}$	Power used to (produce) and distribute supplementary cooling, i.e. free cooling	[W]
W_C	Mechanical work added to a Carnot cycle.	[J]

$W_{e,C}$	Electric work added to a Carnot cycle.	[J]
W_{com}	Shaft energy to the compressor.	[J]
$W_{e,m}$	Electric energy to the compressor motor.	[J]
$W_{ea,h,j}$	Auxiliary electric energy used for fluid transport (pump, fan or other electric auxiliary) of the warm heat transfer media *(h)*, included in the *(j)* system boundary (1-4).	[J]
$W_{ea,c,j}$	Auxiliary electric energy used for fluid transport (pump, fan or other electric auxiliary) of the cold heat transfer media *(c)*, included in the *(j)* system boundary (1-4).	[J]

Heating capacity and energy

P_h	Heating capacity of the unit, included in the system boundary 1.	[W]
P_{sh}	Heating capacity of the supplementary heater on the hot side.	[W]
$P_{h,j}$	Heating capacity of the system, included in the *(j)* system boundary (1-4).	[W]
$P_{h,C}$	Heating capacity according to the Carnot cycle	[W]
Q_h	Delivered thermal heating energy from the unit, included in system boundary 1.	[J]
Q_{sh}	Delivered thermal heating energy of the supplementary heater on the hot side.	[J]
$Q_{h,j}$	Delivered thermal heating energy from the system, included in the *(j)* system boundary (1-4).	[J]

Cooling capacity and energy

P_c	Cooling capacity of the unit, included in the system boundary 1.	[W]
P_{sc}	Cooling capacity of the supplementary cooler of the cold side, for example free cooling.	[W]
$P_{c,j}$	Net cooling capacity of the cooling system, included in the *(j)* system boundary (1-4).	[W]
$P_{c,C}$	Cooling capacity according to the Carnot cycle	[W]
Q_c	Absorbed thermal cooling energy to the unit, included in the system boundary 1.	[J]
Q_{sc}	Absorbed thermal cooling energy of the supplementary cooler of the cold side.	[J]
$Q_{c,j}$	Net absorbed thermal energy to the cooling system, included in the *(j)* system boundary (1-4).	[J]

Other

η_{is}	Isentropic efficiency	[-]
c_p	specific heat capacity	[J/kg,K]
ρ	density	[kg/m^3]
f	Loss factor for heat from compressor	[-]

Subscripts

Subscripts

Heating		*Cooling*	
h	Heating or hot side	*c*	Cooling or cold side
sh	Supplementary heater	*sc*	Supplementary cooler
1	High temperature (conden-ser) side	2	Low temperature (evaporator) side
	General subscripts		
C	Carnot		
e	Electric		
a	Auxiliary		
m	Mechanical		
ref	Reference temperature		

Abbreviations, technical

COP	Coefficient of performance	[-]
SEI	System efficiency index	[-]
SPF	Seasonal Performance Factor	[-]

Abbreviations, other

EU European Union
IOR the Institute of refrigeration, UK
VDMA Verband Deutscher Maschinen- und Anlagenbau - German Engineering Federation

1 Introduction

Rising energy prices create incentives and interest for property owners to influence their energy costs. In order to minimize costs, there is both an interest to maintain the original efficiency, but also to determine if the performance can be improved. It is therefore of interest to compare internal data with similar applications in order to find more profitable solutions.

Modern measuring and control systems can deliver a lot of data at reasonable cost, but the knowledge of how the measured data is interpreted and transformed into useful and usable information to refrigeration technicians is poor or in many cases non-existent. To manually evaluate vast amounts of data can be very time consuming. A simple tool to check system performance in field is missing.

With simple ratios/key performance indicators, the performance variation in a system can be studied over time and changes can be identified to avoid running a bad performing system for a long time. Therefore, dramatic economic consequences can also be avoided. This check of performance indicators is necessary to maintain the system in the installed level of efficiency. Values of comparable the key performance indicators from many units or plants can be used for comparisons between different systems making benchmarking possible. Benchmarking will give answers about the need for and the profitability of investments, adjustments and regular follow-up operation to improve energy efficiency.

This project aims at defining a key performance indicator and developing a measurement and monitoring method to follow up performance of heat pump and air conditioning systems. The purpose of the method is to support energy optimization in these systems. In this work the SEI (System Efficiency Index) is evaluated. The key performance indicator SEI can give valuable information from one single instantaneous measurement in field. The SEI based on instantaneous or continuous field measurements will be used primarily for energy optimization but also for increasing reliability. The project is limited to study SEI for heat pump and air conditioning systems.

Previously, SEI has been used by the organisations IOR and VDMA [4-6] for dimensioning purposes of heat pump and air-conditioning systems. This project does not include SEI as a design tool. However, coordination with the design tools is necessary since the two are integrated and a lack of coordination can result in SEI from design cannot be compared with field measurements.

The report is divided into the following milestones:

- Description of key performance indicators (KPI’s) based on existing standards
- Definition of SEI, system boundaries and categorization of systems
- Guidelines for field measurements and monitoring
- Case studies with data analysis from real estate and residential heat pumps and air conditioning systems, where comparison of SEI becomes relevant.

2 Key performance indicators

At present there is a lack of a concept which can be used to evaluate designs and compare design data with efficiency for installed systems measured temporarily in field. Traditionally COP has been used but it has a weakness as a comparator as it is strongly dependent on operating conditions and a COP-value from a short temporarily measurement can only provide limited information about the system efficiency over time.

2.1 COP

The Coefficient of Performance, COP, is used to a large extent when the performance of heat pumps and air-conditioning equipment are evaluated today. It is given by the ratio of useful energy output for heating, P_h or cooling, P_c over energy input, $P_{e,m}$ for the process.

The definition of COP for heating *(h)* and cooling *(c)* respectively:

$$COP_h = \frac{P_h}{P_{e,m}} \qquad \textbf{Eq. 1}$$

$$COP_c = \frac{P_c}{P_{e,m}} \qquad \textbf{Eq. 2}$$

COP is useful for cost calculations, when comparing heat pump and air conditioning systems with other solutions. However COP does not tell you how well the system works compare to what is theoretically possible under present conditions or compared to other systems operating under similar conditions. Therefore it is not intuitively understood from the COP value how far from optimal operation the machine is working under certain conditions.

Furthermore, COP does not take the entire system or ambient conditions into account. A COP from measurement compared to the design COP does not relate to what is in fact possible under present conditions in the system where the equipment is installed.

The theoretically maximum reachable efficiency of the heat pump is that of a reversible process, the performance of this process is often referred to as the Carnot COP. In a reversible process all work added to the process contributes to the temperature difference between the cold and the hot side in the cycle. This is shown in Figure 2 where the reference temperature for the cold side is T_2 and for the hot side T_1. The calculation of COP Carnot *(C)*, for heating *(h)* and cooling *(c)*, can therefore be simplified to:

$$COP_{h,C} = \frac{P_{h,C}}{P_{e,m,C}} = \frac{T_1}{T_1 - T_2} \qquad \textbf{Eq. 3}$$

$$COP_{c,C} = \frac{P_{c,C}}{P_{e,m,C}} = \frac{T_2}{T_1 - T_2} \qquad \textbf{Eq. 4}$$

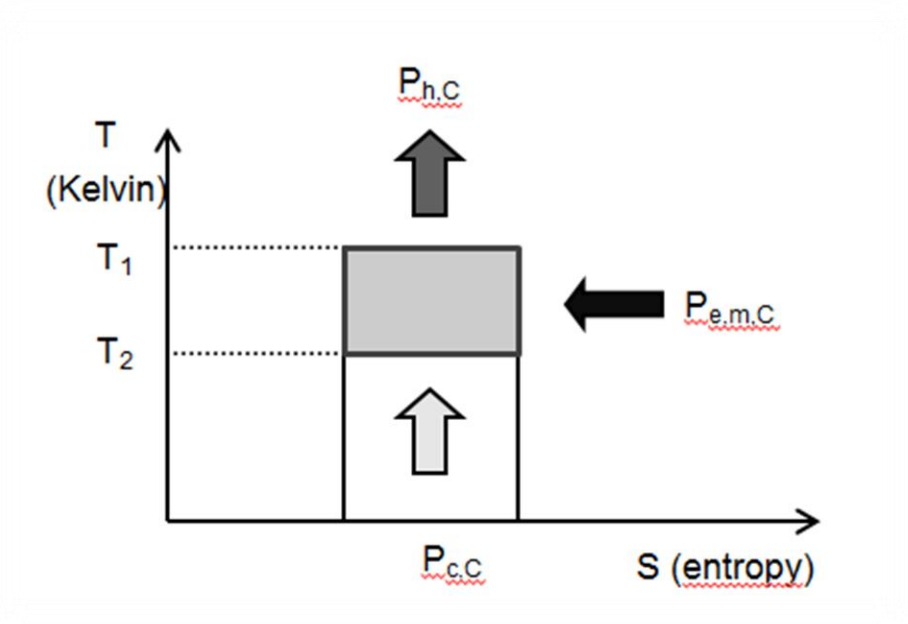

Figure 2 The Carnot process showing T_1, T_2, $P_{c,C}$, $P_{h,C}$ and $P_{e,m,C}$.

2.1.1 SPF

Seasonal Performance Factor (SPF) is a measure of a heat pumping machine's performance over one year or over the heating or cooling season, i.e. taking into account the COP at different operating conditions that occur during the year. The number is used for comparison of different heat pumps.

Within the EU project SEPEMO-Build (2009-2012) [8], system boundaries for evaluation of heat pump system performance in field were established. The seasonal performance factor, SPF was considered for four system boundaries shown in Figure 3. The SPF_H1 concerns the heat pump unit itself, and SPF_H2 concerns the system of source equipment and the heat pump unit. The SPF_H3 concerns also supplementary heat and SPF_H4 includes equipment on the heat sink side.

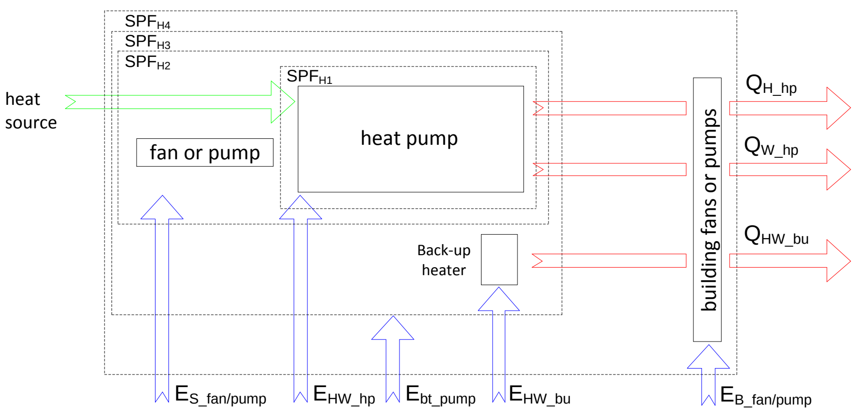

Figure 3 System boundaries defined in the SEPEMO-Build project.

2.2 SEI

The system efficiency index is the ratio of the COP-value and the COP of an ideal, reversible Carnot process, (COP_C). The SEI methodology has been independently proposed by the British Institute of Refrigeration (IOR) and the German Engineering Federation VDMA to provide a measure of energy efficiency for refrigeration systems. The objective for VDMA was for SEI values to be used by designers and purchasers of systems to compare efficiency of different system design approaches at the desired operating conditions for the plant (VDMA Specification No. 24247, Part 2 [5]). It could also be used to compare theoretical design efficiency with the practical efficiency of installed systems.

The Institute of Refrigeration, IOR, developed the concept of SEI to obtain a measure of the efficiency of refrigeration system. The concept was developed with the purpose of improving the efficiency of primary energy use and through this reduce carbon emissions associated with the use of refrigeration system. The project was carried out with support of the Carbon Trust. (Ref Guy 2007)

The SEI presented in this work is a further development of the methodology. The objectives of the present work with SEI are development of robust boundary definitions and to generate benchmark SEI values for systems. The value can also be used to compare the theoretical efficiency in design to the actual efficiency of the installed system.

The SEI value can also be reached as a product of sub-efficiencies for a system (Römer, 2011, [3]). The sub-efficiencies are efficiency measures of different parts of the process which allows identification of optimization potential in different parts of the process. In this work the sub-efficiencies are also further developed, this is described in 3.2.3.1.

SEI is created by defining the ideal COP of a 100 % efficient refrigeration process between the desired temperature levels and comparing the actual COP with this value. The ideal or Carnot COP provides the ultimate reference, consistent with the laws of thermodynamics, for a process of transferring heat energy to a higher temperature level. The design or measured COP is then divided by the ideal COP and this ratio results in an efficiency that changes much less than COP with changes in temperatures and flow rates.

3 System Efficiency Index (SEI) – definitions and system boundaries

The SEI is a measure of efficiency. The advantage of SEI compared to other KPI's such as COP is that it can provide information about a system's efficiency based only on a short instantaneous measurement in field, and correlating it to the theoretical potential performance under the actual operating conditions.

3.1 Basic definition of SEI

SEI is defined as the ratio between the COP-value calculated from measurements for a process and the theoretical maximum COP, the Carnot COP (COP_C) for the same process. The COP_C is calculated from temperatures that is defined as reference temperatures for the process. The definition of SEI is related to a cooling process or a heat pump process with different formulas.

The system efficiency index for systems supplying heat can be calculated from:

$$SEI_{h,i} = \frac{COP_{h,i}}{\dfrac{T_{ref,h,i}}{T_{ref,h,i} - T_{ref,c,i}}} \quad \textbf{Eq. 5}$$

For systems supplying cooling the following formula is used:

$$SEI_{c,i} = \frac{COP_{c,i}}{\dfrac{T_{ref,c,i}}{T_{ref,h,i} - T_{ref,c,i}}} \quad \textbf{Eq. 6}$$

Reference temperatures in Kelvin are used for the calculation. Indices *h* and *c* refer to heating or hot side and cooling or cold side respectively. Index *i* refers to system boundary and the reference temperatures that has been defined for that particular boundary.

Hot side refers to the side where heat is rejected from the unit and cold side refers to the side where heat is absorbed by the unit.

The SEI can be calculated for both heating and cooling no matter which way the unit is used since they will evaluate different parts of the system. The two SEI's can provide information for a system working with heating and cooling simultaneously.

3.2 Categorization, system boundaries and reference temperatures

There are three things that have to be considered for the SEI to make values comparable:

1. Categorization
2. System boundaries
3. Reference temperatures

3.2.1 Categorization

The categorization describes the system type for which SEI is applied. The categorization is important to be able to compare different plants with each other by using the same system boundaries.
There are two main categories, heating and cooling processes, where heat pumps are present in the first and air-conditioning in the latter.

The system design being indirect or direct has an impact on the categorization since the heat exchange is made with different medias. In a direct system the heat exchange is made directly with the media to be heated or cooled and here this is always considered to be air. Water or other medias can be considered in other cases than space heating or cooling such as hot water heating or industrial processes. There are systems with a mixture of direct and indirect circuits for condenser and evaporator. In the Figure 4 an overview of possible systems is given. The kind of heat source and heat sink; i.e. air or liquid, has to be described in order to compare values for different systems. The air can for example be outdoor air or exhaust air.

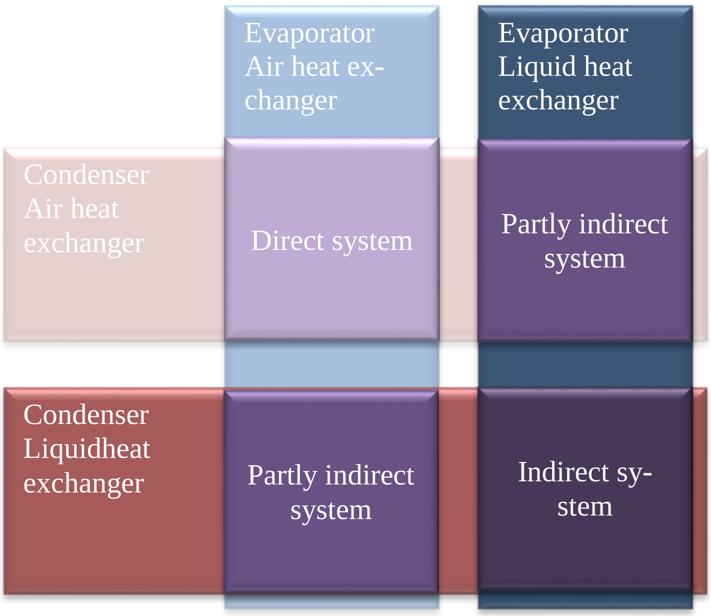

Figure 4 An overview of possible heat exchanger fluids and heat pump systems

In this report, the focus and evaluation of SEI is on indirect systems without oil-cooling and sub-coolers and does not include systems with several evaporators with different heat transfer medias etc. When there is sub coolers, oil-cooling etc. in the system they have to be taken in to account in the SEI calculation.

3.2.2 System boundaries

The SEI has been divided into four main system boundaries according to how much of the system that is included in the measurement. The system boundaries presented here are based on the work in SEPEMO-Build project (Zlottl, Nordman, 2012 [8]), with some changes. In Figure 5 and Figure 6 the system boundaries for SEI 1 to 4 for an indirect and a direct system are illustrated.

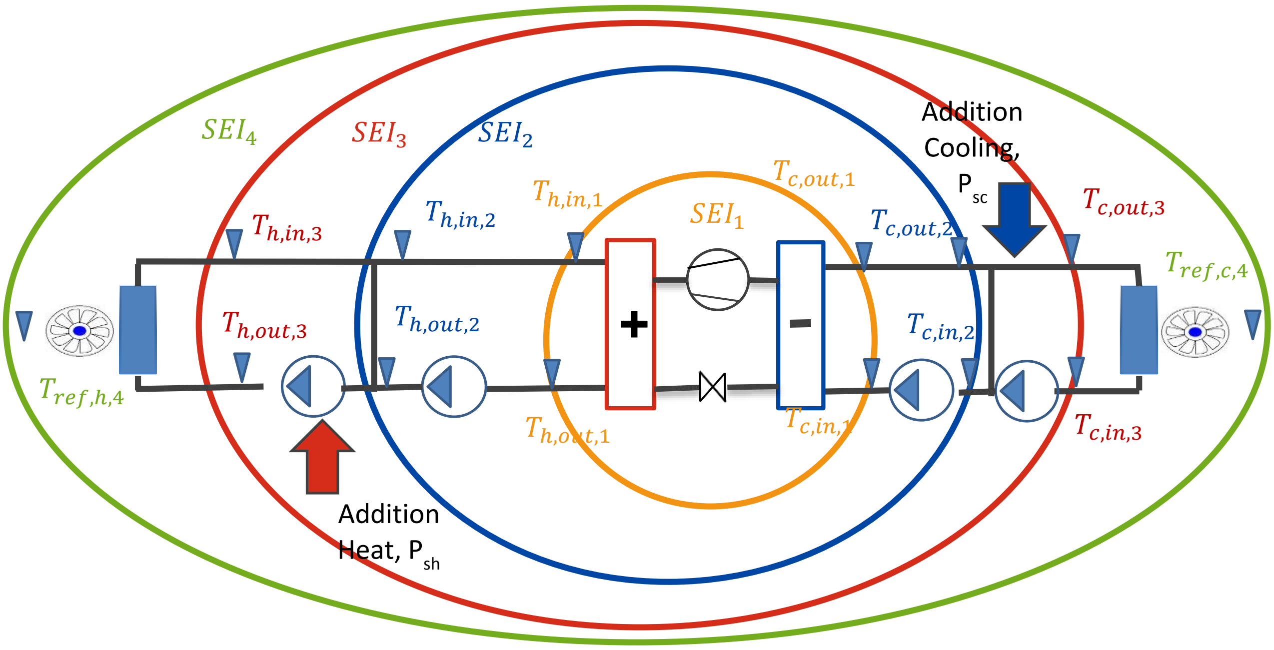

Figure 5 Indirect system with system boundaries and temperatures

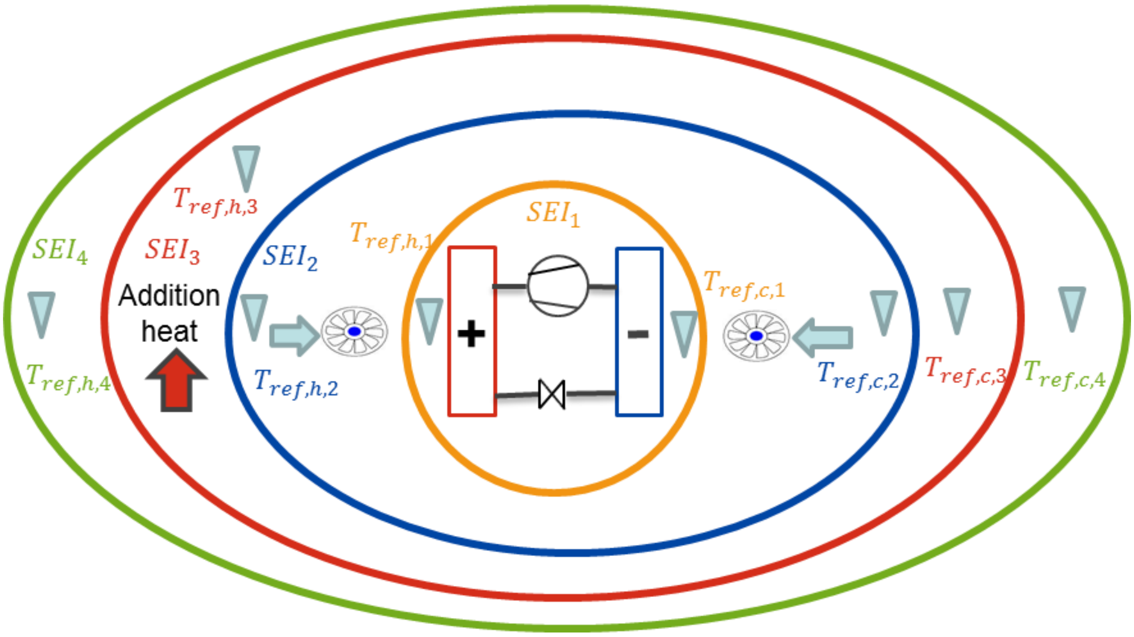

Figure 6 Direct system with system boundaries and reference temperatures

The choice of the system boundary influences the SEI value due to which auxiliary drives included and which reference temperatures used. This will reflect the impact of the different devices on the performance of the system. Therefore the system boundary used for calculation of SEI should be given. For heat pumps and air conditioning systems four system boundaries have been defined. The SEI_1 is the SEI closest to the refrigeration circuit and includes only the refrigerant process and the load from the compressor. The widest SEI, SEI_4 refers to system boundaries including all equipment from the heat sink to the heat delivery.

To calculate the Carnot COP, the reference temperatures have to be defined and measured. They should represent the cold and the warm level of the process according to the system boundaries that the evaluation relates to. This means that the reference temperature will be different depending on the system boundary, the process (heating or cooling) and the system (direct or in-direct).

The SEI_1 is the closest SEI and includes only the refrigerant process and the load from the compressor. When the system boundary is extended the reference temperatures have to be moved, see Figure 5 and Figure 6 where temperatures are indicated for the indirect and direct systems.

The reference temperature in indirect system, see Figure 5, is defined as the mean value of the in-coming and outgoing temperature of heat transfer media. This is applied both for the hot and the cold side, for example the reference temperature on the hot side for SEI_2 used is:

$$T_{ref,h,2} = (T_{h,in,2} + T_{h,out,2})/2 \qquad \textbf{Eq. 7}$$

In direct systems, see Figure 6, the reference temperature of the heat transfer media for heating refer to incoming temperature to the evaporator and leaving temperature from the condenser. For an air to air heat pump this means that the reference temperatures will be the out-door temperature on the cold side and the indoor temperature on the warm side.

The reference temperatures of the heat transfer media for cooling is the other way around; incoming temperature to condenser and out-going temperature from evaporator. For an air-conditioning unit this means the indoor temperature as reference temperature on the warm side the out-door temperature as reference temperature on the cold side. For duct mounted evaporators or condensers, the reference temperature can be measured in the duct.

Practical circumstances might complicate measurement of reference temperatures at the desired position, such as the $T_{c,in,1}$ for an air-conditioner with system boundary for SEI_1. The desired reference temperature would be measured between the fan and the evaporator. Practical measurement however is possible before the fan, meaning that the same reference temperature will be used for SEI_1 and for SEI_2.

Measurements available in the project has included the system boundaries one and two. Measurements with these boundaries provides information about the efficiency of the heat pump or air conditioning unit. Therefore, the focus in this report is evaluations of SEI_1 and partly SEI_2. The reflection on SEI_3 and SEI_4 are theoretical. For large systems the measurement and evaluation of SEI_3 and SEI_4 tend to be complex and hard to put into practical use. They still can be useful for small systems and air to air units.

3.2.3 SEI₁

An evaluation of SEI made with system boundary 1 will provide information about the efficiency of the refrigeration process. This is the closest system boundary and includes only the refrigeration circuit and the load from the compressor as shown for an indirect system in Figure 7. SEI_1 from measurement can be compared to design data or other units to investigate if the unit is performing as expected.

The evaluation of the refrigeration process is the most important use of SEI as a key-performance indicator. The evaluation is useful for the process according to delivered temperatures. To evaluate if the delivered temperatures is the right ones for the demand side of the system, another method should be used.

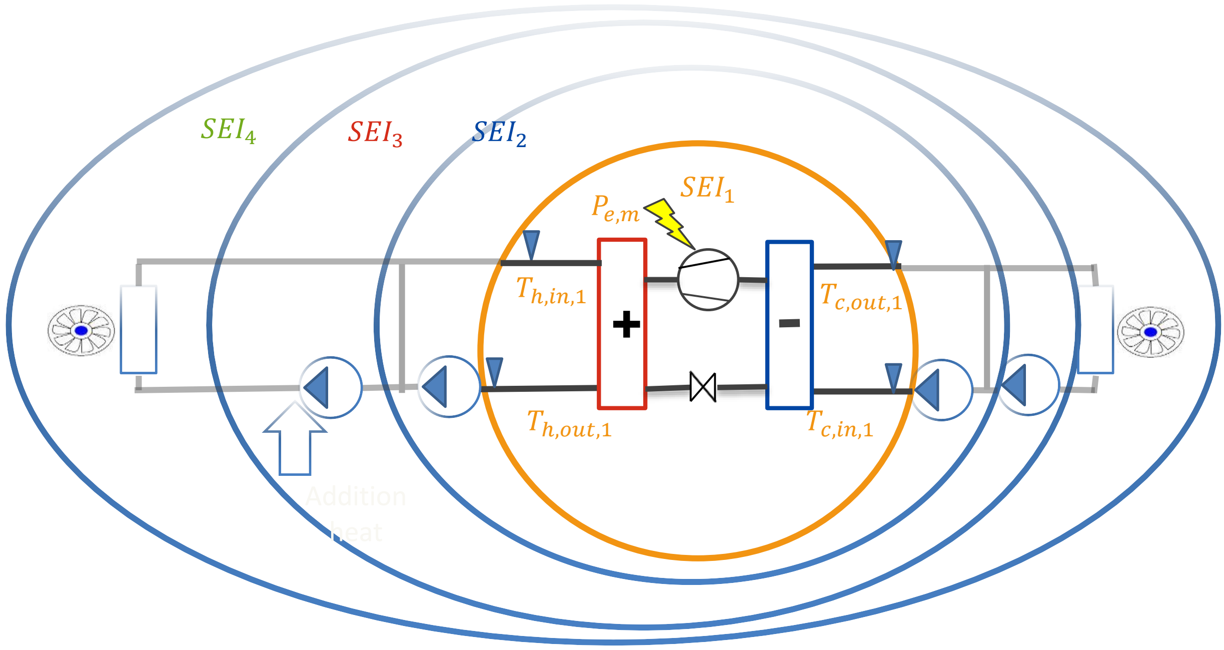

Figure 7 The system boundary of SEI1

Here follows an example structure for calculations and monitoring parameters for evaluation of SEI_1 in an indirect system.

The compressor power, ($P_{e,m}$) has to be monitored as well as the heating or cooling capacity (P_h or P_c) along with the temperatures shown in Figure 7 for calculation of reference temperatures.
The reference temperatures for indirect (liquid/liquid) systems are average temperatures of incoming and leaving heat transfer media according to:

$$T_{ref,c,1} = \frac{(T_{c,in,1} + T_{c,out,1})}{2} \qquad \textbf{Eq. 8}$$

$$T_{ref,h,1} = \frac{(T_{h,in,1} + T_{h,out,1})}{2} \qquad \textbf{Eq. 9}$$

The COP will be calculated according to:

$$COP_{h,1} = \frac{P_h}{P_{e,m}} \qquad \textbf{Eq. 10}$$

$$COP_{c,1} = \frac{P_c}{P_{em}} \qquad \textbf{Eq. 11}$$

The SEI_1 is then calculated with the COP and reference temperatures according to:

$$SEI_{h,1} = \frac{COP_{h,1}}{\frac{T_{ref,h,1}}{T_{ref,h,1} - T_{ref,c,1}}} \qquad \textbf{Eq. 12}$$

$$SEI_{c,1} = \frac{COP_{c,1}}{\frac{T_{ref,c,1}}{T_{ref,h,1} - T_{ref,c,1}}} \qquad \textbf{Eq. 13}$$

3.2.3.1 Sub – efficiencies of SEI

In order to analyse the performance of a system further and to find reasons for poor performance, the SEI sub-efficiencies can be used. Here, the six sub-efficiencies shown for $SEI_{c,1}$ are showing efficiency of components or parts of the refrigeration process. The $SEI_{c,1}$ can be reached as a product of the sub-efficiencies. By analysing the sub-efficiencies it is possible localize where in the process the performance problem exits. Some of the sub-efficiencies are used in the analysis in chapter 6. How to determine sub-efficiencies are described further in Appendix.

$SEI_{c,1}$ for cooling is built up of the following sub-efficiencies:

η_1: Refrigeration cycle efficiency. This sub-efficiency takes into account losses inherent in the refrigeration cycle itself, and can for example be used in the design process to compare effects of different refrigerants, or the effects of the use of an economiser.

η_2: Compressor efficiency. This sub-efficiency includes the effect of compressor efficiency. This is different for heating and cooling operation since the compressor losses (heat into refrigerant) can be part of the useful heat output during heating.

η_3: Pressure drop in refrigerant lines. This effect is probably most significant for large refrigeration systems typically with several long pipes.

η_4: Heat Exchanger efficiency. This effect includes the heat exchangers effectiveness and is extremely important in evaluating or comparing condenser and evaporator temperature differences.

η_5: Fluid transfer efficiency.

η_6: Non useful heat loss/gain. This effect includes ‘non useful’ heat pick up, for example heat pick up by cold suction lines in cooling applications.

3.2.4 SEI₂

An evaluation of SEI made with system boundary 2 will provide information about the efficiency of the refrigeration process and taking the efficiency of circulation pumps into account, shown for an indirect system in Figure 8.
This system boundary includes the refrigeration circuit and the equipment (fans and/or pumps) needed to make source and sink energy available for it. SEI_2 evaluates the performance of the refrigeration process with distribution pumps and/or fans for the condenser and evaporator circuits as illustrated in Figure 8. If there is only one pump for the whole distribution in the heat transfer media, SEI_3 should be used.

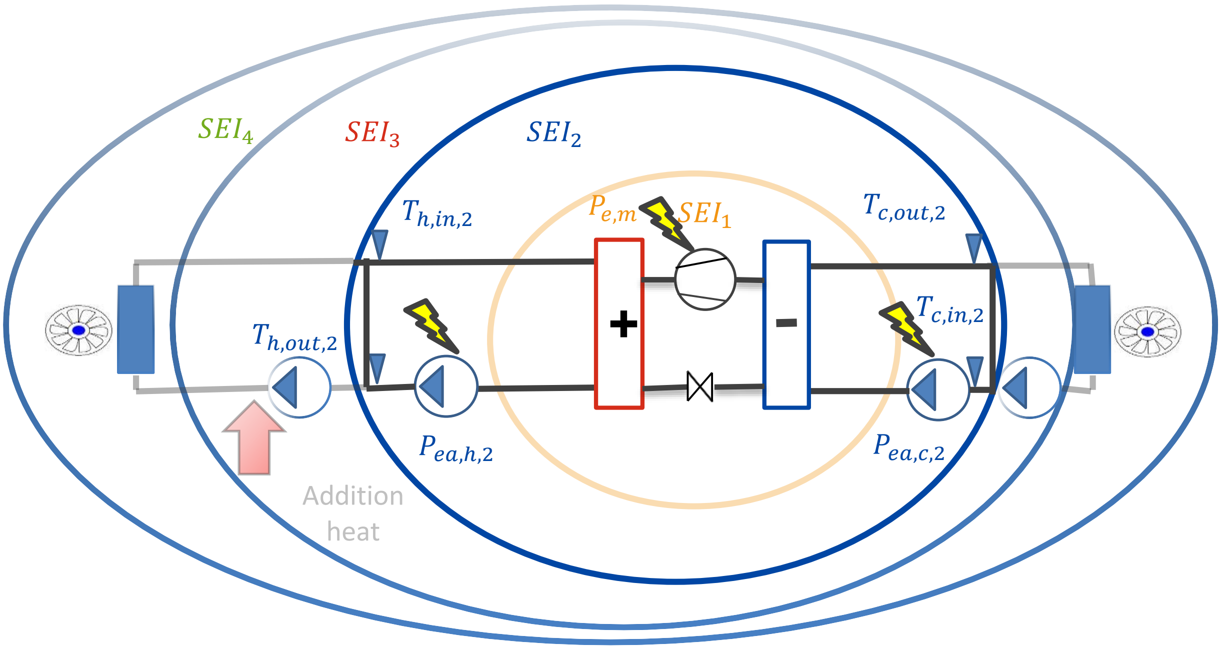

Figure 8 The system boundary of SEI_2

Here follows an example structure for calculations and monitoring parameters for evaluation of SEI_2 in an indirect system.

The compressor power, ($P_{e,m}$) has to be monitored as well as the heating or cooling capacity (P_h or P_c) along with the temperatures shown in Figure 8 or calculation of reference temperatures. The reference temperatures for indirect (liquid/liquid) systems are average temperatures of incoming and leaving heat transfer media as described for SEI_1.

Measurements of pump power $P_{ea,h,2}$ and $P_{ea,c,2}$ is also needed.

The COP will be calculated according to:

$$COP_{h,2} = \frac{P_h}{P_{e,2}} = \frac{P_h}{P_{e,m} + P_{ea,h,2} + P_{ea,c,2}} \quad \textbf{Eq. 14}$$

$$COP_{c,2} = \frac{P_c}{P_{e,2}} = \frac{P_c}{P_{em} + P_{a,h,2} + P_{ea,c,2}} \quad \textbf{Eq. 15}$$

The SEI$_2$ is then calculated with the COP and reference temperatures according to:

$$SEI_{h,2} = \frac{COP_{h,2}}{\frac{T_{ref,h,2}}{T_{ref,h,2} - T_{ref,c,2}}} \qquad \textbf{Eq. 16}$$

$$SEI_{c,2} = \frac{COP_{c,2}}{\frac{T_{ref,c,2}}{T_{ref,h,2} - T_{ref,c,2}}} \qquad \textbf{Eq. 17}$$

3.2.5 SEI$_3$

An evaluation of SEI made with system boundary 3 will provide information about the efficiency of the refrigeration process and distribution system as well as additional heating or cooling. This is shown for an indirect system in Figure 9.
This system boundary includes the refrigeration circuit, the equipment to make the source and sink energy available and also any power used for supplementary heating or cooling. This means that all power for distribution of heat transfer media is included in the SEI$_3$.

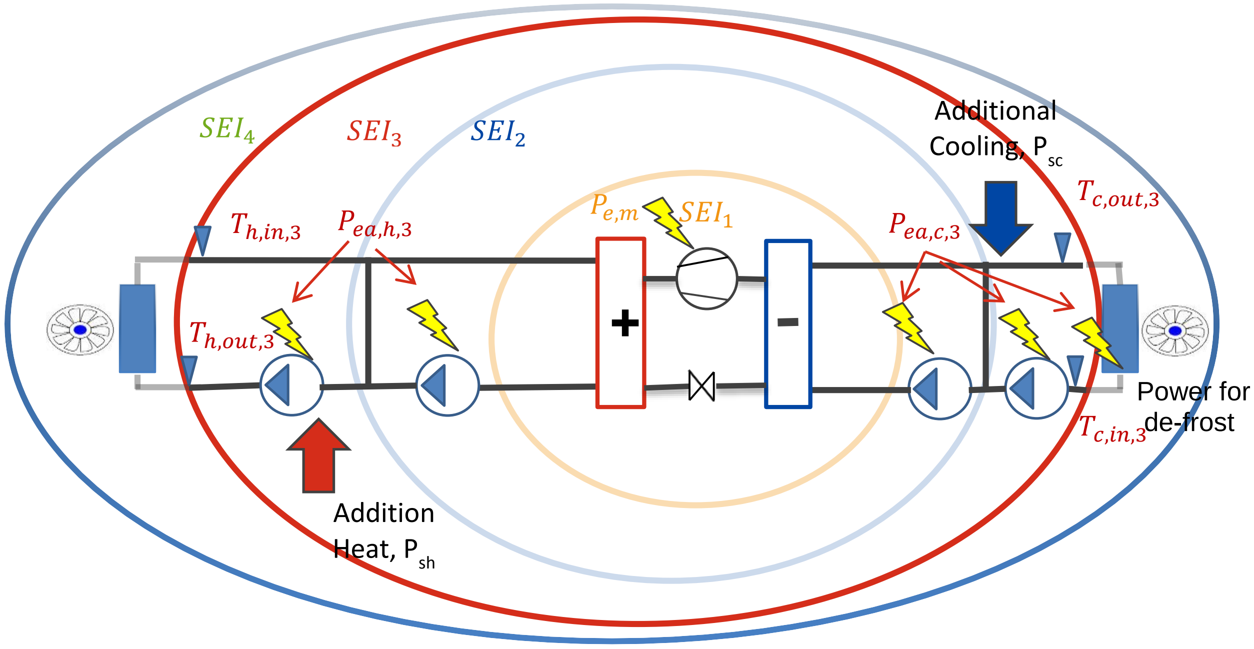

Figure 9 The system boundary of SEI$_3$

Here follows an example structure for calculations, including monitoring parameters for evaluation of SEI$_3$ in an indirect system.

The compressor power, ($P_{e,m}$) has to be monitored as well as the heating or cooling capacity (P_h or P_c) along with the temperatures shown in Figure 9 for calculation of reference temperatures. The reference temperatures for indirect (liquid/liquid) systems are average temperatures of incoming and leaving heat transfer media as described for SEI$_1$.

Measurements of all pump power $P_{ea,h,3}$ and $P_{ea,c,3}$ is also needed. Moreover, the additional heating P_{sh}or cooling P_{sc} delivered to the system has to be monitored. Also, the additional power required to produce and deliver the heating or cooling to the system.

For example in case of oil burners, the supplementary heating P_{sh} will be the heat capacity delivered to the system from the oil burner, whereas the power input $P_{e,sh}$ to the oil burner includes both oil and additional pumps.

For cooling, the P_{sc} is the cooling capacity delivered to the system, whereas the power input $P_{e,sc}$ is the power for pumps and fans used to deliver the cooling capacity.

The COP will be calculated according to:

$$COP_{h,3} = \frac{P_{h,3}}{P_{e,3}} = \frac{P_h + P_{sh}}{P_{e,m} + P_{ea,h,3} + P_{ea,c,3} + P_{e,sh}} \quad \textbf{Eq. 18}$$

$$COP_{c,3} = \frac{P_{c,3}}{P_{e,3}} = \frac{P_c + P_{sc}}{P_{e,m} + P_{ea,h,3} + P_{ea,c,3} + P_{e,sc}} \quad \textbf{Eq. 19}$$

The SEI_3 is then calculated with the COP and reference temperatures according to:

$$SEI_{h,3} = \frac{COP_{h,3}}{\dfrac{T_{ref,h,3}}{T_{ref,h,3} - T_{ref,c,3}}} \quad \textbf{Eq. 20}$$

$$SEI_{c,3} = \frac{COP_{c,3}}{\dfrac{T_{ref,c,3}}{T_{ref,h,3} - T_{ref,c,3}}} \quad \textbf{Eq. 21}$$

3.2.6 SEI_4

The total system is evaluated with the system boundary 4 and the SEI_4 will provide information about the efficiency of the whole process required to deliver heating or cooling to the end user. This boundary is shown for an indirect system in Figure 10.

The system boundary includes the refrigeration circuit, the equipment to make the source and sink energy available, the equipment for supplementary heating/cooling and all auxiliary drives including the auxiliary of the heat sink and heat source system. This system boundary is not practically feasible for many large systems.

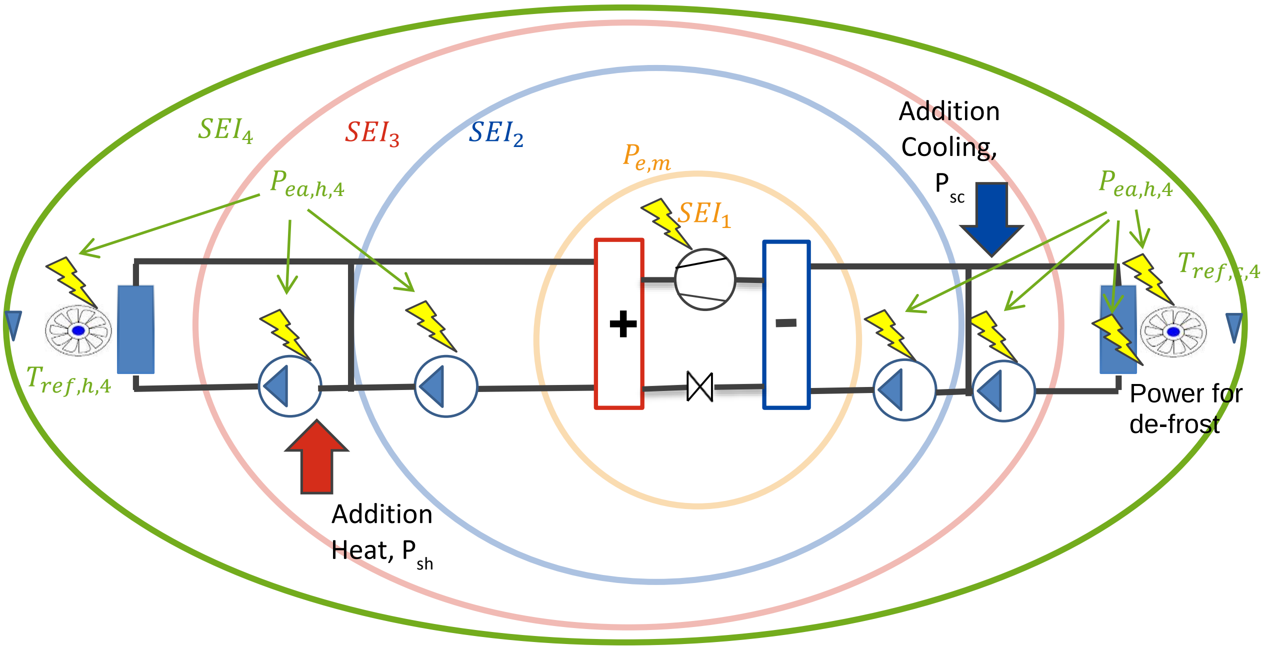

Figure 10 The system boundary of SEI₄

Here follows an example structure for calculations, including monitoring parameters for evaluation of SEI_4 in an indirect system.

The compressor power, ($P_{e,m}$) has to be monitored as well as the heating or cooling capacity (P_h or P_c) along with the temperatures shown in Figure 9 are reference temperatures. The reference temperature in this system boundary are the undisturbed air temperatures on the cold and warm side. For example the outdoor and indoor air temperatures.

Measurements of all power to auxiliary drives $P_{ea,h,4}$ and $P_{ea,c,4}$ is also needed. Moreover, the additional heating P_{sh} or cooling P_{sc} delivered to the system has to be monitored. Also, the additional power $P_{e,sh}$ or $P_{e,sc}$ required to produce and deliver the supplementary heating or cooling to the system. See SEI_3 for examples.

The COP will be calculated with the compressor power, pump power for distribution system and fan power for heat exchange to air according to, additional heating/cooling according to:

$$COP_{h,4} = \frac{P_{h,4}}{P_{e,4}} = \frac{P_h + P_{sh}}{P_{e,m} + P_{ea,h,4} + P_{ea,c,4} + P_{e,sh}} \quad \textbf{Eq. 22}$$

$$COP_{c,4} = \frac{P_{c,4}}{P_{e,4}} = \frac{P_c + P_{sc}}{P_{e,m} + P_{ea,h,4} + P_{ea,c,4} + P_{e,cs}} \quad \textbf{Eq. 23}$$

The SEI_4 is then calculated with the COP and reference temperatures according to:

$$SEI_{h,i} = \frac{COP_{h,1}}{\frac{T_{ref,h,1}}{T_{ref,h,1} - T_{ref,c,1}}} \qquad \textbf{Eq. 24}$$

$$SEI_{c,i} = \frac{COP_{c,1}}{\frac{T_{ref,c,1}}{T_{ref,h,1} - T_{ref,c,1}}} \qquad \textbf{Eq. 25}$$

3.2.7 Discussion of system boundaries

The closest system boundary, SEI_1, is useful for analysis of the refrigerant circuit. It includes power used by the compressor and related to delivered cooling or heating power from the unit. The reference temperatures are for liquid system mean values for the heat transfer media. This system boundary is in practice the easiest one to use for liquid systems. With the internal measurement method, described in chapter 4, there will be no need for extra meters for electrical power, which makes the measurement more easy.

The next system boundary, SEI_2, includes power for pumps that distribute the heat transfer media on the cold and hot side. This boundary fits when the practical solution for the system is divided in two circuits on the secondary side, so there is one pump for distribution to the system and another for the circulation through the evaporator or condenser. In practice there can be a mixture of these boundary which make measurements harder to compare with other systems.

When the distribution to the whole system circuit is included in the power, system boundary SEI_3 is used. This boundary also includes power for supplementary heating and other auxiliary's that can be included in the liquid system. In practice it can be complicated to make a strict SEI_3 measurement according to auxiliaries that can be placed far out in the system, even if distribution pumps easy can be measured in the machine room. The boundaries can also be mixed. For example, in an air conditioning system it can be feasible to measure all auxiliaries on the hot side for heat rejection, but on the cold side there may be many auxiliaries located far away from the unit making measurement more complicated.

When a measurement is possible for all parts of a system, SEI_4 can be defined. In practice this probably only will be possible for small systems and air to air units. There can be many auxiliary's to measure, but there is also a problem with the reference temperature if the system deliver to many different rooms in a building. It's then hard to find a reference temperature that is useful. Also for SEI_4 the measurement can be feasible for one side in the system, but not for the other one.

To continue with the example of the air conditioning unit, it can be easy to measure outdoor temperature as a reference temperature on the warm side and even electrical power for pumps and fans for the heat rejection from the condenser, but on the cold side the system can be large and deliver cooling to many rooms with different temperature and auxiliaries, as fans can be placed in many rooms and make the measurement complicated. A mix of system boundaries can be used in this case, but the recommendation is not to use measurements for mixed boundaries in other cases than for analyse of a specific system. In cases that the system boundary is different on the cold and warm side, the SEI is indexed with two number, where the first number is the system boundary on the cold side. When a SEI-measurement is presented, the system boundaries must be clearly defined.

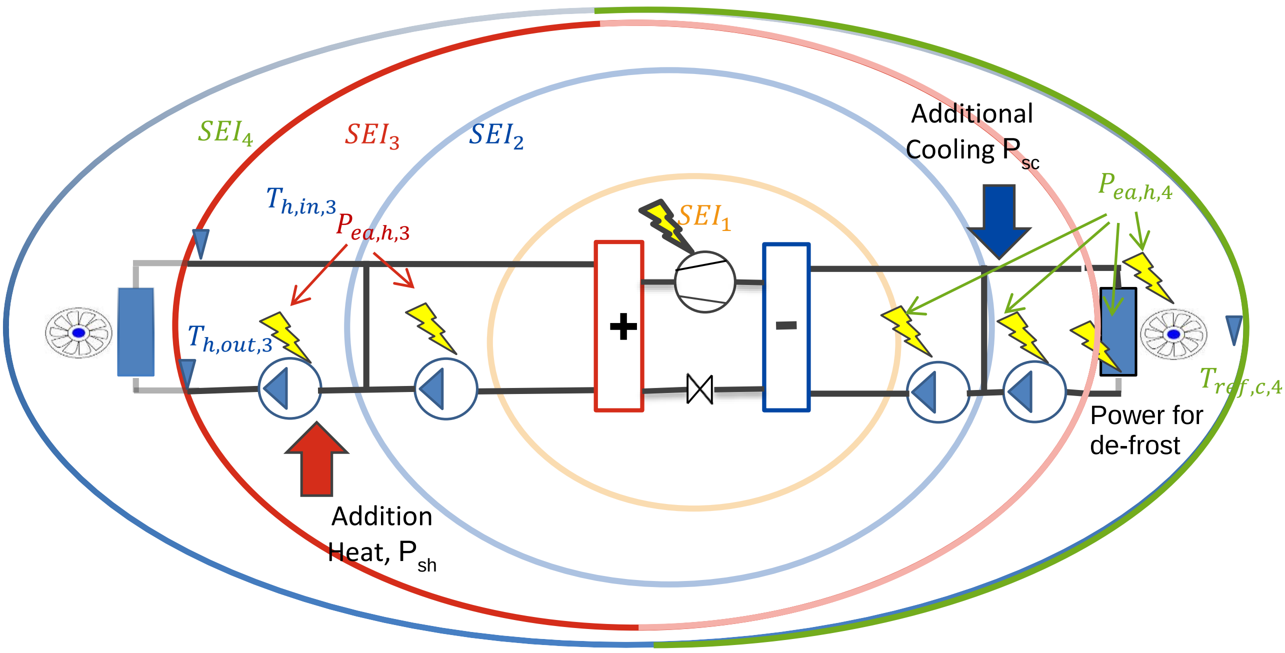

Figure 11 Example of mixture system boundaries. Here $SEI_{4,3}$ can be measured. System boundary 4 on the cold side and 3 on the hot side.

4 Measuring methods to determine SEI

The key performance indicator, SEI, is based on instantaneous or continuous field measurements and the results can be used for energy optimization. In previous works by IOR and VDMA [4-6] focus for SEI was on the construction process where SEI was used as indicator for choosing the best components in the system. If manufacturing data is available, the connection between construction and field measurement gives more knowledge about the system performance.
According to the theory of chapter 2 and 3, SEI provides a weighted measure of the coefficient of performance of a specific system in relation to its operating conditions. Hence, to determine SEI we need the COP and a reference thermal operating condition at a specified system boundary. To determine COP in practical installations there are two basic types of standardized measuring methods, the internal and external method. These methods will be further described in this chapter.

4.1 Conditions for measurements

The SEI not being dependent on a specific operating condition is an advantage in field measurements. However to determine SEI from a measurement, the operation range for the system must be known and the operating point should both be within this range and according to the common operation of the unit. The operating condition should also be stable. The operation should be according to the demand in the system to get stable conditions. Defrost operation should be avoided. For measurements of SEI with wider system boundaries, the operation and function of the system should be known, as for example operation of supplementary heating or free cooling.

4.2 Methods for measurements

The Carnot COP is calculated from measurements of the reference temperatures on the cold and hot side according to the system boundaries chosen for the measurement. To calculate the COP, delivered cooling or heating capacity has to be measured and the power used to produce the cooling and heating capacity. Used power is measured by an electric meter to the compressor and to other auxiliaries included in the system boundary.

Heating and cooling capacity can be measured with two methods: the external and internal method. In the external method the cooling or heating capacity is measured in the secondary liquid system with a flow meter and temperature sensors. It can also be done for air to air systems with temperature sensors and air flow meter, but it is rather complicated. The internal method is based on temperature and pressure measurements in the refrigerant circuit, which together with an estimation of the heat loss from the compressor is used for calculation of the heating and cooling power. For analyse of the total SEI both the external and the internal method is useful. For a more detailed analysis with sub-efficiencies described in 3.2.3.1 the internal method has to be used.

The external and internal methods have also been known as direct and indirect methods. Fahlén introduced the classification external and internal as measurements are either conducted outside the refrigeration circuit of the unit (external) or inside the refrigeration circuit of the unit (internal[21]). Irrespective of the chosen method, however, stable operation under a sufficiently long period and suitable and accessible measuring positions are essential requirements in order to generate relevant data

4.2.1 External method

4.2.1.1 Liquid systems

The traditional way of determining heating or cooling capacity [18] is by measurement of flow rate and temperature difference. Quantities that may need to be measured are:

- Temperatures of warm and cold heat transfer media (water or brine).
- Flow rate for warm and cold heat transfer media (water or brine)
- Electric powers (compressors, pumps, fans, supplementary heaters) according to system boundary for calculation of COP.

Thermodynamic properties for the cold and heat transfer media, as density and specific heat capacity, must be known or measured for the measured temperatures. Thermal capacity for heating or cooling (P) is derived from measured values of volume flow rate (q_v), density (ρ), specific heat capacity (c_p) and the temperature difference between outlet (t_{out}) and inlet (t_{in}) according to:

$$P = q_v \cdot \rho \cdot c_p \cdot (t_{out} - t_{in}) \qquad \textbf{Eq. 26}$$

Electric power ($P_{e,m}$) is directly measured by an electric power meter or indirectly by means of an energy meter and time, voltage-current-power factor etc. The coefficient of performance may then be calculated via the well-known relation, here shown for cooling:

$$COP_c = \frac{P_c}{P_{e,m}} \qquad \textbf{Eq. 27}$$

For liquid systems there are methodology well described in the Swedish standard SS2620[29] and the Nordtest standard NTVVS 076 [16,25]There are many handbooks[18] that describe different methods of measuring flow rate and temperature. Usually measurements are invasive (i.e. sensors inserted in the fluid system) but there are also non-invasive measurement, e.g. by studying the propagation of heat pulses, by use of tracers [24,26] or by means of external ultra-sound meters [14].

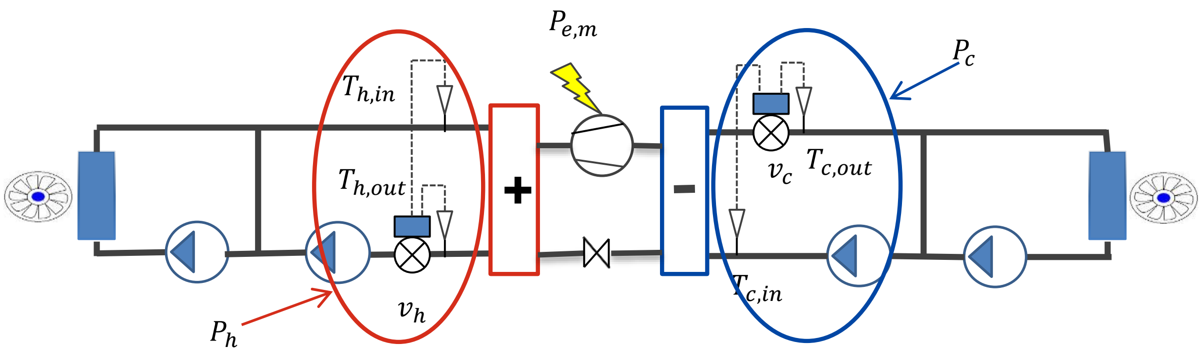

Figure 12 Measurement points needed for the external method to calculate COP_c and COP_h

Invasive meters have to be mounted in the tubes, which makes the measurement complicated for system that is not prepared with equipment at installation.

4.2.1.2 Air systems

There are methods to measure heat and cooling capacity for air heat exchangers, as for example SP method no. 1721[35] which is a method for field testing of electrically driven air to air heat pumps in heating or cooling mode, but the measurements are complicated. The method haven't been used in this project. The principle and formulas is the same as for liquid systems, but the variation in temperature and flow over the surface is bigger for air heat-changer, which makes the measurement more complicated in practice.

4.2.1.3 Prerequisites

The external method presumes that thermo-physical property data (ρ and c_p) for heat transfer medias for heating and cooling are sufficiently well known. In the case of air and water this is no problem while cooling agent mixtures/solutions often lack validated data (Fahlén [19] has shown that this in many cases may be a larger source of uncertainty than the measurements per se). Density shall be assessed from values of pressure and temperature at the position of the flow meter while the specific heat capacity is taken for the mean values of the inlet and outlet temperatures (theoretically this should be the integrated mean value but c_p varies relatively little within a reasonable temperature interval). Another important prerequisite is that it must be possible to correctly install sensors at relevant system boundaries. Please note that flow meters often require long inlet and outlet straights (longer than usually prescribed by flow meter suppliers). It is also important to choose flow meter according to the measurement area to get the right accuracy for the measurement.

4.2.1.4 Uncertainty and practical experience

Experience [19] from laboratory testing (e.g. by means of audits and Round Robin tests) show that it is possible to determine heating capacity with an uncertainty that is less than 1-2 %. In field situations it is rarely possible to achieve uncertainties lower than 5 % [21,22]. In very large systems, however, such as district heating or district cooling installations, it is possible to achieve stability and accuracy of measurement approaching laboratory levels.

4.2.2 Internal method

The method is based on the possibility to thermodynamically determine the refrigeration process by assessing the specific enthalpy changes in various parts of the refrigerant system. (Berglöf, 2004) [7].

Quantities that may need to be measured are:

- Surface temperatures in the refrigerant system
- Refrigerant high (condensing) and low (evaporating) pressure
- Electric powers (compressors, pumps, fans, supplementary heaters) according to system boundary for calculation of COP.

Knowing the temperature, pressure and type of refrigerant at a particular location it is possible to get the value of the specific enthalpy in tables or charts or to calculate the value by means of known correlation or by means of programs. In a simple, one-stage refrigeration cycle measurement of the condensing and evaporating pressures, the outlet and suction temperatures of the compressor and the sub-cooled liquid temperature after the condenser will suffice. These measurements, i.e. two pressures and three temperatures provide sufficient information in order to visualize the process by means of points 1, 2, and 7 in the diagram. Assuming isenthalpic expansion point 8 will also be known.

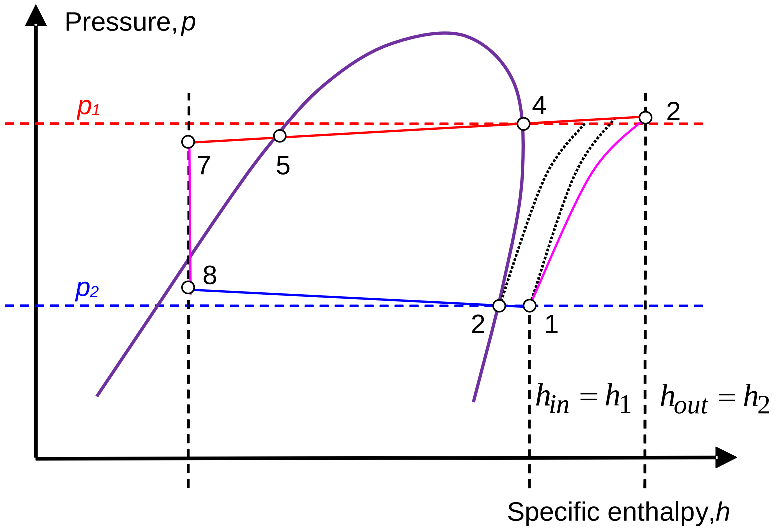

Figure 13 The refrigeration process in a diagram of specific enthalpy versus pressure.

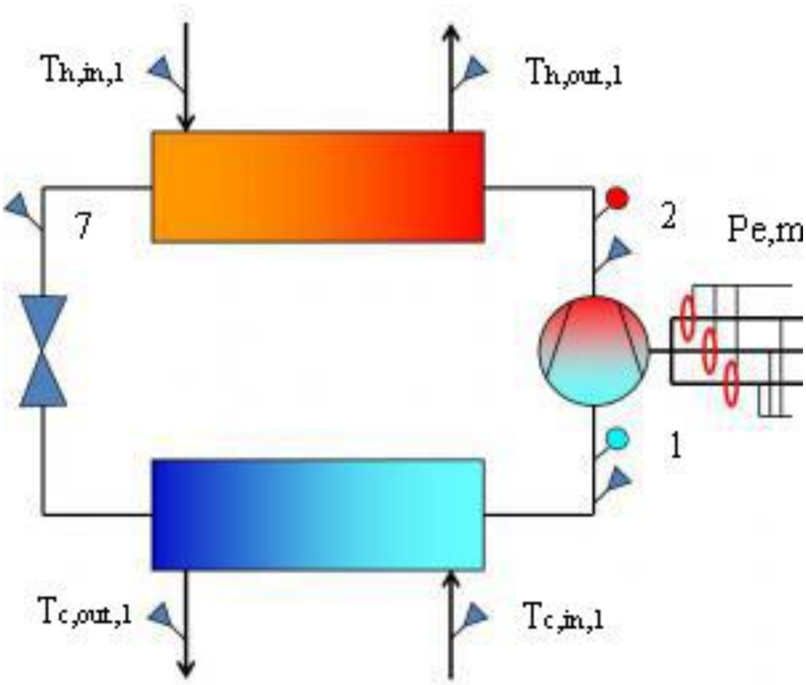

Figure 14 Measurement points, (number 1, 2 and 7 according to Figure 13) for the internal method (COP) and reference temperatures for SEI$_1$.

By means of computerized equipment it is possible to use the measured pressures and temperatures to calculate the specific enthalpies on the saturation curve as well as in the regions of superheated vapour and sub-cooled liquid. When the process has been mapped it is possible to calculated the heating or cooling coefficient of performance, e.g.

$$COP_h = \frac{h_2 - h_7}{h_2 - h_1}$$ **Eq. 28**

where $h_2 - h_7$ represents the heat delivered by the refrigerant to the condenser and $h_2 - h_1$ represents the work supplied to the refrigerant by the compressor. Unfortunately, reality is a little more complex as not all of the compressor work will result in a specific enthalpy increase from point 1 to point 2 due to thermal losses to the ambience.

In a simple, hermetic compressor, the losses P_{loss} may be expressed as a fraction f of the motor power input $P_{e,m}$. This enables a calculation of the refrigerant mass flow rate $q_{m,R}$ according to

$$q_{m,R} = \frac{P_{e,m} \cdot (1-f)}{h_{out} - h_{in}} = \frac{P_{e,m} \cdot (1-f)}{h_2 - h_1} \quad \text{[kg/s]} \qquad \text{Eq. 29}$$

Then the heat pump heating coefficient of performance COP_h is given by

$$COP_h = \frac{P_h}{P_{e,m}} = \frac{q_{m,R} \cdot (h_2 - h_7)}{P_{e,m}} = \frac{P_{e,m} \cdot (1-f) \cdot (h_2 - h_7)}{P_{e,m} \cdot (h_2 - h_5)} \quad \text{[-]} \qquad \text{Eq. 30}$$

i.e.

$$COP_h = \frac{(1-f) \cdot (h_2 - h_7)}{(h_2 - h_1)} \quad \text{[-]} \qquad \text{Eq. 31}$$

For cooling the coefficient of performance COP_c in corresponding way is given by:

$$COP_c = \frac{(1-f) \cdot (h_1 - h_7)}{(h_2 - h_1)} \quad \text{[-]} \qquad \text{Eq. 32}$$

The method can either be used to directly determine the coefficient of performance, using a known value of the loss factor, or to study performance changes after calibration by means of a parallel external measurement or simply to assess relative changes.

4.2.3 Prerequisites

The internal method presumes that it is possible to make a reasonably good power balance of the compressor. This means that the method is most viable in systems where the compressor motor is primarily cooled by means of the refrigerant flow, i.e. in systems with fully or semi hermetic compressors. It will also work reasonably well with open compressors if one knows the ambient losses (electric and mechanical transmission losses). In situations where the compressor is externally cooled, e.g. by means of air, water, oil, liquid injection etc., it is necessary to determine the cooling capacity with sufficient accuracy.

The calculation of the specific enthalpy change of the refrigerant presumes that all refrigerant becomes liquid in the condenser and gas in the evaporator. To ascertain that these prerequisites are complied with, certain minimum values of sub-cooling and superheat are required. As both of these values are measured, a check that the prerequisites are fulfilled is always available. It is also possible to obtain complementary information such as isentropic efficiency, Carnot efficiency etc. and to provide warnings when these values become unreasonable. Also, just as the case will be with any type of method, the internal method presumes that the unit can operate with stable conditions during a sufficiently long period of time.

4.2.4 Uncertainty

The total uncertainty of measured $COP_{h,}$ and COP_c is composed partly of methodical errors and partly by measuring errors. The methodical errors for the internal method comprise:

- assumptions regarding certain compressor losses
- uncertainty of tabular data or equations of state regarding thermo-physical refrigerant properties
- uncertainty regarding the refrigerant vapour quality (vapour ratio)
- uncertainty regarding the fraction of enthalpy change that is transferred to the heating agent

Measuring errors comprise the influence of measuring uncertainties for pressure and temperature on calculated values for specific enthalpy and thus the influence on COP_h and COP_c according to Eq. 31 the measuring uncertainties consist of:

- uncertainty of sensors (calibration), computational algorithms)
- uncertainty due to installation conditions (straight lengths, sensor positioning, insulation etc.).

4.3 Measurement overview

The measurements that are required to be taken on a site for each stage (SEI_1 / SEI_2) are shown in the table. The parameters needed to calculate SEI are **T_{ref1}, T_{ref2}, and COP.**

Table 1 Measurement points

Measurement points for COP, external method		Measurement points for COP, internal method	
Temperatures		*Temperatures*	
$t_{h,in,1}$		$t_{h,in,1}$	
$t_{h,out,1}$		$t_{h,out,1}$	
$t_{c,in,1}$		$t_{c,in,1}$	
$t_{c,out,1}$		$t_{c,out,1}$	
Flow		t_1	
q_{vh}		t_2	
q_{vc}		t_7	
Power		*Power*	
$P_{e,m}$		$P_{e,m}$	
		Pressure	
		p1	
		p2	

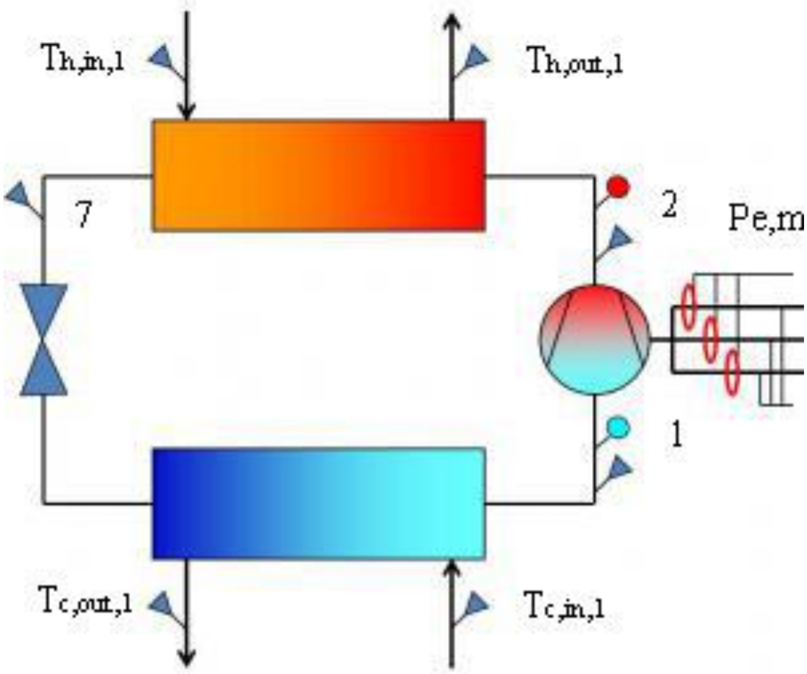

Figure 15 Measure points for the internal method

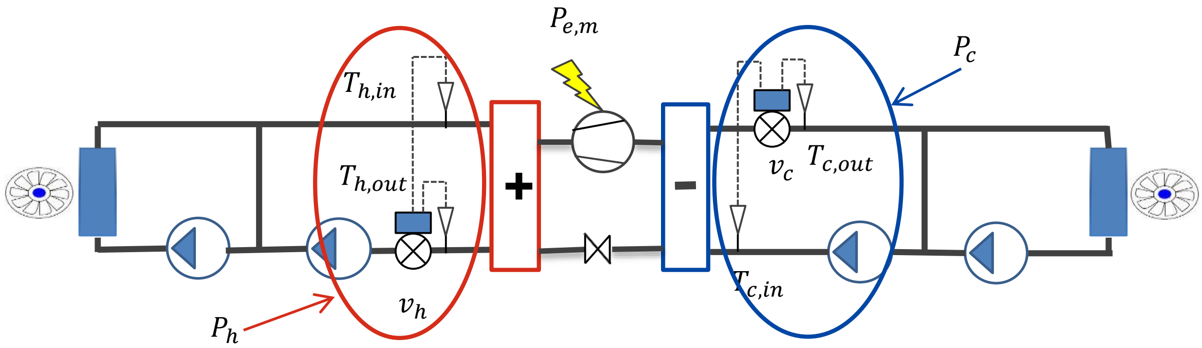

Figure 16 Measurement points for the external method

SEI can be measured both with the external and internal method. Measurement points are illustrated in Figure 15 and Figure 16 . The external method requires installed meters in the pipe system, if not non-invasive meters are used. There is also a requirement of straight inlet and outlet to get accuracy in the measurement.
The internal method require connection of meter for pressure in the refrigerant circuit. There are in many case a preparation for the connection in the refrigerant circuit, so the operation is easy if you have the right competence to do it, but there will be a small leaking of the refrigerant.
In both method temperature measurements are required for the reference temperatures and electric power from the compressor and auxiliary's according to the system boundary.
For both methods it's important to use calibrated meters with god accuracy and make a proper installation to get a measure with god accuracy.

5 Error propagation

Uncertainty in SEI measurements depends on uncertainty in used meters, the property in the installation of meters, property in used thermodynamic data and the operation of the unit that is measured. This means stability in operation.

For analysis of the uncertainty the GUM method has been used. GUM (Guide to the Expression of Uncertainty in Measurement) is the result of a joint international work with certification, standardization and research organizations.

SEI consists of COP and COP Carnot. For heat pumps with the internal method, it can be expressed according to the equation:

$$SEI_{h,1} = \frac{COP_{h,1}}{COP_{C,h}} = (1-f)\frac{\frac{(h_2-h_7)}{(h_2-h_1)}}{\frac{T_1}{T_1-T_2}} \qquad \text{Eq. 33}$$

In previous works the uncertainty of COP measurements with the internal method has been worked out. To evaluate according to [21,22], the uncertainty for a proper installation with the internal method can be +/- 5%.
This analysis will therefore focus on the uncertainty in measurements when determining Carnot COP.
The following expression will give the deviation in COP carnot for a certain deviation in the temperatures T_1 and T_2.

$$\Delta COP_{C,h} = \frac{\partial COP_{C,h}}{\partial T_1}\,\Delta T_1 + \frac{\partial COP_{C,h}}{\partial T_2}\,\Delta T_2 \qquad \text{Eq. 34}$$

For heat pumps the Carnot COP is calculated from:

$$COP_{C,h} = \frac{T_1}{T_1-T_2} \qquad \text{Eq. 35}$$

Using the following rules

The product rule: $D(fg) = f'g + fg'$ Eq. 36

The division rule: $D(f/g) = \frac{f'g-fg'}{g^2}$ Eq. 37

Using the rules above gives the following sensitivity factors:

$$C_1 = \frac{\mathrm{dCOP}_{Ch}}{\mathrm{dT}_1} = \frac{1}{T_1-T_2} + \frac{T_1*(-1)}{(T_1-T_2)^2} = \frac{(T_1-T_2)}{(T_1-T_2)^2} - \frac{T_1}{(T_1-T_2)^2} = \frac{-T_2}{(T_1-T_2)^2} \qquad \text{Eq. 38}$$

$$C_2 = \frac{\mathrm{dCOP}_{Ch}}{\mathrm{dT}_2} = \frac{T_1*(-1)(-1)}{(T_1-T_2)^2} = \frac{T_1}{(T_1-T_2)^2} \qquad \text{Eq. 39}$$

For cooling Carnot COP is calculated from:

$$COP_{C,c} = \frac{T_2}{T_1 - T_2} \qquad \text{Eq. 40}$$

Giving the following sensitivity factors:

$$C_1 = \frac{\mathrm{dCOP}_{C,c}}{\mathrm{dT}_1} = \frac{T_2 * (-1)}{(T_1 - T_2)^2} = \frac{-T_2}{(T_1 - T_2)^2} \qquad \text{Eq. 41}$$

$$C_2 = \frac{\mathrm{dCOP}_{C,c}}{\mathrm{dT}_2} = \frac{1}{T_1 - T_2} + \frac{T_2 * (-1)}{(T_1 - T_2)^2} = \frac{(T_1 - T_2)}{(T_1 - T_2)^2} - \frac{T_2}{(T_1 - T_2)^2} = \frac{T_1}{(T_1 - T_2)^2} \qquad \text{Eq. 42}$$

These factors, C_1 and C_2, are used in the calculation of the standard deviation according to:

$$uc^2(y) = C_1^2 u^2(x1) + C_2^2\, u^2(x2) \qquad \text{Eq. 43}$$

where *uc(x)* is the name of the combined standard uncertainty, and c stands for combined.

For the sensitive factors it is considered that the reference temperatures are measured with one sensor, not the mean value of two sensors, which is the case in liquid/liquid measurements in this report.
The difference in uncertainty for COP and SEI is the measurement of reference temperatures for Carnot COP. With good sensors and application, the result probably would be at the same level as the COP measurement. More evaluation is needed for the uncertainty to use it in practice.

In Table 2 some values for uncertainty are presented for sensors used in ClimaCheck measurements. The deviation of COP is based on [21,22].

Table 2 Example of uncertainty for measurement equipment for SEI

Uncertainty for PT1000	0.15+0.002*T
Deviation from measured COP, internal method	5%

6 Case-studies and practical results

The systems for analysis have been chosen among measurements available from the project partners ClimaCheck and Ambient Control LTD. The measurements chosen have been units that are fairly simple to evaluate. Among the systems there are one ground source heat pump and one chiller both placed in Sweden. There are also two chillers placed in UK. Besides field measurement the analysis with SEI is also done with data from laboratory measurements on ground source heat pumps.

A unit that is fairly simple to evaluate is a liquid/liquid unit without extra equipment that can complicate the analysis. Data for stable conditions have been chosen, and when possible, for different operating conditions. Measurement data has in these cases been collected and analysed with ClimaCheck measurement equipment and program. The equipment is in most cases permanently installed in systems by trained contractors. The data has been downloaded from ClimaChecks database for analyse.

The analysis in these chapters is in most cases based on measurements with the internal method, described in chapter 3.2.3.1.Some of the evaluated data is from short time measurements done with ClimaCheck portable. The measurements done in laboratory are done with the external method.

6.1 Analysis of heat pump systems

6.1.1 Ground source heat pump, Sweden

The analysis of a ground source heat pump [1] has been done with data from measurement on unit placed in Sweden. The heat pump is used for space heating in a building that holds both offices and apartments. The building also has a need for cooling and the borehole is used for free cooling during the summer period. During a part of the summer when the borehole is too warm, the heat pump is also used for cooling through the evaporator (not by reversed cycle). The heat transportation is liquid based on both sides of the heat pump (the heat transfer media is ethanol on the cold side and water on the warm side).

The SEI_{h1} (for heating) has been calculated for the process during several cycles. With this system boundary the refrigeration circuit is analysed. The reference temperature on the cold side is the average temperature of the incoming and leaving heat transfer media (ethanol), see definition for indirect systems in chapter 3.2.1. On the warm side the reference temperature also is the average temperature of the incoming and leaving heat transfer media (water). The calculated SEI and COP are shown in Figure 17 against the temperature difference between the reference temperatures. The figures are based on cycles during the heating period. The figure shows that the $SEI_{h,1}$ is on a stable level during the heating mode and the system is also designed primarily for heating.

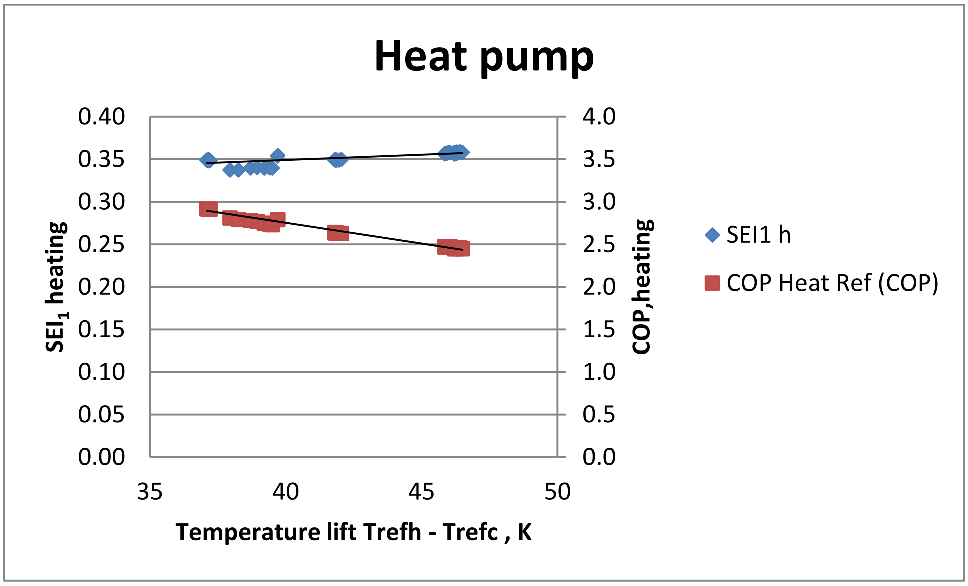

Figure 17: **SEI_{h1} heating and COP_{h1} heating for a ground source heat pump**

During space heating operation, the temperature on the cold side varies around 0 °C in an interval from -2 °C up to 5 °C. The reference temperature on the warm side during heating period varies between 37 °C up to 45 °C. All reference temperatures are the mean value of the in-coming and leaving temperature of the heat transfer media.

In the Figure 17 the difference in variation for SEI and COP is shown. The SEI value is stable around 0.35 and slightly higher at the higher temperature lifts. This means that the heat pump has an even performance in the working area, with slightly better performance for higher temperature lift. The COP is dependent on operating conditions and declines from a higher level at a low temperature lift to a lower level at high temperature lift. In chapter 0 the heat pump will be compared with other measurements to find out more about the efficiency of this unit.

6.1.2 Ground source heat pump lab test

An evaluation of SEI_{h1} has been done based on laboratory measurements of ground source heat pumps. The size of the heat pumps are typically for space heating in single family houses in Sweden. The laboratory measurement is based on EN14511 and done with the external method.

In the diagram in Figure 18, SEI_{h1} is calculated for six test point for each one of the eight ground source heat pumps. The measurement is done at different test points, where the heat pump deliver 35, 45 or 55 ºC to the heating system according to EN 14511. The difference between the test point ‘rad’, which means heating by radiators, and ‘floor’, which means floor heating, is the temperature difference and flow of the heat transfer media. In all cases except the test point -5/55, the temperature from the borehole is 0 ºC. In the evaluation the reference temperatures is calculated as mean value of the in-coming and leaving heat transfer media temperatures. For heating with radiators the flow is lower than for floor heating. This results in a lower reference temperature for the radiator heating.
The diagram shows the profile for the heat pumps according to performance. All the heat pumps in the test are qualified as ‘good’ performing heat pumps, even if there is a spread between the best and poorest.

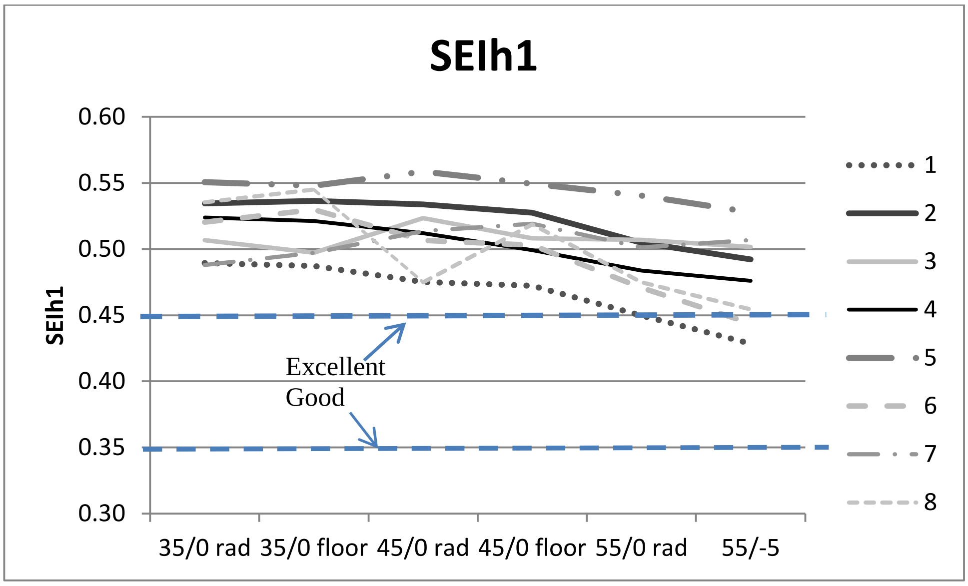

Figure 18 Test result for eight different heat pumps.

According to the scale that is established in 0 the values over 0.35 are good. Most of the heat pumps above are even excellent. Based on the different measurement points it can be noted that some of the units performance varies with the flow, for example unit 3 will have a lower SEI in the floor heating case with higher flow rate whereas unit 8 works the other way around. Unit 5 has a quite stable SEI_{h1} in all test points. The last test point of 55/-5 is typically closer to the edge of the working range for the heat pumps and thus, the SEI values are falling for most of them.

To compare the benefit of SEI according to COP three of the heat pumps are put into separate diagrams to show the difference. In Figure 19 the COP values for the three heat pumps is plotted. COP for all heat pumps varies with the same behavior according to the temperature lift. In the diagram in Figure 20 the SEI-value for the same measurement as in diagram in Figure 19 is presented.

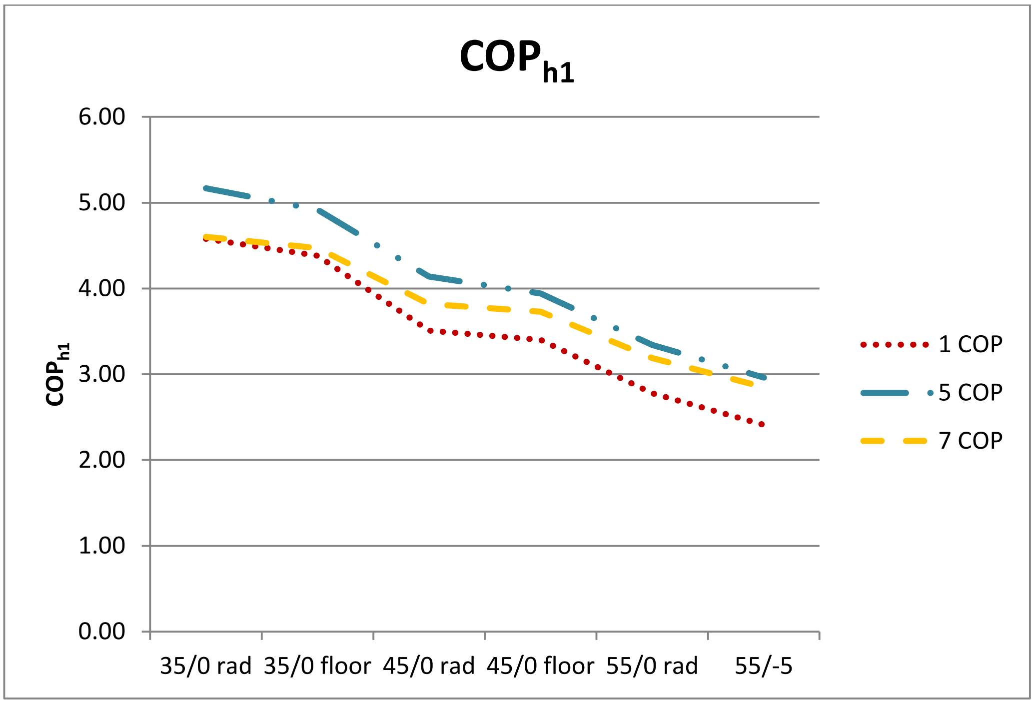

Figure 19 COP_{h1} for three different heat pumps tested in laboratory

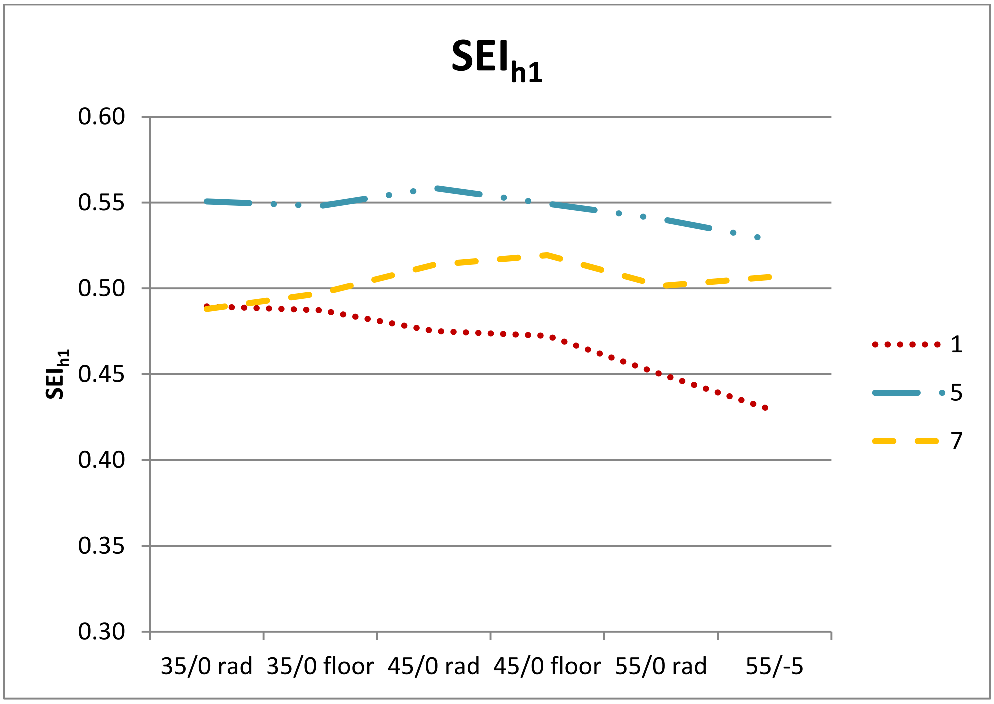

Figure 20 SEI_{h1} for three different ground source heat pumps tested in laboratory

The individual ranking between the heat pumps is the same with SEI as for COP, with number 5 at the highest level, but it is easier to see that number 1 is losing in performance at high temperature lifts and that the performance of heat pump number 5 and 7 is more even in the whole working range. This information together with information about the SEI can make it possible to regulate the system in a more optimal way.

6.2 Analysis of air conditioning system

6.2.1 Chillers for air conditioning, Sweden

Measurements on a system for air conditioning [1]have been used for analysis of SEI_{c1}. The system is placed in Sweden and consists of two units with two circuits in each. The presented results are from measurements in one circuit in one of the units, the SEI_{c1} is shown in Table 3 and Figure 21. However, measurements for the other circuit and units have been evaluated too.

The SEI should by the definition be constant if the unit works properly in all cases. Here, the SEI is lower when the temperature lift is low, even though the COP shows high values. A deeper analysis of the measured data gives the explanation. In the Table 3 some of the measured values are presented. When looking at the values during low temperature lifts, see line 1, they will show a high leaving temperature of the heat transfer media on the cold side, $T_{c1,out}$. The superheat is also found high. A conclusion of these parameters is that the expansion valve in the unit doesn't work well at higher evaporating temperatures.

The difference between the condensing temperature and the reference temperature on the hot side, $T_{ref\ h1}$, has been calculated and added to the difference between the reference temperature on the cold side, $T_{ref,c}$, and the evaporating temperature. This parameter says something about the heat exchangers. When the difference is high the SEI is low.

Table 3 Measurement points for one circuit in an AC unit. Point 1 to 4 is half load and point 5 to 11 full load

	$T_{c1,in}$ Deg C	**$T_{c1,out}$ Deg C**	**Super-heat Deg C**	**$T_{ref,c1}$ Deg C**	**$T_{ref,h1}$ Deg C**	**Temp diff in heatexchanger Deg C**	**Temp. lift, Deg C**	**COP_{c1}**	**SEI_{c1}**
1	14.6	11.5	7.7	13.1	29.1	19.3	16.1	4.15	0.23
2	10.2	7.5	6.5	8.9	28.9	17.9	20.1	3.79	0.27
3	8.6	5.9	6.2	7.3	27.7	17.4	20.4	3.77	0.27
4	9.3	6.6	5.8	8.0	29.4	17.3	21.5	3.67	0.28
5	11.9	6.8	5.2	9.4	32.2	16.7	22.9	3.62	0.29
6	11.1	6.3	5.9	8.7	32.9	17.0	24.2	3.50	0.30
7	11.2	6.5	6.3	8.9	33.4	16.7	24.6	3.43	0.29
8	11.0	6.3	6.2	8.7	33.3	16.7	24.6	3.43	0.30
9	10.9	6.3	6.2	8.6	33.3	16.6	24.7	3.43	0.30
10	10.7	6.0	5.9	8.4	33.3	16.4	25.0	3.40	0.30
11	10.7	6.0	5.8	8.4	33.6	16.4	25.3	3.38	0.30

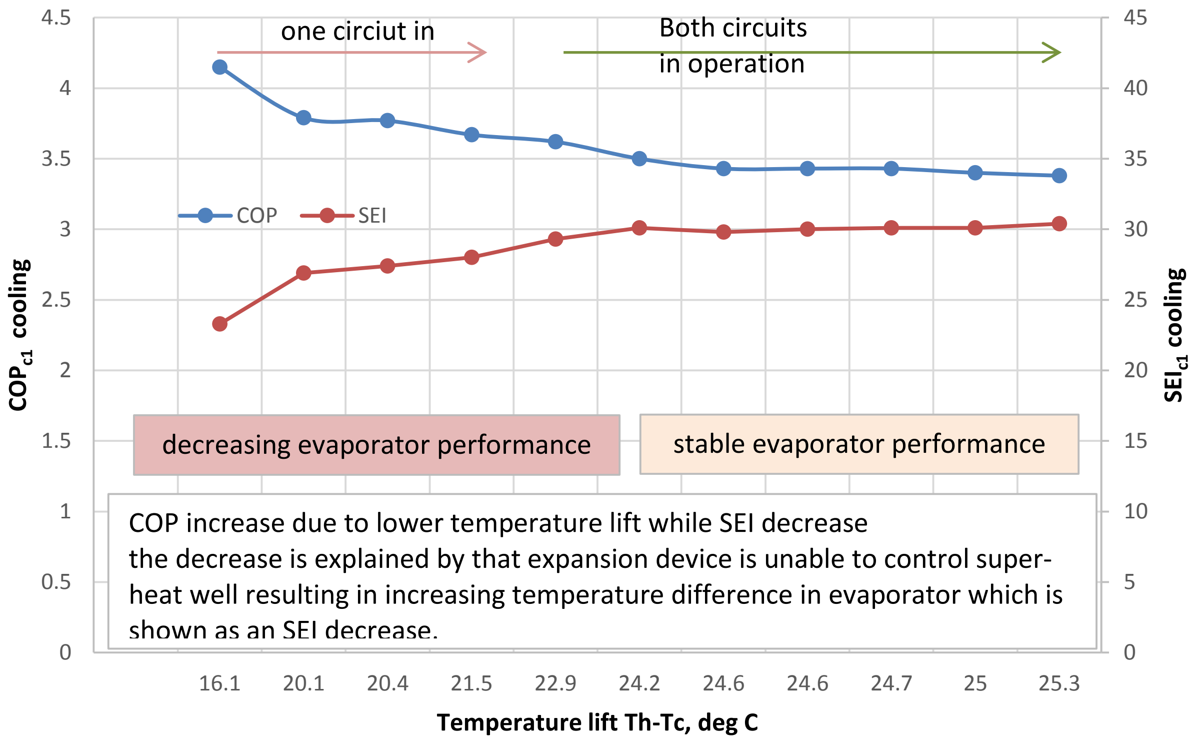

Figure 21 : SEI_{c1} and COP_{c1} for an air conditioning unit, values from **Table 3**. Please note that the SEI values are here multiplied by a factor 100.

6.2.2 Chillers in England

Two chillers [2] placed in UK have been analysed for SEI_c as well. The measurement used were made on two R134a water cooled chillers. The systems are similar, but the units have different capacity. Both of them have been measured at some different operating conditions.

Table 4 contains data from the chillers measured with ClimaCheck. Here, some of the sub-efficiencies of different components are also analysed. They are described further in Appendix 1. One of the sub-efficiencies η_2 is the compressor efficiency, it is calculated from system pressure and temperature values using the internal method. The heat exchanger efficiency η_4 is the ratio between Carnot COP based on condensing and evaporating temperatures and Carnot COP based on reference temperatures on hot and cold side:

$$\boldsymbol{\eta_4} = \frac{T_2}{(T_1 - T_2)} \bigg/ \frac{T_{ref,c}}{(T_{ref,h} - T_{ref,c})} \qquad \textbf{Eq. 44}$$

Assuming that a water cooled chiller ultimately rejects heat to ambient air via a cooling tower, working with condenser water inlet and outlet temperatures it can be considered as treating part of the total system. Ultimately, for comparison with air cooled equipment, the air temperature would become the reference point and SEI_{c2} would need to account for cooling tower and water pump power.

Table 4 Measured data for two R134a water chillers, reference temperatures based on condenser water inlet and outlet

System	Cooling Capacity kW	Evap temp, mid-point °C	Cool reference temp. °C	Evap temp difference K	Warm reference temp. °C	Cond temp, mid-point °C	Cond temp difference K	Compr efficiency %	Heat exchanger efficiency %	COP_{c1} Cool	SEI_{c1} %
Chiller 1 full load	180	0.3	10	9.7	43.4	54.5	11.1	63.2	59.5	2.42	28.5
Chiller 1 part load	115	4.1	9.8	5.7	44.1	52.3	8.2	47.6	69.8	2.13	25.9
Chiller 2 full load	375	-3.4	2.2	5.6	40.4	46.2	5.8	66.6	75.5	2.86	39.8
Chiller 2 part load	150	-0.7	2.2	2.9	44.2	47.6	3.4	59.6	86.1	2.59	39.5
Chiller 2 part load	115	0.4	2.8	2.4	43.6	46.3	2.7	53.6	88.2	2.49	36.9

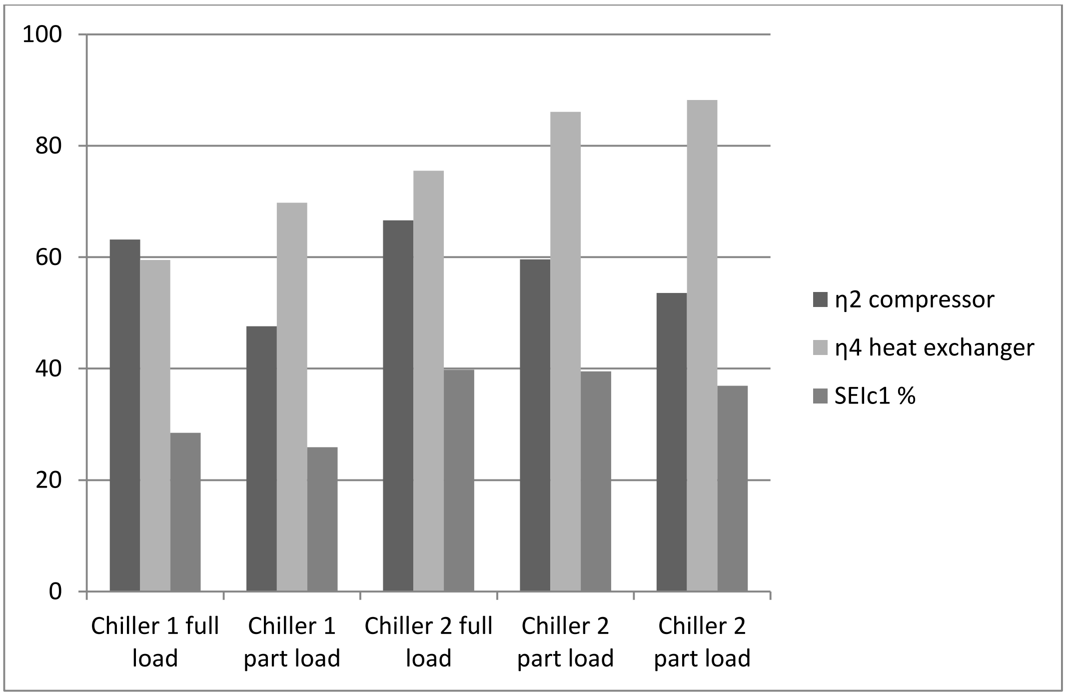

Figure 22 Performance for Chiller 1 and 2

Chiller 2 performs significantly better than chiller 1. Both condenser and evaporator temperature differences are smaller. Also the compressor efficiency is better at full load. This is reflected in a significantly better SEI (40%) whereas the COP is only 15% better. This evaluation and comparison of SEI and sub efficiencies can be done although differences in operating conditions that would by themselves result in significant differences in COP. This would make it irrelevant to compare the two systems based on COP.
There is a clearly noticeable drop in compressor efficiency when operating at part load. In the case of Chiller 1 it falls from 63.2% to 47.6% and the capacity is partly offset by a reduction in condenser and evaporator temperature differences. The net result is a reduction

of SEI. The increase in COP is misleading because with a better part load efficiency compressor, the COP should be significantly higher. For Chiller 2 the drop in compressor efficiency is much less and the reduction in condenser and evaporator temperature differences is less even with the larger capacity decrease due to the much better performance at full load with the result that the change in SEI is small.
Chiller 2 is a much better performing system than Chiller 1 at all the measured conditions, but this is not apparent from the COP value where the differences in performance is hidden by the differences in chilled water temperature.
Further analysis with more data, together with quantification of the effects of the heat exchangers and other system losses is planned so that benchmark SEIs for different system types can be investigated.

6.3 Analyse of sub-efficiencies examples

Here follows an example with several sub-efficiencies is shown in Figure 23. The system uses R407C as refrigerant and the following parameters are measured:
Evaporating pressure 3.72 bar (temperature midpoint -8.3, dew point -6°C)
Condensing pressure 14.75 bar (temperature midpoint 35.8, dew point 38.4 °C)
Cold heat exchanger media inlet temperature: 0°C
Warm heat exchanger media outlet temperature: 35°C
Subcooling: 1.3K
Superheat: 5K
Cooling capacity: 11.6kW (COP 3.3)
Heating capacity: 14.9kW (COP 4.25)
Compressor electrical power input: 3.5 kW
Thermal efficiency: 0.95

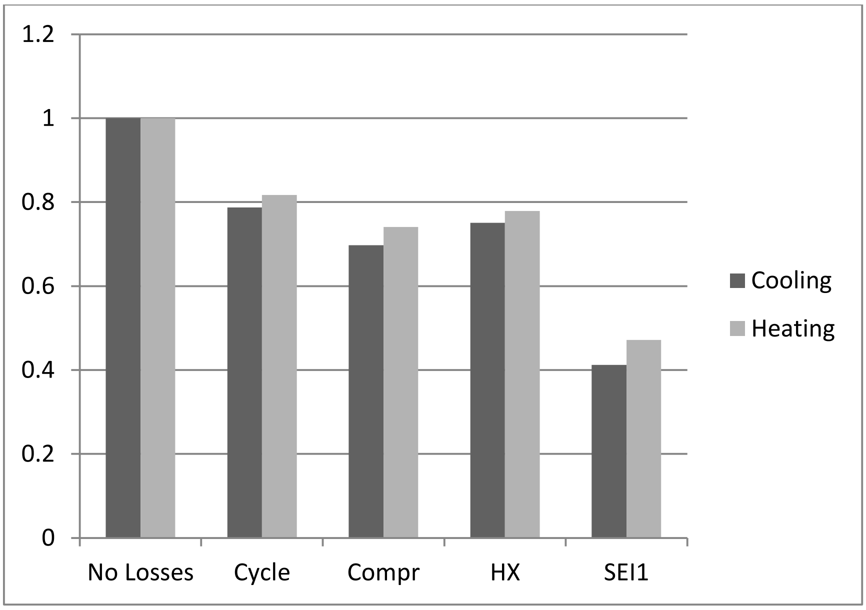

Figure 23 ***System sub efficiencies and SEI$_{c1}$ value for an R407C Heat Pump***

Figure 23 represents the sub efficiencies (Heat exchangers combined effect, pressure drops negligible), and the final SEIc$_1$ values. These are calculated for the same measured data, i.e. the system is not reversed, just considered in terms of cooling value or heating value.
In Figure 24 two liquid/water heat pumps/chillers are compared and it can clearly be seen the consistency of the SEIc$_1$ over a significant envelope whereas the COP cannot be used in any easy to explain way to compare system performance or highlight where the short comings of system is. This example clearly show the generally poor performance of

Compressor in System I as well as that the evaporator at high temperature on cold heat transfer media the evaporator is losing performance. Due to the characteristics of the refrigeration process the evaporator performance impact will decrease at lower Heat transfer media temperatures on the cold side due to the decreasing capacity of the compressor. This is all as expected and it high-lights the need to apply and compare systems in the envelope they are intended to be used. SEI offer a new option to highlight how well adopted a design is to a specific condition and also in the field validating how well it works and where the losses occur.

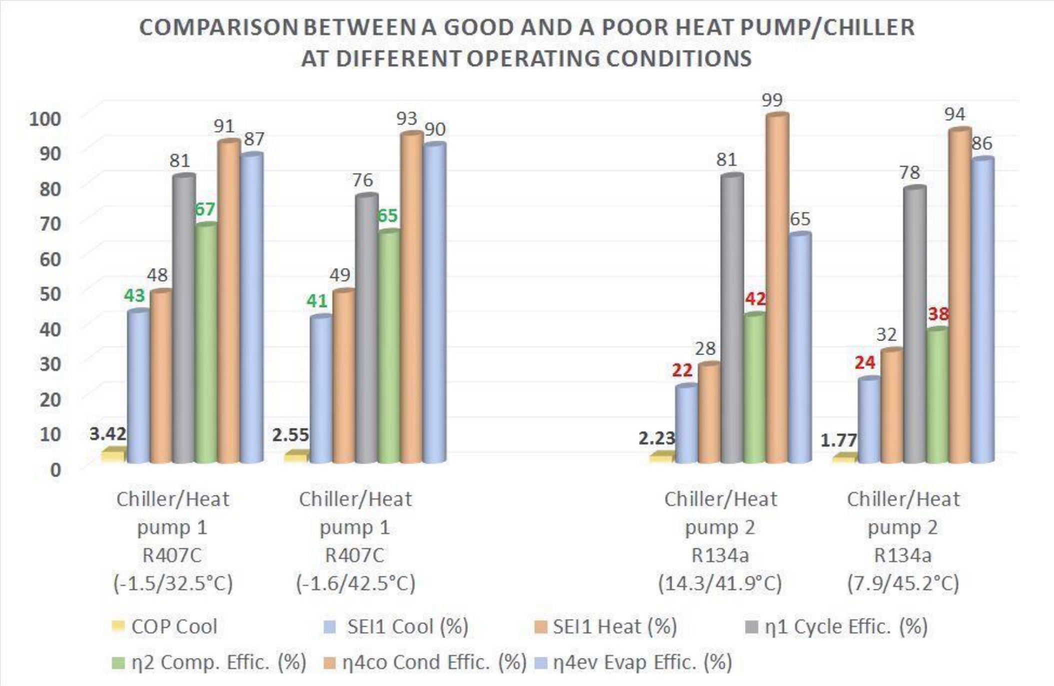

Figure 24: ***Example of two brine to water heat pumps/chillers one with relatively high performance and one with low performance showing total SEI_1 for cool and heat as well as sub efficiencies high-lighting the cause of the decreased performance.***

6.4 Intervals for SEI and scale from good to poor performance

Based on evaluated measurements a hypothesis about SEI_1 intervals from poor to excellent has been proposed. In Table 5 measurements presented in this report is summarized and given a review. This review is based on experience and knowledge about the measured units. All measured units are liquid/liquid, and the application can be heat pump or air conditioning. It would according to this circumstances be relevant to compare them with the same scale. More measurements need to be evaluated in combination with sub efficiency and known circumstances to give accuracy to the scale.

Table 5 Summary of measured SEI1 values in the report, here all SEI values are multiplied by a factor 10.

Measurement	SEI_{c1}	SEI_{h1}	Scale
From Table 4 Chiller 1	25-28		Poor
Chiller 2	37-40		Good
From Figure 18 Heat pumps		>40	Good and Excellent
Figure 17 Heat pump		35	Poor –good
Figure 21Table 3Chiller	23-30		poor
Figure 23 Heat pump	40	45	excellent
Figure 24good heat pump	41-43	48-49	Excellent
Figure 24 poor heat pump	22-24	28-32	Poor

A summarize of the review in Table 5 gives proposal for scale presented in table Table 6and Table 7.

Table 6 Scale hypothesis: for SEI1 heating

SEI Limit	Grading	Action
<.2	Unacceptable	Investigate
0.2-0.35	Poor	Investigate further with sub efficiencies
0.35-0.45	Good	
>0.45	Excellent	

Table 7 Scale hypothesis: for SEI1 cooling

SEI Limit	Grading	Action
<.1	Unacceptable	Investigate
0.1-0.3	Poor	Investigate further with sub efficiencies
0.3-0.4	Good	
>0.4	Excellent	

7 Harmonization – SEI as a tool for energy follow-up

In this project evaluation of measurement data and the work with definitions of the SEI has showed that the key performance indicator can be useful as a tool for evaluation of different refrigeration and heat pump processes in instantaneously measurements. With continuous work it can be possible to develop the method further and make it a useful tool for installers and service companies in different countries for analysis of performance and investigation of heat pump and air-conditioning systems and compare performance with manufacturing data. The SEI presented here could be an easy-to-use tool for energy follow up that can be used refrigeration and heat pump systems.

Since the methodology has been co-developed by German and UK partners, there is a good possibility to make this an industry standard in regular check and follow-up of installed systems' performance. The methodology's KPI index can also be a benchmark for facility owners in the commissioning of new installations. Results from the project has been discussed and presented in the participating countries, as well as to the VDMA and IOR organisations.

8 Discussion

The aim of the project was to develop a method for measurements and follow up to support energy optimization of air-conditioning and heat pump systems. In this work the performance indicator SEI (System Efficiency Index) has been evaluated and further developed. This performance indicator show the energy optimization potential and hence, if a system has a good or poor performance. Previously, SEI has been used by IOR and VDMA for dimensioning purposes of refrigeration systems. With the developments achieved through this project, performance evaluation of heat pump and air conditioning systems is possible with data from field or laboratory measurements.

Typically, COP is used as a key performance indicator for heat pump and air conditioning systems. The purpose of SEI and COP is different and they supplement each other. COP has a weakness as comparator as it is strongly dependent on operating conditions.

COP answers the question about how **much energy** is used for production of cooling and heating in a specific measurement point. The indicator is specific for the measurement point and it is hard to relate to other circumstances.

SEI answers the question how **efficient** the process is in the same point. The measured value can be compared with values for other conditions. In this way SEI is a general indicator. The difference tells about the performance in the measured point according to ideal performance, other measured points or dimensioning data. It shows the potential for optimization and the quality of COP.

For an ideal cooling process that works well in all operating modes the difference of COP and SEI can be described like this. The characteristic of COP is increasing with decreasing temperature lift. The characteristic of SEI is relatively constant in the operating range, however an optimum can be found. Near the limit of operating range the SEI typically drops. This means that a variation in the operation mode is easier to notify with SEI than with COP. But still the COP tells about the energy performance in the specific operating point.

To be able to compare evaluated SEI-values, stable conditions are needed for the system. This means that the unit should operate at an even level in balance with the system. In part-load cases this can be hard to find.

Measurements have been evaluated and analyzed in the project and the results shows that the SEI has the strength to be possible to be comparable in a wider range than COP. The proposed scale in 0 is a start to get something to compare measurements with. In the work with analysis it has been clear that it is important to define system boundaries properly, especially when wider system boundaries such as SEI_2-SEI_4 is used. Systems has many different solutions, for delivering of heat and cool, which gives many possibilities for system boundaries. One conclusion is that level one and two is most suitable to compare between systems and that higher levels, if they are measurable, are more useful for analysis of one system.

9 Conclusion

The aim of the project has been to develop a methodology for evaluation of short term measurements for heat pump and air conditioning systems, used for reliable and energy optimization. This have been done by evaluation of the key performance indicator System Efficiency Index, SEI, earlier defined in previous works done by IOR in England and VDMA in Germany.
The project group have consisted of partners from Sweden, UK and Germany, connected to VDMA and IOR. Clima Check is a partner in the project group and have contributed with measurements for evaluation.
A literature review on key indicators, measurement methods and standards on energy efficiency for heat pumps and air conditioning units has been done. The key indicator in most cases used today is COP. A lack of an indicator useful for short term measurements was considered.
The key performance indicator System Efficiency Index, SEI, describes the efficiency for a heat pump or air-conditioning system according to the best possible action for the case. Values for COP, including used and delivered power for the process, and temperatures to define the theoretical best action according to Carnot COP have to be measured.

According to existing heat pump and air conditioning systems, a categorization has been done and system boundaries have been developed in four levels. SEI can be calculated from measurements according to delivered heat (SEI_h) or cooling (SEI_c). In the project focus have been on simple liquid/liquid systems according to system boundary one. This boundary includes the refrigeration process in the unit, with reference level for COP Carnot in the mean value of temperatures in the heat exchanging medias on the cold and hot side respectively.

Methods for measurement of the SEI have been described in the report. There are two main methods, the external and internal method. The difference is how the cooling and heating capacity is measured. The external method uses flow meters and temperature sensors in the heat exchanging media for calculation of the capacity. The internal method is based on measurements of pressure and temperature in the refrigerant circuit. To get the SEI, measurements of electric power to the compressor and auxiliaries included in the system boundary also is needed. Temperatures according to system boundaries have to be measured for calculation of theoretical best action according to Carnot COP.

Evaluation of laboratory and field measurement data for heat pumps and chillers has been done. The evaluation shows that SEI is a useful indicator that contributes with information about the performance of the system. Using sub-efficiencies, that shows the contribution from different parts in the refrigerant process, the evaluation can get even further. A scale for identification of good performance in the system have been defined, based on measurement results.

For a cooling and heat pump processes that works well in all operating modes the difference of COP and SEI can be described like this. The characteristic of COP is increasing with decreasing temperature lift. The characteristic of SEI is relatively constant in the operating range, however an optimum can be found. Near the limit of operating range the SEI typically drops. This means that a variation in the operation mode is easier to notify with SEI than with COP. But still the COP tells about the energy performance in the specific operating point.

The conclusion of the project is that SEI is a good key indicator for heat pump and air conditioning system that is useful for field measurements. It gives new information about the performance that previous indicator as COP can’t tell. COP has a weakness as comparator as it is strongly dependent on operating conditions.

COP answers the question about how much energy is used for production of cooling and heating in a specific measurement point. The indicator is specific for the measurement point and it is difficult to relate to other circumstances.

SEI answers the question how efficient the process is in the same point. The measured value can be compared with values for other conditions. In this way SEI is a general indicator. The difference tells about the performance at the measured point according to ideal performance, other measured points or dimensioning data. It shows the potential for optimization and the quality of COP.

To develop SEI even more will give a power full tool for reliable and energy optimization.

10 Further work

According to the result of this project, SEI is a useful indicator for momentarily measurements on heat pump and air conditioning systems. To get the method even more useful, further work with analysis of more systems has to be done to establish the scale and make it useful for more system types. In this project for example air to air systems has not been analyzed.
SEI can be a useful indicator both in construction and evaluation. The connection between these cases can also be developed in further works.
To analyze the same type of unit in laboratory and field measurement would give more accuracy to the method. See 6.1.2 for more comments.

11 Publications from this project

Scientific publications

Lane, A-L., Benson, J., Berglöf, K., 2014. Benefits of Establishing System Efficiency Index during field measurements on air. 11th International Energy Agency HEAT PUMP CONFERENCE 2014: Global Advances in Heat Pump Technology, Applications & Markets, Montreal, Kanada

Lane, A-L., Benson, J., Hundy, G., Berglöf, K., 2014. System Efficiency Index, SEI, a key performance indicator for field measurements of refrigeration, air conditioning and heat pump systems. 3rd IIR International Conference on Sustainability and the Cold Chain, London, UK

Publications of popular science

Berglöf, K., 2012. Systemeffektivitetsindex bättre än COP (System Efficiency Index better than COP). Kyla+ Värmepumpar nr 2/2012, p 56-57

Berglöf, K., 2012-04-01. Press release to Swedish and international trade and cleantech press, Published in 10+ medias and several articles

Presentations

Kuhl, H., 2012-04-04, Glauchau, Presentation of the project to several teachers and students

Hundy, G., Berglöf, K., 2012-04-05, London, IOR Meeting with the SEI work group

Berglöf, K., 2012-04-13, Business Edge, Hampshire UK, ClimaCheck International Training performance analysing, presentations to experts from 7 countries

Kuhl, H., 2012-05-21, Frankfurt (M), Presentation of the project at VDMA-Working group meeting about VDMA24247

Kuhl, H., 2012-06-20, Lindau, Presentation of the Project to several companies and 2 high schools

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Planned activities
A popular science article in Kyla is planned during the autumn 2014

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Appendix 1 - Principles of the thermodynamic evaluation of sub-efficiencies

This Appendix describes the sub-efficiencies more thoroughly. It is an extract from a document, worked out mostly by Klas Berglöf and Guy Hundy.

According to previous chapters the SEI is calculated from instantaneous measured values in field. The SEI value gives an indication about the system performance in total. In order to analyse the performance of a system further and to find reasons for poor performance, the SEI sub-efficiencies can be used. Here is shown how SEI_{c1} can be divided into six sub-efficiencies each of them showing efficiency of components or parts of the refrigeration process. By analysing the sub-efficiencies it is possible localize where in the process the performance problem exits. To determine sub-efficiencies described in this chapter the internal method has to be used for measurement.

$SEI_{c,1}$ for cooling is built up of the following sub-efficiencies:

η_1: Refrigeration cycle efficiency. This sub-efficiency takes into account losses inherent in the refrigeration cycle itself, and can for example be used in the design process to compare effects of different refrigerants, or the effects of the use of an economiser.

η_2: Compressor efficiency. This sub-efficiency includes the effect of compressor efficiency. This is different for heating and cooling operation since the compressor losses (heat into refrigerant) can be part of the useful heat output during heating.

η_3: Pressure drop in refrigerant lines. This effect is probably most significant for large refrigeration systems typically with several long pipes.

η_4: Heat Exchanger efficiency. This effect includes the heat exchangers effectiveness and is extremely important in evaluating or comparing condenser and evaporator temperature differences.

η_5: Fluid transfer efficiency.

η_6: Non useful heat loss/gain. This effect includes 'non useful' heat pick up, for example heat pick up by cold suction lines in cooling applications.

Most of these effects are recognised in both the VDMA and the IOR approaches.

11.1 Description of sub-efficiencies

11.1.1 Cycle efficiency - η_1

In this section the overall scheme for defining each of the sub efficiencies that make up the total SEI is outlined.

In the basic single stage vapour compression cycle, **Figure 25**, the best possible COP that could be obtained with a perfect compressor would be the ratio of enthalpy differences

$(h_1-h_8)/(h_{2a}-h_1)$. The enthalpies can be found from a few pressure and temperature measurements and reference to refrigerant properties. This 'ideal COP' is lower than Carnot because of inherent losses in the cycle. The cycle efficiency η1 can be defined by:

$$\eta_1 = \frac{\textbf{Vapour compression cycle Ideal COP}}{\textbf{Carnot COP(reference evap and cond temps)}} \qquad \textbf{Eq. 45}$$

$$\eta_1 = \frac{(h_1-h_8)}{(h_{2a}-h_1)} * \frac{T_c-T_e}{T_e} \qquad \textbf{Eq. 46}$$

The value of η_1 provides information about the inherent loss of efficiency of the cycle with the chosen refrigerant and enables a refrigerant comparison to be made, alt
hough refrigerant properties also influence other sub-component efficiencies. Superheat and subcool will also have an impact on η_1.

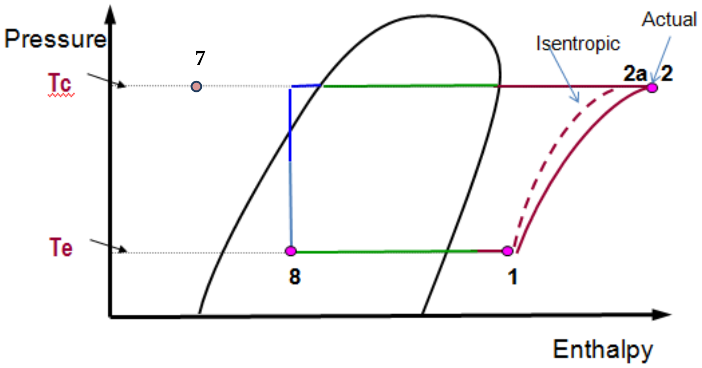

Figure 25 Simple vapour compression cycle.

11.1.2 Compressor efficiency - η_2

An actual compressor will have an isentropic efficiency <1, causing the discharge temperature, point 2, to be higher than ideal. The compressor efficiency can for a compressor <u>without</u> heat losses be found from the ratio of enthalpy differences $(h_{2a}-h_1)/(h_2-h_1)$. This ratio is the compressor isentropic efficiency η2 .

$$\eta_2 = \eta_{is} = \frac{\textbf{COP (actual compression)}}{\textbf{COP (isentropic oompression)}} = \frac{(h_1-h_8)}{(h_2-h_1)} / \frac{(h_1-h_8)}{(h_{2a}-h_1)} \qquad \textbf{Eq. 47}$$

11.1.3 Efficiency due to pressure drop - η3

All systems will have pressure drops but in most cases the pressure drop in suction and discharge line are not treated separately and become a part of the losses in the evaporator respective condenser as these will in reality experience a lower dT than that measured at the compressor. This does not affect the overall SEI but if the pressure drop is significant it can be treated separately. This does not introduce any challenges beyond knowledge of the pressures at the condenser inlet and evaporator outlet.
Pressure drop in the system increases the "lift" at which the cycle needs to operate in order to provide the required load temperatures. The sub efficiency η_3 is expressed as the ratio of Carnot COPs, COP for a system with no pressure drop and COP for a system with pressure drop. In practice pressure drop is only applicable to refrigeration systems with

long suction or discharge lines. In most other cases η_3 is close to 1 or is a part of the heat exchanger efficiency.

$$\eta_3 = \frac{\text{Carnot COP}}{\text{Carnot COPdp}} \qquad \textbf{Eq. 48}$$

Where Carnot COP is based on midpoint condensing and evaporating temperatures for a cycle operating at compressor inlet and outlet pressures and Carnot COPdp is based on actual condensing and evaporating midpoint temperatures

11.1.4 Heat exchanger efficiency - η_4

Heat exchanger, (condenser and evaporator) temperature differences increase the “lift” at which the cycle needs to operate in order to provide the required load temperatures. If pressure drop effect, η_3, is not measured separately, i.e. η_3=1, its effect will be accounted within η_4 because the temperature differences will increase with the pressure drop.
The total effect of heat transfer temperature differences is defined as:

$$\eta_4 = \text{COP Ratio} = \frac{\text{Carnot COP based on Condensing and Evap temps}}{\text{Carnot COP based on Tref,h,1 and Tref,c,1}} \qquad \textbf{Eq. 49}$$

The proportionate effect of condenser and evaporator is of significant interest to highlight where the losses are and may be evaluated, and this is dealt with in 11.8.

11.1.5 Fluid transfer efficiency - η_5

The fluid transfer efficiency, η_5 , is the ratio of compressor power and total power. It may be split according to the warm and cool sides by using η_{5h} and η_{5c}. A percentage of the ancillary power applied to the warm side may enter the secondary fluid stream and make an additional contribution to the heating capacity of a heat pump. In general, for an enclosed pump where the motor is cooled by the secondary fluid stream this percentage will approach 100%. A similar situation exists on the cold side.

11.1.6 Non-useful heat pick up or loss - η_6

The non-useful heat pick up or loss, η_6 is the ratio of evaporator cooling capacity and total cooling capacity, where the total cooling capacity includes the suction line heat pick up. This is applicable to cooling applications. For heating application it will be the impact of non-useful heat losses relative total heating capacity.

11.2 SEI cooling, the product of the sub efficiencies

The SEI cooling can be derived from the sub efficiencies:

$$SEI_{c1} = \eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \eta_4 \qquad \textbf{Eq. 50}$$

SEI_2 takes account of auxiliary power as defined in 3.2.4 and the non-useful suction line heat pick up:

$$SEI_{c2} = \eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \eta_4 \cdot \eta_5 \cdot \eta_6 \qquad \textbf{Eq. 51}$$

The SEI_{c1} (cooling) derived from sub efficiencies includes all losses and is the same as the SEI_{c1} from the basic definition in chapter 3.

11.3 Temperature definitions

For condensing and evaporating temperatures the midpoint is used for glide refrigerants in the System Efficiency Index definitions. Midpoint is in the condenser defined as the mean temperature between dew and bubble point and in the evaporator the mean value between temperature entering evaporator and dew point. The midpoint definition gives the best representation of the heat exchanger performance. This is the definition of mid-point is given in the Alternative Refrigerant Evaluation Program, AREP and used widely in the industry.
Figure 26 summarises the temperature definitions required. Heat exchanger refrigerant pressure drops are shown, although usually negligible, in which case $p_4 = p_5$ and $p_8 = p_9$.

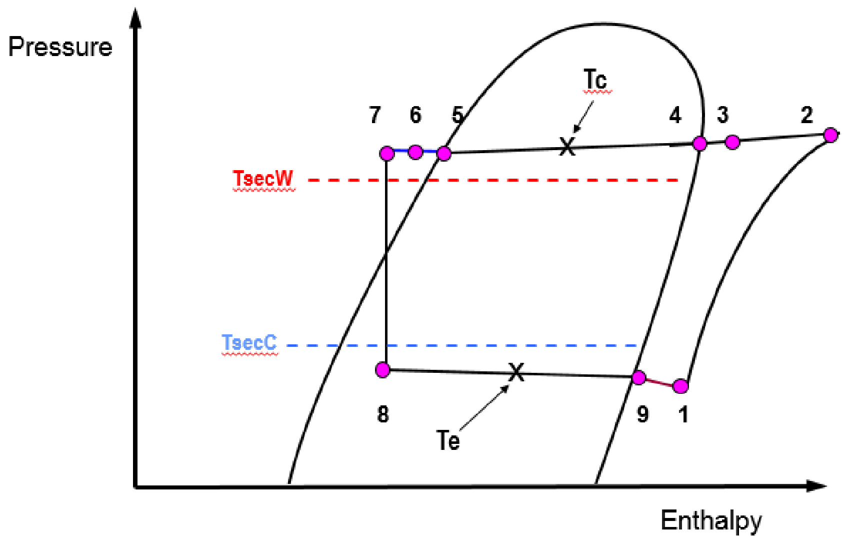

Figure 26 Diagram showing the effect of refrigerant pressure drop and heat transfer losses. The effects have been exaggerated for clarity. Secondary reference temperatures $T_{ref,h1}$, called TsecW and $T_{ref,c1}$, called TsecC are added for illustration only.

Key to Figure 26

1. Compressor inlet. Te is the midpoint temperature at pressure (1) unless it is separately defined by η_3
2. Compressor outlet. Tc is the midpoint temperature at pressure (2) unless it is separately defined by η_3
3. Condenser inlet if pressure drop and heat loss between compressor and condenser.
4. Condensation commences –dew point
5. Condensation complete – bubble point
6. Condenser outlet
7. Expansion valve inlet if pressure drop and heat loss between condenser and expansion valve
8. Evaporator inlet.
9. Evaporator outlet (will same as compressor inlet if heat pick up and pressure drop is not considered

Te evaporation midpoint = (t_8+dew point at evaporation pressure) /2.

Tc condensing midpoint = (tdew point + tbubble point at condensing pressure)/2

11.4 SEI for Heat pumps

In the above analysis the cooling COP definition is the one used. For heat pumps it is conventional to define COP in terms of the heating capacity. The analysis can be reworked in terms of heating COP and this results in higher values for SEI and all sub efficiencies because heat rejection includes input power. Both the SEI_c (cooling) and the SEI_h (heating) can be defined as described in this report. There is no straight forward conversion factor from SEI_c(cooling) to SEI_h(heating) so they need to be calculated separately. To avoid multiple ranges for what is good and bad on a component level it is practical to use the sub efficiencies defined for cooling also for heat pump sub efficiencies although these sub efficiencies will not multiply to give the SEI_h(heating). A good cooling compressor, evaporator or condenser is at a given condition good also for a heat pump. The slight difference is that Heat losses from the compressor shell has no impact on the COP_c cooling whereas they have an impact on the COP_h heating. It is considered essential to avoid working with two sets of sub efficiencies and hence use of the SEI_h(heating) as the total efficiency together with the sub efficiencies with the cooling references is not considered to have any practical impacts except that they cannot be multiplied together to achieve the SEI_h(heating) as they can with SEI_c(cooling).

11.5 Compressor heat losses

For design purposes the point 2 in Figure 2 may be found from the known compressor isentropic efficiency. This would normally be derived from the manufacturer's published data. When measuring the compressor discharge temperature the heat losses from the body or shell of the compressor must be taken into account.
The point 2b shown in Figure 3 represents the enthalpy that would be attained if the compressor is totally insulated with 100% conversion of the power input into enthalpy rise. In a practical system this does not occur because some heat from the compressor is lost to surroundings. Point 2 is the actual measured temperature T2.
The point h_{2b} cannot be measured unless compressor is insulated to reduce heat loss to near zero. Because the heat losses for normal semi hermetic and hermetic compressors without cooling are small relative to power input it has proven to be possible to use a straightforward loss factor with good accuracy. The heat loss is defined as compressor thermal efficiency, η_{th}:

η_{th} = (1 – heat loss fraction of compressor power) **Eq. 52**

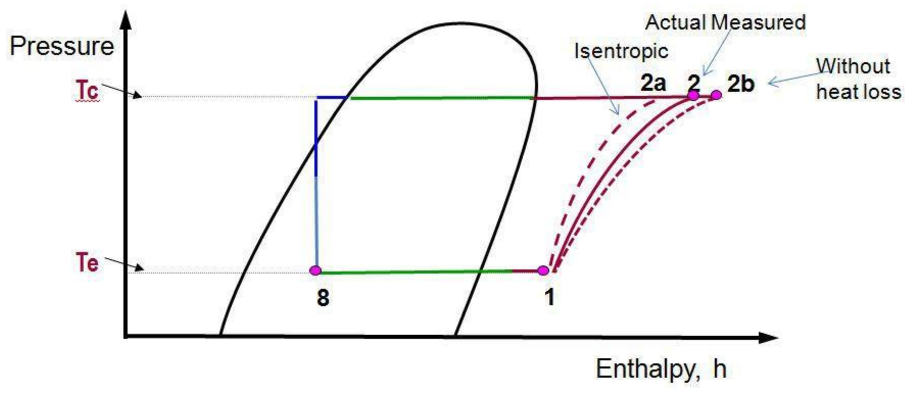

Figure 27 Simple vapour compression cycle with compressor heat loss.

The heat loss is typically 3-10% of the motor power for hermetic and semi-hermetic compressors, i.e. the η_{th} is between 0.90 and 0.97 of the power (shaft power for open compressors) results in enthalpy increase of the refrigerant. It is possible to adapt to operating conditions and compressor models but due to that a fixed value only introduce an error of a few % it is possible to use a fixed value for compressors working in normal ambient conditions without forced air over compressor shell or other means to increase cooling. A fixed value of η_{th} =93% has proven to give a good correlation for hermetic and semi hermetic compressors without cooling.
Equation 8 becomes modified by η_{th} and enables SEI to be directly determined in the field using temperature and pressure measurements in conjunction with refrigerant properties.

$$\mathbf{SEI_{c,1} = (h_1-h_8)/(h_2-h_1)*\eta_{th}*(T_{ref,h,1}-T_{ref,c,1})/T_{ref,c}} \qquad \textbf{Eq. 53}$$

For heating applications, $SEI_{h,1}$ (heating) is defined as:

$$\mathbf{SEI_{h,1} = (h_2-h_7)/(h_2-h_1)*\eta_{th}*(T_{ref,h,1}-T_{ref,c,1})/T_{ref,h,1}} \qquad \textbf{Eq. 54}$$

For practical purposes it is convenient to work with SEI as defined in equation 13 and 14 for benchmarking.
When evaluating η_2 it is necessary to modify equation (9) as follows:

$$\mathbf{\eta_2=\eta_{is}= COP\ (actual\ compression)/COP\ (isentropic\ compression) = (h_1-h_8)/(h_2-h_1)*\eta_{th}/ (h_1-h_8)/(h_{2a}-h_1) = (h_{2a}-h_1)/(h_2-h_1)*\eta_{th}} \qquad \textbf{Eq. 55}$$

11.6 Open drive compressors

For open drive motors the motor losses can be treated in a similar way by the introduction of ηm, motor efficiency.

$$\mathbf{\eta 2= (h_{2a}-h_1)/(h_2-h_1)*\eta_{th}*\eta_m} \qquad \textbf{Eq. 56}$$

Open drive compressor isentropic efficiency is conventionally based on shaft input power, whereas η_2 should always be based on electrical power input.

11.7 Effect of condenser and evaporator

The heat exchanger efficiency η_4 is of interest to analyze further to identify the efficiency of the condenser separately from the evaporator. To evaluate and benchmark the condenser impact the evaporator influence can be eliminated by introducing the evaporation temperature as the reference temperature, instead of the $T_{ref,c,1}$ in equation (11). To evaluate the evaporator separately the reference temperature on the hot side, Tref,h,1can be replaced by the condensing temperature in the same equation.

For the condenser:

η_{4co1} = Carnot COP based on Condensing and Evap tempsCarnot/ COP based on Secondary Warm and Evap temp

Eq. 57

and for the evaporator:

η_{4ev1} = Carnot COP based on Condensing and Evap tempsCarnot/ COP based on Secondary Cool and Cond temp

Eq. 58

11.8 Proportionate effect of condenser and evaporator

The above condenser and evaporator definitions will not if multiplied together equal η_4. It so desired a proportionate effect of condenser, η_{4co} and evaporator, η_{4ev} can be defined so that:

$\eta_{4co} * \eta_{4ev} = \eta_4$ Eq. 59

In order to establish the proportionate effect of condenser and evaporator, the effect of each alone, η_{4co1} and η_{4ev1} is first established.
The proportionate effect is found by use of a factor, k defined such that:

$\eta_{4co} = \eta_{4co1}$ x k

and

$\eta_{4ev} = \eta_{4ev1}$ x k

η_{4co} x $\eta_{4ev} = \eta_{4co1}$ x k x η_{4ev1} x k $= \eta_4$

$k = \sqrt{(\eta_4/(\eta_{4co1}*\eta_{4ev1}))}$ Eq. 60

$\eta_{4co} = \eta_{4co1}* \sqrt{(\eta_4/(\eta_{4co1}*\eta_{4ev1})}$ Eq. 61

$\eta_{4ev} = \eta_{4ev1}*. \sqrt{(\eta_4/(\eta_{4co1}*\eta_{4ev1}))}$ Eq. 62

The COPs and COP ratios are summarized in Table 8.

Table 8 ***Heat exchanger Carnot COP parameters and efficiencies for cooling systems***

Carnot COPs	**Temperature Ratio**	**Temperature differences**
Carnot COP with zero Temperature differences	Te(Tc−Te)	No TD
Carnot COP with Cond. Temperature difference	$Te(T_{ref,h,1}-Te)$	Cond TD
Carnot COP with Evap. Temperature difference	$T_{ref,c,1}(Tc-TsecC)$	EvapTD
Carnot COP with both differences	$T_{ref,c,1}(T_{ref,h,1}-T_{ref,c,1})$	BothTD
CARNOT COP RATIOS		
η_{4co1} COP Ratio	$Te(Tc-Te)\ .\ (T_{ref,h,1}-Te)Te$	No TD/ Cond TD
η_{4ev1} COP Ratio	$Te(Tc-Te)\ .\ (Tc-T_{ref,c,1})\ T_{ref,c,1}$	No TD/ Evap TD
Combined effect of HX temperature differences, η_4	$Te(Tc-Te)\ .\ (T_{ref,h,1}-T_{ref,c,1})T_{ref,c,1}$	No TD/ Both TD

The temperatures should be defined according to the Guideline SEI Measurements:
- Te, Tc are midpoint evaporating and condensing temperature
– $T_{ref,c,1}$, $T_{ref,h,1}$ are evaporator and condenser secondary fluid reference temperatures, defined per application
These are boundary conditions applicable to SEI_1 and SEI_2 that must be respected during any system comparison.
Further evaluation is required to evaluate the benefits of using the n_{4co1} and n_{4ev1} or the proportionate effect as described above for evaluation. The differences in value are not

large and for a given type of system either could be used for benchmarking but to create universal benchmarking parameters there can be advantages and disadvantages with the different definitions.

Appendix 2 - Basic principles of measurement

Irrespective of the chosen method (internal or external) to determine SEI, there are a number of physical quantities that need to be measured, e.g. temperature, pressure, flow rate, composition of liquid cooling agents, humidity, electric power and energy. These quantities are required as inputs to the calculation of derived quantities such as coefficient of performance and heating or cooling capacity but also as specification of the operating conditions of a test.

Measurements can be made by means of invasive or non-invasive techniques (i.e. inserting a sensor directly into the measured medium or measuring through an intermediate medium). Examples of invasive measurement are flow meters and temperature sensors installed directly in a fluid flow or electric power meters installed without measuring transformers while non-invasive examples are externally mounted flow meters (e.g. clamp-on ultrasonic types), surface temperature sensors for pipes and electric power meters installed via external transformers. This chapter provides deeper information on the measurement of temperature, pressure and flow rate. Nomenclature not defined in the list can occur. Most of the information in this appendix is written by Per Fahlén according to his previous works.

11.9 Temperature measurement in piped systems for heating and cooling agents

Accurate measurement requires invasive installation by means of directly inserted sensors or use of temperature wells located in suitable positions and use of accurate and well-calibrated sensors. Requirements on the measuring accuracy can be found for instance in the standards SS2620[29] and NTVVS076[25] and advice regarding installation in specialist literature (e.g. Adunka[10], Fahlén[15,18,19]) or standards (e.g. EN306[29], VDI/VDE3511[31,32,33]). The following information provides a brief overview of the measuring problem.

11.9.1 What to measure?

Cooling or heating capacity can be determined by measurement of flow rate and temperature difference between the inlet and outlet of the system. The temperature difference must reflect how the heat content of the fluid is changed by the system and hence it is paramount to find temperatures that are representative of this change. The mean temperature T_B that represents the heat content of the fluid is known as the bulk temperature and is defined as

$$T_B = \frac{1}{q_v} \langle T(r,\varphi) \cdot v_z(r,\varphi) \cdot r \cdot dr \cdot d\varphi \rangle \qquad \text{[W]} \qquad \text{(eq. 0.1)}$$

where <> denotes the spatial mean value over the cross-section of the pipe and q_v is the volumetric flow rate. The definition indicates that it is necessary to consider the product of the local temperature $T(r, \varphi)$ and the local velocity $v_z(r, \varphi)$ as well as the position in the pipe cross-section (r, φ) at a given position along the pipe (z). (r = position in a radial direction, φ = angular position, z = position along the pipe axis.) Thus, if the temperature varies over the cross-section it will not be sufficient to insert extra temperature sensors and calculate their mean value. It is also necessary to know the fluid velocity at each measuring position.

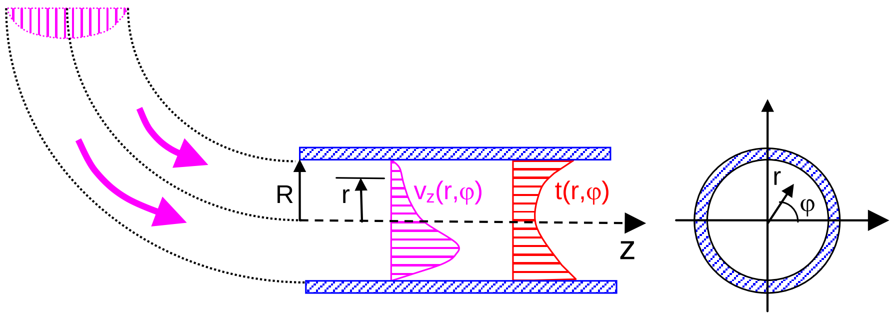

Figure 0.1 In the definitionen of bulk temperature it is necessary to consider both the local temperature T(r, φ, z) and the local fluid velocity in the axial pipe direction v_z (r, φ, z).

Figure 0.1 illustrates that velocity profile and temperature profile can be totally different at a specific cross-section of the pipe. How they will appear depends on the types of flow disturbances and how heat is supplied and removed from the system.

11.9.2 Where to measure?

As it usually is not possible to determine both velocit and temperature in a number of positions the practical solution is to find measuring positions where the fluid is totally mixed and has a homogeneous temperature. Such positions may be found after flow disturbing components such as seveveral successive bends (see **Figure 0.2**) or after low generators (pumps or fans). When measuring after flow generators, it will be necessary to consider the heat input to the fluid from the pump work.

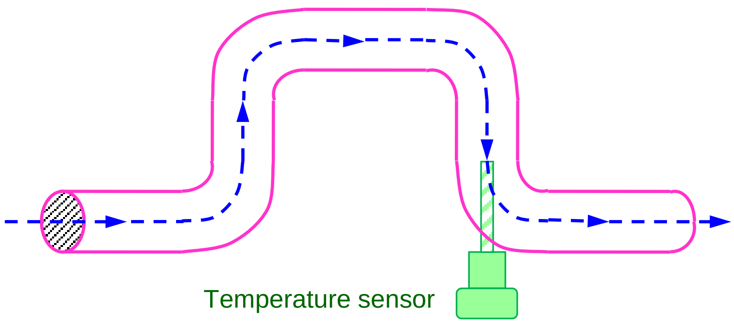

Figure 0.2 Example of a suitable measuring position to determine bulk temperature. Successive double bends will ascertain good mixing and high turbulence (good heat transfer).

It is, of course, paramount that the measuring positions coincide with the specified system boundary for the test. Contractual situations, in particular, require careful specification of whether the prescribed performance applies to the heat pump, the heat pump system (including the heat pump flow generators) or the heating or cooling system (including heat distribution and supplementary heating or cooling). Figure 0.3 provides examples of alternative system boundaries and the risk of gross errors in case the temperature sensors are not installed on the same side of a by-pass or mixing valve as the flow meter (they must be mounted so that the same flow passes both types of sensor). NT VVS standard [25,27,28] provide useful information on system boundaries, measuring requirements etc.

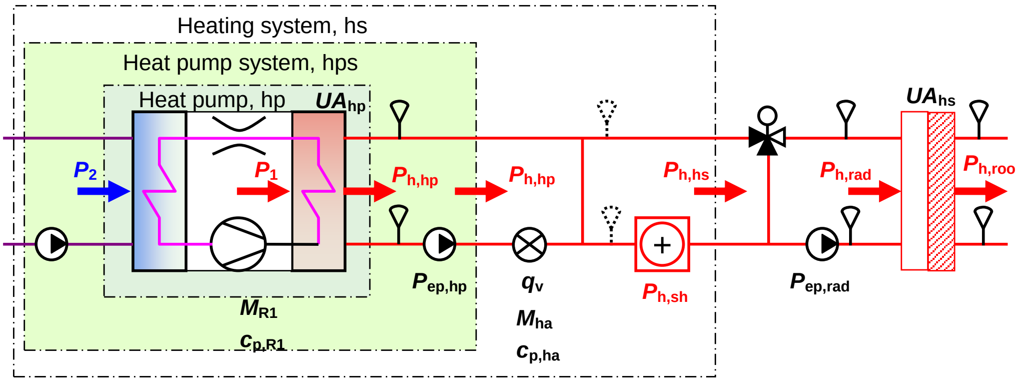

Figure 0.3 Example of measuring positions (P_h = heating capacity, P_c = cooling capacity, hp = heat pump, hps = heat pump system, hs = heating system, ha = heating agent, ep = electric power to pumps).

11.9.3 Where not to measure?

Directly after heaters of coolers, close to mixing valves and in uninsulated pipes temperature stratification may occur and such places should be avoided. Particularly in laminar flow, not uncommon in cooling agent systems, temperature stratification may remain over rather long distances.

Figure **0.4** shows an example where the measured temperature may vary between 20 °C and 80 °C depending on how far the sensor is inserted. Irrespective of where one measures, pipes **must** be sufficiently insulated. This applies not only to the actual measuring position but also to a sizeable distance upstream and downstream of the sensor (> 10 pipe diameters in both directions; c.f. the comments regarding heat exchange in thermometer wells)

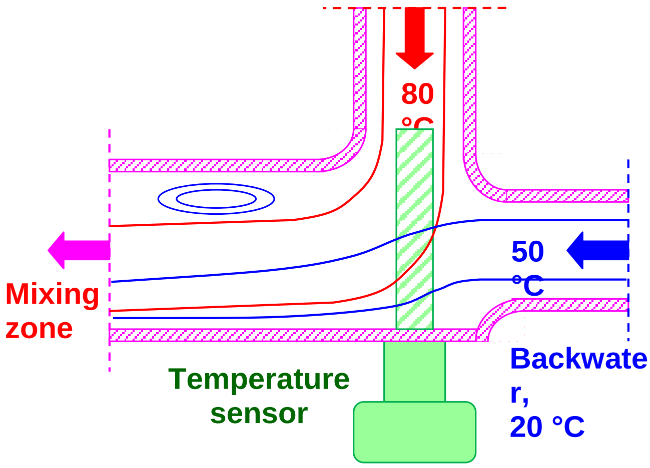

Figure 0.4 Possible temperature variation at a mixing position (Fahlén [20]).

11.9.4 How to measure - Invasive measurement?

If possible, invasive measuring techniques should be used to ascertain a close agreement between senor temperature and the desired fluid temperature. In this report, measurements will be designated as invasive when sensors are inserted in direct contact with the fluid even if this is achieved by means of a thermometer well. Wells are practical to use since they facilitate onsite calibration, functional testing and exchange of sensor. When practically possible, mounting of the sensor in a pipe bend according to Figure 0.5 and Figure 0.2 is a good alternative. By mounting the thermometer well counter-current to the flow the tip of the well measures an undisturbed fluid temperature, i.e. not affected by the sensor, and heat transfer will be high as the tip becomes a stagnation point of the flow.

Figure 0.5 illustrates the basic heat exchange between thermometer well, fluid and pipe walls as well as between the connecting box and its ambience. Thermal resistances determine the heat flows generated by the fluid-to-ambience temperature difference (R = thermal resistance; subscript λ = conduction, α = convection, r = radiation). The figure shows that if the temperature T_1 of the well tip is to come close to that of the fluid, T, then $R_{\alpha 1}$ must be small and all R_λ must be large. That is why a large insertion depth is beneficial for accuracy as a long well will yield a large value of $R_{\lambda 2}$. This also implies that it is advantageous to use a well material of low thermal conductance, e.g. stainless steel. This is contrary to one's immediate notion that high conductance is good for letting the heat reach the sensor but it also makes it easy for heat to be lost before it reaches the sensor. For special applications there are wells on the market that use a high-conductivity material at the tip and a low-conductivity material for the rest of the well (usually not necessary).

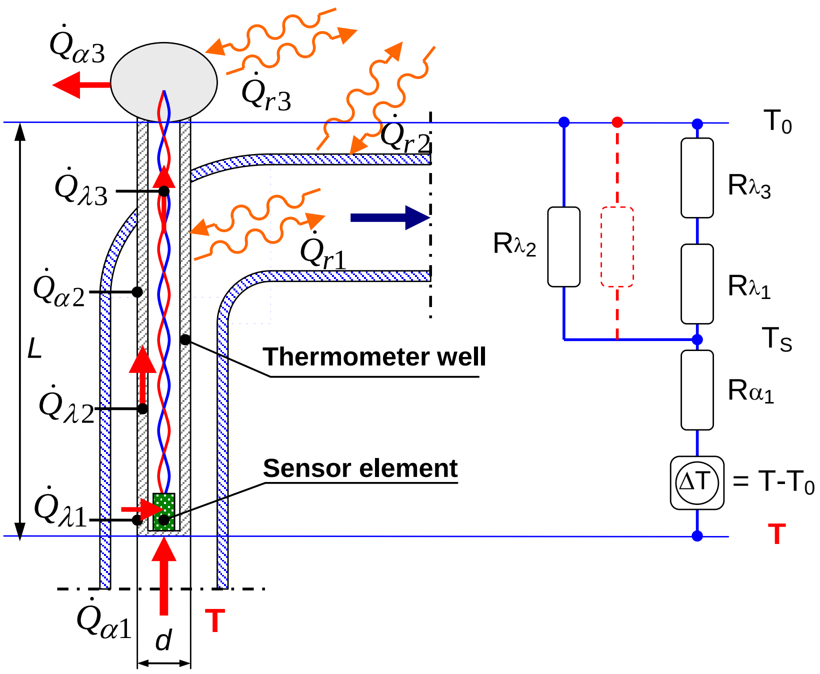

Figure 0.5 Installation of a thermometer well in a 90 ° bend (Fahlén [19]) Heat exchange occurs between the well, the fluid and the surrounding pipe walls as well as between the connection box and the surroundings.

The measuring error caused by heat transport in the well is directly proportional to the temperature difference between the fluid (T) and the temperature of the well-fitting in the pipe (T_0). Thus the importance of insulating the pipe becomes obvious as the well-related error approaches zero when the pipe temperature becomes the same as that of the fluid. By means of a well-insulated pipe, a well-insulated sensor installation and good mixing of the fluid it is possible to measure accurately even with relatively short thermometer wells.

The heat exchange causes an overestimated temperature in parts of the system with lower temperature than that of the surroundings and an underestimated temperature in parts with higher temperature. Note that heat transfer between the well and sensor is very important and must be ascertained by use of conductive paste or similar products. The paste should only be applied to the extent of the sensor and not fill the well. If the well is filled, then its thermal resistance $R_{\lambda 2}$ will be short-circuited and the point of using a long well is forfeited.

11.9.5 How to measure – Non-invasive measurement?

Non-invasive measurements are usually carried out by means of surface mounted sensors on the outside of pipes or ducts. This is the prevalent situation for measurement of temperature in the refrigerant system as required in the internal method to determine *COP*. Information on temperature in various parts of the refrigerant system is also important to assess the status of the system (e.g. compressor outlet temperature, sub-cooling, superheat) and adjustment of throttling devices such as thermostatic expansion valves etc. Unfortunately, installations are rarely equipped with temperature or pressure taps with correct design and in the right position. This section will present some aspects of surface temperature measurement.

11.9.5.1 Measurement of surface temperature

When temperature wells are missing, e.g. the most common situation, the usual procedure is to determine temperature indirectly (non-invasively) by measuring the surface temperature of the refrigerant pipe. This temperature will always be somewhere between the temperature of the fluid and the temperature of the ambience. This also means that too high a temperature will be measured in the cold parts of the system and too low a temperature in the warm parts of the system. Practical investigations[15] on uninsulated pipes show that the measured temperature, at a difference of 30 K between fluid and surroundings, may differ 4-5 K from the correct value even with careful execution of the measurement in a laboratory situation (use of thermal paste, good clamping, insulation of the actual measuring installation etc.).

11.9.5.2 How to measure?

Since the measured value represents the pipe temperature while the desired value is that of the fluid, the first action is to ascertain that these two temperatures differ as little as possible. This may be achieved by good heat transfer on the inside of the pipe and poor heat transfer on the outside. As it usually is not possible to affect the inside of the pipe, it remains to lower the outside heat transfer by means of good insulation. Figure 0.6 illustrates the difference in temperature profile between uninsulated and insulated pipes when the refrigerant has a lower temperature than the ambience.

Note that metal pipes, copper pipes in particular, are good conductors of heat also in the axial direction. Therefore the pipe must be insulated over a sufficient length in order to avoid that axial heat transport spoils the effect of insulation at the measuring position. A rule-of-thumb is to insulate at least 10 cm on both sides of the measuring position. More insulation over a longer pipe section will, of course, be required when the temperature difference between fluid and ambience is increased.

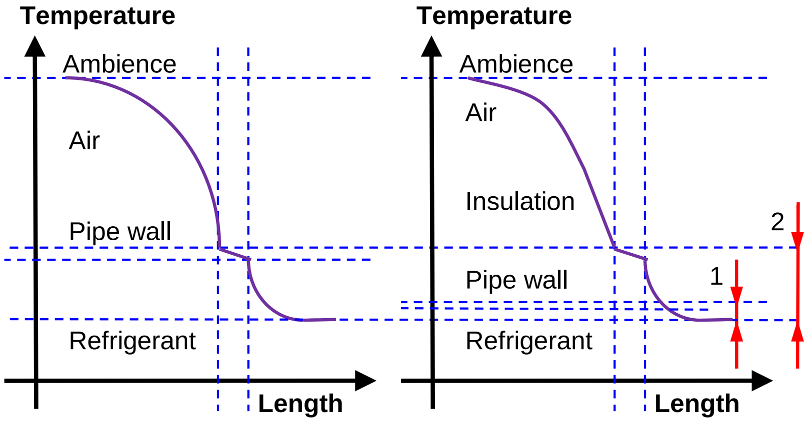

Figure 0.6 Temperature profile of an uninsulated and an insulated pipe respectively. 1 = deviation of the surface temperature with insulated pipe; 2 = deviation of the surface temperature with uninsulated pipe.

Surface temperature may be measured by means of many alternative types of instrument, e.g. contact thermometers, thermocouples, resistive temperature sensors designed for surface mounting (see
Figure **0.7**). Irrespective of sensor type, the sensor and instrument must be calibrated. When using thermocouples, note that the main part of the sensor output is generated where the largest temperature difference is, i.e. the part that protrudes through the insulation. Hence this part of the thermocouple should also be included in the calibration bath/oven. Bernhard et al[34], among others, describe alternative methods of calibrating contact thermometers.

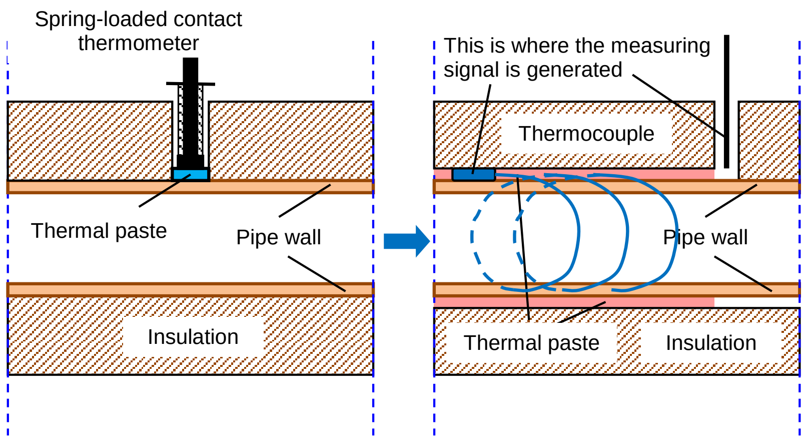

Figure 0.7 Alternative ways of measuring surface temperature. When using thermocouple wires thermal paste and at least 1 dm of thermocouple should be in contact with the pipe.

I special cases various types of IR-instrument may be used. One important advantage of this type of sensor is that it is possible to relatively quickly establish how the temperature varies over large surfaces. This may for instance be used to find problems in equipment such as overheated motors, faulty insulation of hot and cold parts etc. Another advantage is that the surface temperature to be measured will not be disturbed in the way it is when using surface mounted sensors. A disadvantage is that if one strives to make accurate quantitative measurements, it is necessary to know the radiant properties of the surface (emissivity, absorbance, reflectance and transmittance) as well as the temperature of the background radiation. The lower the emissivity and the greater the reflectance of a surface, the greater will be the measuring problems.

11.9.6 Consequences of erroneous temperature information

The experienced refrigeration engineer will immediately recognize the consequences of flawed measurements. If, for instance, one believes that the superheat is 7 K but the real value only is 2 K there will be a risk for the compressor of operating with wet vapour (2 K superheat may result in 20 % liquid in the outlet vapour in the form of little droplets). Also, there is a great risk of instable operation through insufficient stability margin for the combination evaporator/expansion valve.

In the measurement of compressor outlet temperature there is a risk of underestimating the real value. Without very careful execution of the measurement, 5-10 K is not unlikely. Furthermore, the gas temperature in the compressor is higher than it is in the outlet pipe. Assessing the risk of future compressor failure, this must be born in mind.

To the extent that temperature and pressure measurements are used to analyse the performance of a heat pump, above all the error in the temperature of the compressor outlet will affect the result. Whether a pressure/enthalpy chart or a heat pump analyzer is used for the analysis, the underestimated outlet temperature will result in an overestimated COP (up to 2 % per K).

11.10 Pressure measurement

Many types of field measurement require measurement of pressure. Examples of such cases are:

- Flow measurement based on pressure difference transducers (flanges, orifices, venturi tubes, bends etc.).
- Determination of pressure drop or available external pressure difference of a unit.
- Pressure measurement during checking of pressure vessels or safety valves.
- Pressure measurement to determine evaporating or condensing pressure temperature) as well as superheat and sub-cooling.
- Pressure measurement in connection with assessment of COP by means of the internal method.

Just as the case was for temperature accurate measurement of pressure requires measuring taps located in suitable positions in the system and accurate, well-calibrated sensors. Unfortunately, heat pumps are rarely delivered with properly designed pressure taps in the right positions. Requirements on pressure measurement in heat pumps may be found in for instance SS2620[29] and NTVVS076[25] and advice on design in specialist literature (e.g. Fahlén [17,19]) or standards (e.g. SS-EN306[30], ANSI/ASHRAE [11,12,13,23]or ISO). The following paragraphs contain some views on the measurement of pressure.

11.10.1 What to measure?

Pressure is defined as force per unit area and is expressed in the unit Pa. It may be of interest to note that pressure is also a measure of the energy content per unit volume of a fluid (Pa = N/m^2 = J/m^3) while temperature is related to the energy content per molecule. The consequence of this is readily observed in the fact that high pressure refrigerants provide compact systems since their volumetric energy content is high. This also implies the risk involved in a ruptured refrigerant system may not necessarily increase with a high pressure refrigerant as the available pressure energy (pressure x volume), for a given thermal capacity, will not be very different.

Depending on the absolute pressure level and the flow regime of the fluid many different types of pressure may be distinguished. It is therefore necessary to first clarify which type of pressure one needs to know when planning a measurement. According to Figure 0.8 all types of pressure measurement are some form of **difference measurement**. Depending on the absolute level of the reference pressure the terms **absolute pressure** *p*(a), **gauge pressure** *p*(e) or **differential pressure** are used for comparison with vacuum, atmospheric pressure or an arbitrary pressure respectively.

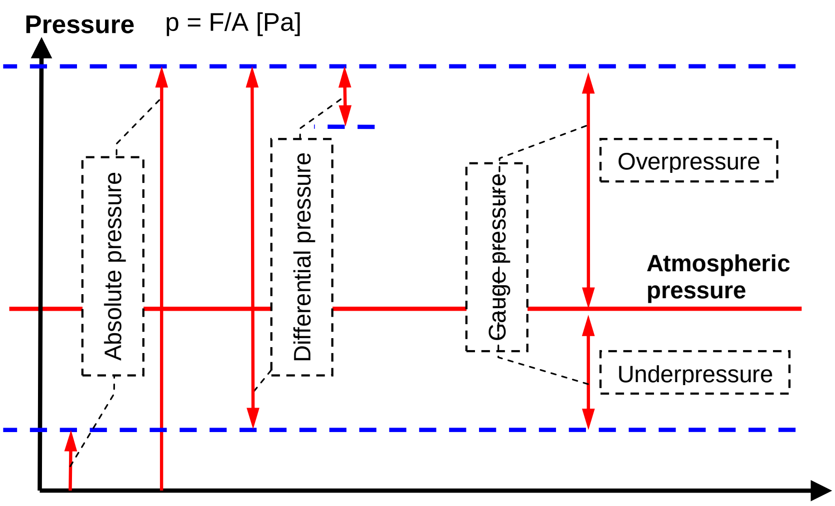

Figure 0.8 Alternative designations of pressure.

In connection with measurement of pipe flow, an additional component of the total pressure related to the kinetic energy of the fluid has to be considered. This component is directionally dependent and has its maximum in the flow direction. To distinguish the various pressure components during measurement in flowing fluids the terms **static** pressure, **dynamic** pressure and total pressure or stagnation pressure are used.

11.10.1.1 Static pressure

If a fluid is at rest, the pressure at a given point will be equal in all directions and may be expressed as the sum of three components:

$$p = p_1 + p_2 + p_3 \qquad \text{[Pa]} \qquad \text{(eq. 0.2)}$$

The pressure p_1 is caused by the kinetic energy of the molecules, p_2 is due to an external pressure on the fluid boundary (e.g. via a piston) and p_3 comes from a force field (e.g. gravitational or electro-magnetic) acting on the entire fluid volume. If the fluid is a **gas**, p_2 is included in p_1 due to the compressibility of the gas, i.e.

$$p = p_1 + p_3 = p_1 + h \cdot \rho \cdot g \approx p_1 \qquad \text{[Pa]} \qquad \text{(eq. 0.3)}$$

where h is the height of the fluid column, ρ is the fluid density and g is the gravitation of the earth. For gases, p_3 may in most cases be neglected due to the low density of gases.

When the fluid is a **liquid**, p_1 may be neglected. For liquids, however, p_2 may not be neglected since liquids cannot be considered as compressible. Often it is not even possible to neglect p_3 due to the relatively high density of liquids;

$$p = p_2 + p_3 = p_2 + h \cdot \rho \cdot g \qquad \text{[Pa]} \qquad \text{(eq. 0.4)}$$

Finally, if the fluid is a **vapour**, i.e. a gas in equilibrium with its liquid phase, the static pressure is directly related to temperature via the saturation curve of the fluid (in mixtures the relation will vary with local composition of the vapour and liquid respectively).

11.10.1.2 Dynamic pressure and total pressure

The dynamic pressure derives from the kinetic energy of the fluid flow and is given by

$$p_d = \frac{1}{2} \cdot \rho \cdot v^2 \qquad \text{[Pa]} \qquad \text{(eq. 0.5)}$$

where v is the flow velocity at a given point.

Total pressure is a measure of the pressure that is obtained at point where the fluid has been retarded to the velocity $v = 0$. In the simple case when the fluid is assumed to be incompressible and the fluid flow is frictionless the total pressure can be written as

$$p_{tot} = p_s + p_d = p_s + \frac{1}{2} \cdot \rho \cdot v^2 \qquad \text{[Pa]} \qquad \text{(eq. 0.6)}$$

This simple case is known as the Bernoulli relation and it tells us that the total pressure will be constant along a streamline. According to the relation above this means that when flow velocity changes the static pressure will also change. The Bernoulli relation applies reasonably well to normal liquids and for gases as long as the flow velocity is much lower than the sound velocity. In situations with compressible flow the total pressure will always be larger than the sum according to Bernoulli.

In many cases the static pressure will be the measurand of interest. This is the pressure that decides the density of compressible fluids when one aims to calculate the mass flow rate, it is used in the saturation pressure/temperature relation and it is decisive for requirements on pipe strength. In other cases, e.g. using Prandtl tubes for velocity determination, the dynamic and/or total pressure may also be used.

A pressure sensor provides an output signal which is a function of the total force acting on its sensing element. This force is given by the area of the sensing element multiplied by the total pressure acting on this area. As total pressure will vary with position in the fluid and orientation in relation to the local fluid velocity it is important to consider this in the design of pressure taps. Otherwise the result of the pressure measurement may be erroneous in spite of using accurate and calibrated sensors.

11.10.2 Where to measure?

The discussion in the previous section indicates that the magnitude and direction of any dynamic pressure component must be considered. In some cases differences in altitude must also be considered (mainly in liquid measurements). This leads to the following recommendations:

- when $p_d << p_s$ the measuring plane can be chosen at will (however, note the possible influence of $h \cdot \rho \cdot g$)

- when p_d cannot be neglected the measuring plane should be at least $1 \cdot D_h$ upstream of any flow disturbance and at least $5 \cdot D_h$ after a disturbance.

In the finding of suitable measuring positions for pressure much the same rules apply as for installation of flow meters, i.e. be careful in the vicinity of bends, area changes, confluences and other changes in the pipe system that may affect the flow situation. In normal refrigeration systems the dynamic pressure is usually less than a few per cent of the static pressure and should therefore be unproblematic. If one is uncertain, it is quite simple to estimate the situation in a specific case. When measuring in distribution system with heating or cooling agents, however, it is usually not possible to neglect the dynamic pressure.

11.10.3 How to measure?

When holes are drilled for pressure taps it is important to shape the holes in a way that they will not disturb the flow along the pipe wall, thereby inducing a dynamic pressure component. ISO Standards Handbook 15 [24]contains the following recommendations in Standard 5167,

- Holes must be round, orthogonal to the pipe axis, with sharp edges and without burrs on the inside of the pipe.

- Holes must have a diameter of less than 0.08*D*, preferably not exceeding 12 mm. Minimum diameter is limited by practical considerations such as the risk of blockage by dirt, damping of pressure variations etc.

- Levelling of any remaining variations is preferably made by joining several taps by means of a ring pipe or an equalizing chamber (see the example according to ISO/DIS/4064/3 in Figure 0.9).

Depending on the type of fluid there may by further practical considerations on the design of pressure taps. ISO Standards Handbook 15, Standard 2186, provides a detailed description which is illustrated by several figures that present recommend solutions.

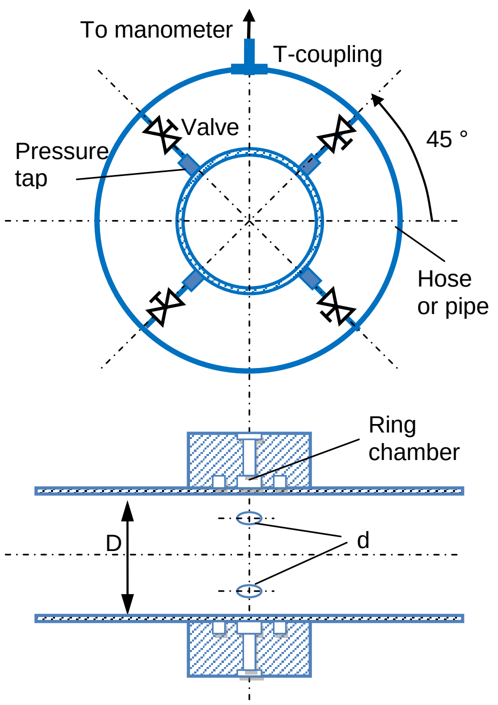

Figure 0.9 Example of pressure-tap design (Fahlén [22], based on ISO Standards Handbook 15[24] 1983).

In the selection of a measuring plane it is often practical to locate this to a vertical pipe as the circumferential position of the taps in this case will not be very critical (no difference in gravitational force). If, however, the measuring plane is located in a horizontal pipe, the following points should be observed,

- clean liquids: position the tap within 0-45° below the horizontal plane,
- dirty liquids: position the tap within 0-45° above the horizontal plane,
- gas: position the tap on the top of the pipe in the vertical plane,
- vapour: position the tap in the horizontal plane.

In order to facilitate the dismounting of pressure sensors, there should always be stop cocks between the pressure tap and the sensor. It is also advantageous to have control valves to moderate pressure variations and valves for deaeration of liquid systems. In the case of differential pressure sensors, there should always be an equalizing valve between the high and low pressure sides to avoid loading the sensor with the line pressure as differential pressure when connecting or disconnecting the sensor.

In liquid systems, the connecting pipes should be monotonously inclined downwards from the tap to the sensor to ascertain that the measuring system is at all times filled with liquid. If the tap is above the horizontal plane to avoid the risk of clogging, e.g. due to sedimentation, the transfer pipes should be monotonously inclined upwards to a gas separator and then go monotonously downwards to the sensor. Summarizing, the following rules of thumb should be followed to avoid static errors during measurement of pressure:

- use calibrated instruments and sensors,
- select an appropriate measuring plane with correctly designed pressure taps,
- adapt the sensor position according to the measuring situation,
- use valves for shut off, pressure equalization and deaeration (in liquid systems),
- do not use too long transmission pipes for the pressure,
- transmission pipes must be of sufficient diameter, equally long and kept together,
- transmission pipes must be monontonously and sufficiently inclined.

11.10.4 Consequences of erroneous pressure measurement

Errors in the determination of pressure will directly influence the errors of derived values of e.g. flow rate, evaporating or condensing temperature, superheat etc. In the measurement of fluid flow by means of orifice plates or other differential pressure devices the flow rate error will approximately half the error of the pressur measurement. The internal method to determine coefficient of performance requires pressure measurements to determine the change in specific enthalpy over the compressor, condenser and sub-cooler. The requirements on accuracy, however, are rather modest in this application[47].

11.11 Flow rate measurement

To measure accurately the thermal capacity by means of the external method, accurate measurement of flow rate is required not only to determine capacity per se but also to assess the operating condition of the heat pump. The flow rate measurement is usually the most difficult and the most expensive measurement conducted on heat pumps (one of the primary aims of developing various types of internal evaluation methods have been to avoid the flow measurement). The external method requires accurate and well-calibrated flow meters which are suitably positioned in the system. Requirements on accuracy may be found e.g. in SS2620 or NTVVS076 and advice on the design may be found in specialist literature (e.g. Fahlén [18,19]) or standards (e.g. EN306 [30]). Unfortunately installations are rarely prepared for measurement and usually lack suitable measuring sections in the right place and of the correct design. Further details may be found in the references.

If possible, one should use full flow measurement, i.e. the flow meter should provide an output that is a measure of the integrated mean value of the flow velocity over the pipe cross-section. Most methods for flow measurement require some kind of interfering with the pipe system (invasive methods) but there are some methods that avoid this (non-invasive methods). Examples of various categories of flow meter are:

- **Full flow - Invasive**
  - Displacement meters
  - Velocity meters
  - Mass-flow meters
- **Full flow - Non-invasive**
  - Tracers
  - Temperature pulse
  - Supplementary heat

It is often expensive and difficult to mount full-flow meters, e.g. in large air ducts or in liquid systems (that must be drained, refilled and deareated). In such cases various types of velocity/area methods may be of interest. Some alternatives are given below:

- **Velocity/area - Invasive**
  - Prandtl or pitot tubes
  - Anemometers
  - Turbine or inductive probes
- **Velocity/area -Non-invasive**
  - Ultrasonic meters

Ultrasonic meters of the clamp-on type may appear as full-flow meters but they normally measure velocity in specific rays and full flow is calculated based on an assumption regarding the velocity distribution over the pipe cross-section.

Errors in flow determination directly influence the error in determination of the thermal capacity. If the flow error is 5 % then the error of the calculated heating or cooling capacity will also be 5 %. Furthermore, the flow measuring error will also result in an erroneous perception of the operating point and this is important in contractual testing. Heat transfer, sub-cooling and pressure drop will be affected by flow rate. Hence, capacity as well as coefficient of performance will be affected and the measured value will be compared with an incorrect guaranteed reference value.

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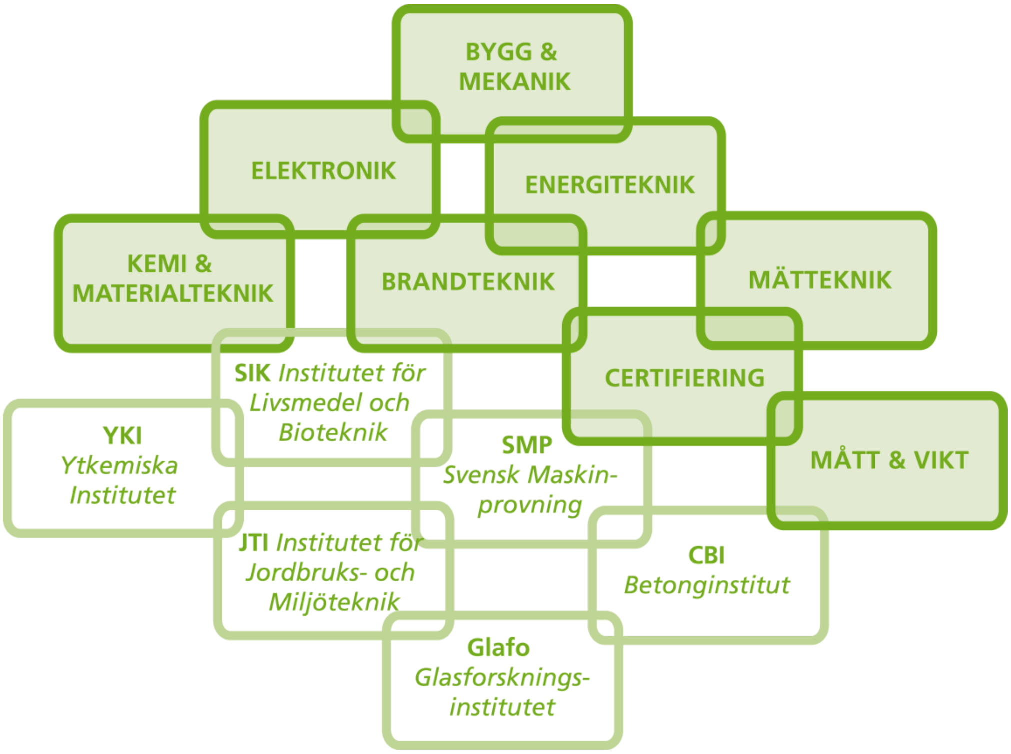

SP Sveriges Tekniska Forskningsinstitut

Box 857, 501 15 BORÅS

Telefon: 010-516 50 00, Telefax: 033-13 55 02

E-post: info@sp.se, Internet: www.sp.se

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