VYSOKÉ UČENÍ TECHNICKÉ V BRNĚ
BRNO UNIVERSITY OF TECHNOLOGY

FAKULTA STROJNÍHO INŽENÝRSTVÍ
ENERGETICKÝ ÚSTAV

FACULTY OF MECHANICAL ENGINEERING
ENERGY INSTITUTE

HEAT PUMP PERFORMANCE DATA ANALYSIS

ANALÝZA PROVOZNÍCH DAT TEPELNÉHO ČERPADLA

BAKALÁŘSKÁ PRÁCE
BACHELOR'S THESIS

AUTOR PRÁCE LIBOR ŠEDA
AUTHOR

VEDOUCÍ PRÁCE Ing. JIŘÍ HEJČÍK, Ph.D.
SUPERVISOR

BRNO 2013

Vysoké učení technické v Brně, Fakulta strojního inženýrství

Energetický ústav
Akademický rok: 2012/13

ZADÁNÍ BAKALÁŘSKÉ PRÁCE

student(ka): Libor Šeda

který/která studuje v **bakalářském studijním programu**

obor: **Strojní inženýrství (2301R016)**

Ředitel ústavu Vám v souladu se zákonem č.111/1998 o vysokých školách a se Studijním a zkušebním řádem VUT v Brně určuje následující téma bakalářské práce:

Analýza provozních dat tepelného čerpadla

v anglickém jazyce:

Heat pump performance data analysis

Stručná charakteristika problematiky úkolu:

Účinnost tepelného čerpadla je posuzována pomocí topného faktoru (COP), jehož velikost je závislá na provozních podmínkách. Z tohoto důvodu se v posledních letech přechází ke stanovení tzv. sezónního topného faktoru (SCOP), který představuje „účinnost" topného čerpadla v průběhu celého roku.

Cíle bakalářské práce:

Z naměřených dat provést vyhodnocení výkonových parametrů tepelného čerpadla.

Seznam odborné literatury:

ŠPANIHELOVÁ, K. Metodika zkoušení tepelných čerpadel. Brno: Vysoké učení technické v Brně, Fakulta strojního inženýrství, 2012. 38 s. Vedoucí bakalářské práce Ing. Jiří Hejčík, Ph.D.

Vedoucí bakalářské práce: Ing. Jiří Hejčík, Ph.D.

Termín odevzdání bakalářské práce je stanoven časovým plánem akademického roku 2012/13.

V Brně, dne 20.11.2012

L.S.

doc. Ing. Zdeněk Skála, CSc.
Ředitel ústavu

prof. RNDr. Miroslav Doupovec, CSc., dr. h. c.
Děkan

ABSTRACT

This thesis deals with heat pumps testing methods and coefficients that are commonly used for ranking heat pumps in terms of performance and ecological friendliness. For testing according to ISO standards the inputs for equations are explained. In practical part, data from a running heat pump are evaluated to rank performance and parameters of the compressor, e.g. number of working hours or number of defrost.

KEYWORDS

heat pump, coefficient of performance, seasonal coefficient of performance, low potential heat sources, data evaluation

ABSTRAKT

V bakalářské práci jsou shrnuty dostupné metody a koeficienty užívané k testování výkonu a ekologické náročnosti tepelných čerpadel. Pro testování dle norem ISO jsou zde také vysvětleny vstupující faktory. V praktické části je vyhodnocen běh reálného tepelného čerpadla z hlediska výkonnostních parametrů a parametrů týkajících se běhu kompresoru, jako je počet pracovních hodin nebo počet odmrazení.

KLÍČOVÁ SLOVA

tepelné čerpadlo, topný faktor, sézonní topný factor, nízkopotenciální zdroje tepla, vyhodnocení dat

ŠEDA, L. *Analýza provozních dat tepelného čerpadla.* Bachelor’s thesis. Brno: Brno University of Technology, Faculty of Mechanical Engineering, 2013. 67 p. Supervised by Ing. Jiří Hejčík, Ph.D..

DECLARATION

I declare, that I have elaborated my bachelor's thesis on theme of „Heat Pump Performance Data Analysis" independently, under the supervision of the bachelor's thesis supervisor and with use of the technical literature and other sources of information which are all quoted in the thesis and detailed in the list of literature at the end of the thesis.

Brno …….……………. ………………..……………….

(Author's signature)

ACKNOWLEDGEMENT

The author would like to thank his supervisor, Ing. Jiří Hejčík, PhD. for his kind help and comments that helped him to write this thesis.

Further thanks go to Ing. Peter Hipik and Ing. Tomáš Konečný from company Emerson Inc. for providing the evaluated data and the possibility to visit the manufacturing plant in Mikulov.

Contents

1 Introduction

With the price of electrical energy getting higher every year, the market with renewable energy devices is getting bigger and bigger. One of the examples is the one with heat pumps. Since 90s, when heat pumps got into general awareness, the demand was growing, and still is.

With this demand on a market, the importance of heat pump performance evaluation was growing. In the end, the customers are partly purchasing such devices as an investment and the return on investment is then questioned, as needed for calculation of benefits.

Till recently, the only factor used for such comparison is Coefficient of Performance, which testing procedure can be found in European standard EN 14511. In last 10 years, seasonality is one of the most important factors, and Seasonal Coefficient of Performance (described in European standard EN 14825) has much greater importance, as it is giving us better overview of how the device is working during the whole year.
For both coefficients, there is also a methodology of testing included in the standard and there are certified laboratories that can perform the testing. The manufacturers are also testing their devices themselves, and claiming values of such coefficients. On the other hand, the comparisons made by unbiased laboratories showed great differences between proclaimed values and reality.

The aim of author’s work is to make a research on possible raking methods used for testing heat pumps on performance and ecological running, mainly in European Union and then evaluate performance of a running heat pump.

2 Principle

Heat pump is a machine that allows us to take away from the environment heat on a low-energy level and draw it to a high-energy level. This heat can afterwards be used for heating and domestic hot water (DHW). For the drawing, there must be a certain amount of work added.

The theoretical basis of this system lies in the middle of 19th century, when thermodynamics as a branch of natural science was born. During this time, 4 laws of thermodynamics were set, that are valid for all systems that fall within the constraints implied by each. In the various theoretical descriptions of thermodynamics these laws may be expressed in seemingly differing forms. [1]

First usage of a heat pump is dated back to 1856, when Czech engineer, Peter von Rittiger, recognized the principle of the heat pump while conducting experiments on the use of water vapour's latent heat for the evaporation of salt brine. As a result, in Austria the heat pump was used to dry salt in salt marshes. [7]

During the years, the main parts of a heat pump remain unchanged. It basically consists of two parts – low-pressure part and high-pressure part. While in heating mode, in low pressure part we can find evaporator and in high-pressure part a condenser. These parts are separated by compressor on one side and expansion valve on the other side. In cooling mode, the order is vice versa. [22]

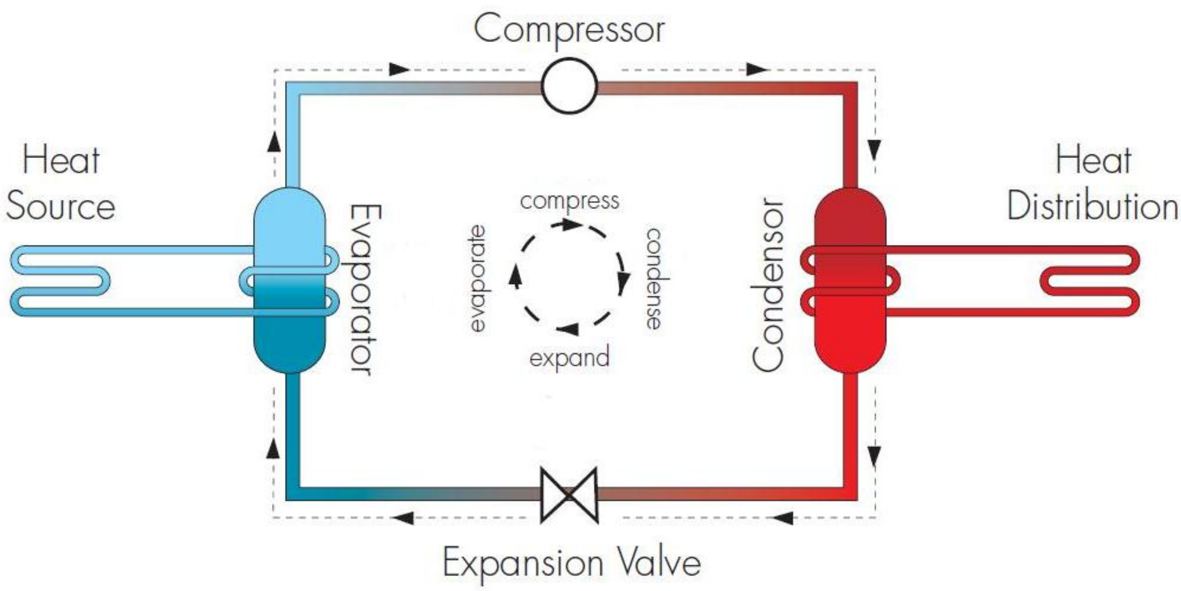


Fig. 2.1.: Heat pump cycle. [8]

Theoretical thermodynamical cycles that are most often used to describe a run of a heat pump are Carnot cycle and Rankin-Clasius cycle.

2.1 Carnot cycle

Carnot cycle is reversible, ideal and the most efficient heat engine cycle allowed by physical laws. It sets the limiting value on the fraction of the heat that can be used for an engine operating between two temperatures. It is not a practical engine cycle because the heat transfer into the engine in the isothermal process is too slow to be of practical value.

In Carnot efficiency, only temperatures of cold and hot reservoir are taken into consideration.

$$\eta = 1 - \frac{T_c}{T_h} \tag{2.1}$$

Theoretical totally reversible Carnot cycle is the one that is most commonly used to describe what is happening inside a heat pump. In pressure-volume diagram it consists of two isotherms and two adiabats. [2]

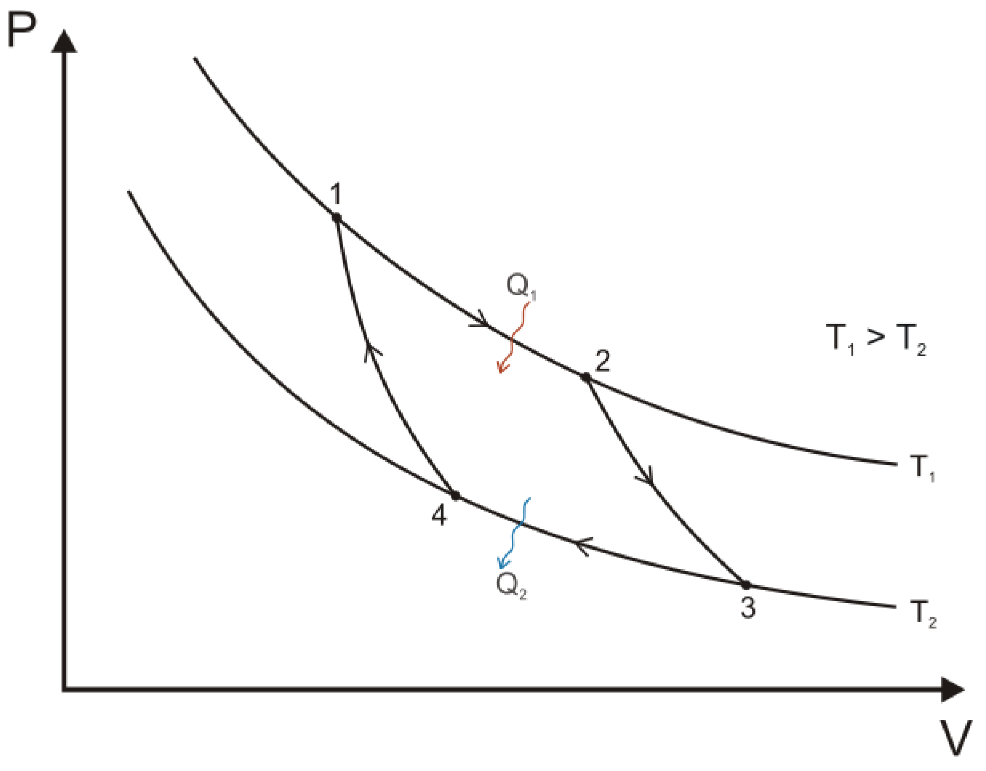


Fig 2.2.: Carnot cycle. [2]

Reversible isothermal expansion of the gas at the "hot" temperature, T_H (isothermal heat addition or absorption). During this step (1-2) the gas is allowed to expand and it does work on the surroundings. The temperature of the gas does not change during the process, and thus the expansion is isothermal. The gas expansion is propelled by absorption of heat energy Q_1 and of entropy from the high temperature reservoir.

$$\Delta S = \frac{Q_1}{T_H} \tag{2.2}$$

Isentropic (reversible adiabatic) expansion of the gas (isentropic work output). For this step (2-3) the piston and cylinder are assumed to be thermally insulated, thus they neither gain nor lose heat. The gas continues to expand, doing work on the surroundings, and losing an equivalent amount of internal energy. The gas expansion causes it to cool to the "cold" temperature, T_C. The entropy remains unchanged.

Reversible isothermal compression of the gas at the "cold" temperature, T_C. (Isothermal heat rejection) (3-4) now the surroundings do work on the gas, causing an amount of heat energy Q_2 and of entropy to flow out of the gas to the low temperature reservoir. (This is the same amount of entropy absorbed in step 1, as can be seen from the Clausius inequality.)

$$\Delta S = \frac{Q_2}{T_C} \tag{2.3}$$

Isentropic compression of the gas (isentropic work input). (4-1) Once again the piston and cylinder are assumed to be thermally insulated. During this step, the surroundings do work on the gas, increasing its internal energy and compressing it, causing the temperature to rise to T_H. The entropy remains unchanged. At this point the gas is in the same state as at the start of step 1. [12]

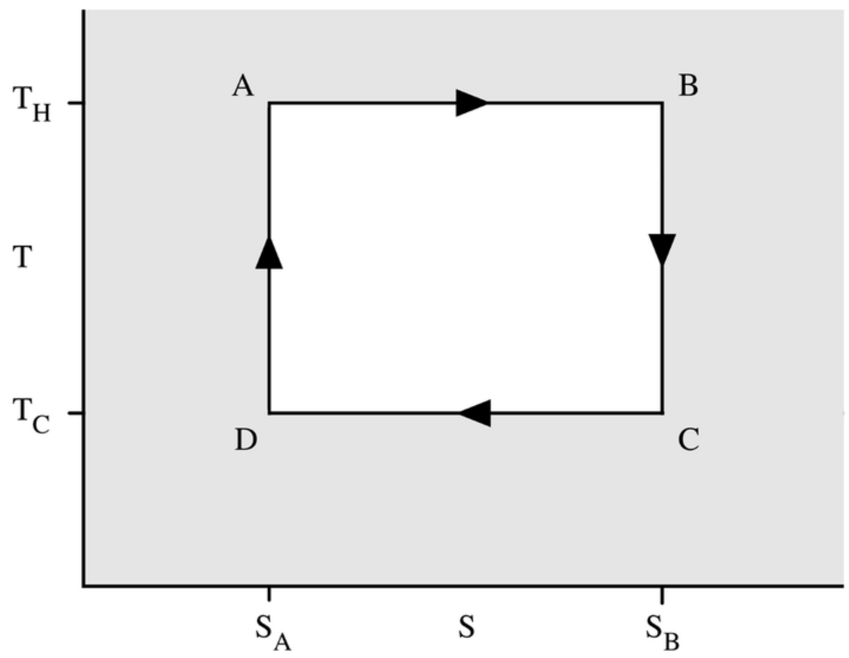


Fig. 2.3.: T-S diagram of Carnot cycle

2.2 Rankine cycle

Idealised Rankine anticlockwise cycle can approximately describe heat pump cycle with real refrigerant. Rankine cycle count only with isolated heat pump, lossless compression, no subcooling or overheating of refrigerant and also no pressure loss in a circulation and azeotropic refrigerant without temperature glide.

Efficiency of simple Rankine cycle is

$$\eta = \frac{\dot{W}_{turbine}}{\dot{Q}_{in}} \tag{2.4}$$

$\dot{W}_{turbine}$ … Mechanical power provided by the system per unit of time

$\dot{Q}_{in}$ … Heat energy consumed by the system per unit of time

Rankine cycle in T-S diagram looks like this:

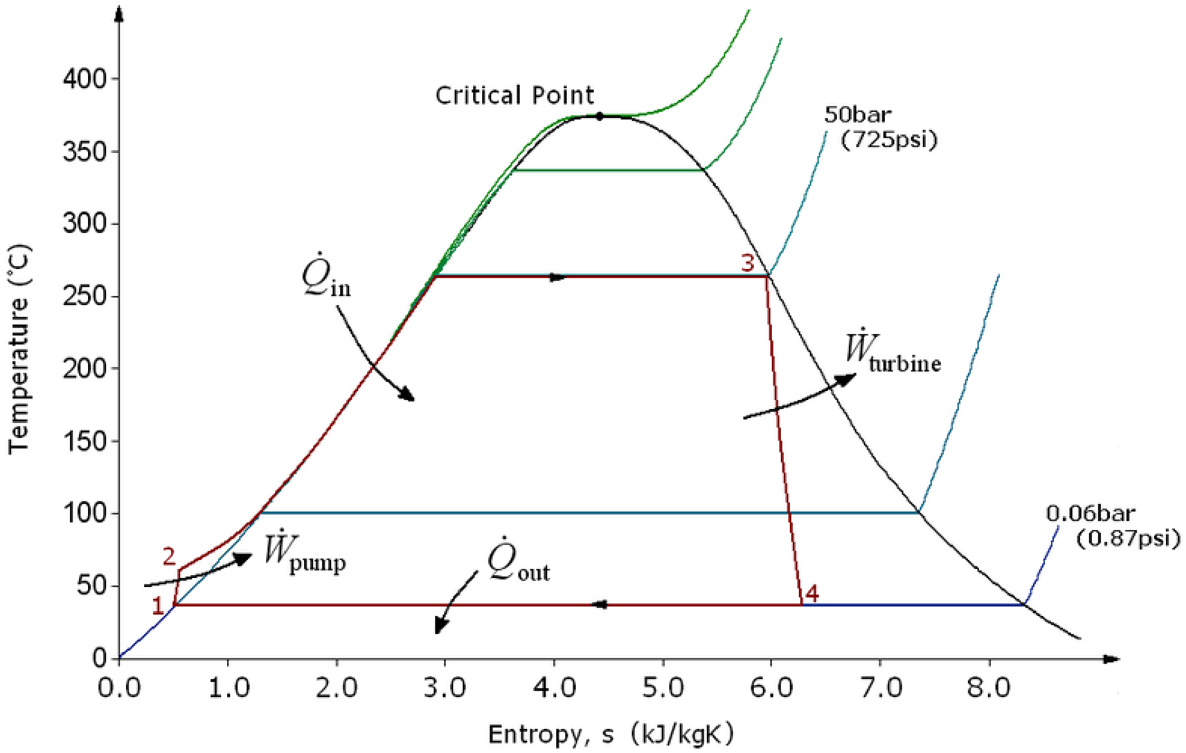


Fig. 2.4.: Rankine cycle. [13]

2.3 Real cycle of a heat pump

Real cycle of a heat pump is not exactly the same as theoretical Rankine cycle, the main differences are in:

- Overheat of refrigerator vapour, when on compressor inlet there is overheated vapour when Rankine cycle counts only with saturated vapour. This overheat is although very beneficial, because it causes higher efficiency and prolongs the lifetime of a compressor
- Subcooling of liquid refrigerant under the line of saturated liquid. This is only functional, as due to this, the right function of expansion valve is ensured
- Compression of refrigerant's vapour, as the real compression is not lossless

2.4 Types

Divided by heat sources, drive and heat distributors

2.4.1 Heat sources

Bedrock

A bedrock heat pump uses stored solar energy from the bedrock via a ground collector installed in one or more vertical boreholes, which can be up to 200 meters deep, the energy can then be used to provide space heating and hot water. This method of energy collection is ideal if the plot size is limited or if minimal impact on the plot is required, for larger installations where high levels of heat extraction are required multiple boreholes can be installed. It is always advisable to consult a specialist company regarding the design and installation of a vertical collector system. [16]

- No large plot required
- The hole in the rock maintains an even temperature throughout the year
- Little impact on the plot
- Permits passive cooling
- Suitable for all building types; large and small

Ground source

Ground heat pumps harness the stored solar energy from just beneath the ground surface, via a horizontal collector loop which is located at a depth of approximately one metre below ground level and at a distance of approximately 0.75 m apart. This is an ideal system for larger plots of land, with the exact area required being depending upon the capacity of the heat pump and the thermal conductivity of the soil in the specific location. [16]

- No drilling needed
- Lower installation costs
- The coil in the ground maintains an even temperature throughout the year
- Permits passive cooling

Air source

With an air source heat pump there is no need to dig, drill or have a large plot of land. Instead, the energy is collected directly from the surrounding air using an externally located air module. This can be located up to 30 metres from the outside of the building and is then linked to a heat pump unit which is located indoors for maximum efficiency. [16]

- Lower investment costs
- No impact on the ground
- No large plot required
- No heat losses - heat pump unit is inside, only air unit is outside
- Normally no obligation to report installation to municipal environmental health boards

Lake water source

Lake water heat pumps extract the solar energy stored in lake water through a collector loop that is lowered to the bottom of the lake. The hose is laid on the bottom of the lake or watercourse, with weights holding it securely in place and the energy is then collected in the same way as for the ground heat collector system. [16]

- No drilling needed
- Little impact on your plot
- The lake coil holds an even temperature throughout the year
- Permits passive cooling

Groundwater source

A groundwater heat pump collects energy from the groundwater. The water is pumped up from a groundwater borehole to a heat exchanger, where the energy is recovered. The water is then discharged back through another borehole. [16]

- No great size of plot required
- Little impact on your plot

2.4.2 Drive

Electrically driven heat pumps

This kind of heat pumps use electricity to run a compressor that is providing the needed pressure and flow of refrigerant

Absorption driven heat pumps

Absorption heat pumps use an ammonia-water absorption cycle to provide heating and cooling. As in a standard heat pump, the refrigerant (in this case, ammonia) is condensed in one coil to release its heat; its pressure is then reduced and the refrigerant is evaporated to absorb heat.

The difference in absorption heat pumps is that the evaporated ammonia is not pumped up in pressure in a compressor, but is instead absorbed into water. A relatively low-power pump can then pump the solution up to a higher pressure. The heat afterwards essentially boils the ammonia out of the water, starting the cycle again.

Although mainly used in industrial or commercial settings, absorption coolers are now commercially available for large residential homes, and absorption heat pumps are under development. The 5-ton residential cooler systems currently available are only appropriate for homes on the scale of 360 square meters or more. [6]

Absorption coolers and heat pumps usually only make sense in homes without an electricity source, but they have an added advantage in that they can make use of any heat source, including solar energy, geothermal hot water, or other heat sources. They are also amenable to zoned systems, in which different parts of the house are kept at different temperatures. [6]

2.4.3 Heat distributors

Water

Water is mainly used in Europe, Canada and north eastern part of the United States. Low temperature radiators and convectors of today are designed for maximum operation temperature of 45-55ºC, although conventional radiator systems require high distribution temperature, typically 60-90 ºC

Air

Air is the most common distribution medium used in Japan and the United States that are the major heat pump markets. The air is either passed directly into a room by the space-conditioning unit, or distributed through a forced-air ducted system. The output temperature of an air distribution system is usually in the range of 30-50°C.

Tab. 2.1.: Temperatures of heat distribution systems. [11]

Application		Supply temperature range (°C)
Air distribution	Air heating	30 - 50
	Floor heating; low temperature (modern)	30 - 45

Hydronic systems	Radiators	45 - 55
	High temperature (conventional) radiators	60 - 90
	District heating - hot water	70 - 100
	Under floor heating	30 - 35
District heating	District heating - hot water/stream	100 - 180
	Cooled air	10 - 15
Space cooling	Chilled water	5 - 15
	District cooling	5 - 8

3 Overview of evaluable factors

3.1 Performance factors

3.1.1 COP

COP

Coefficient of Performance is the most used coefficient that is used to define the efficiency of a heat pump. It is also used by heat pump manufacturers to compare and rate performance of machines they produce. Used like that, it can be very powerful marketing weapon used to either convince or hoodwink the customer, who is not familiar with the process of analysing performance.

In order to prevent these misunderstandings and to bond the manufacturers, technical standard EN 14511 was made in May 2004. Since then, there have been two upgrades of this standard – in December 2007 and November 2011 (English originals – Czech translations were made after date of publishing of each standard).

Name of the standard is „Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling“ and it consists of four parts:

- Part 1: Terms and definitions
- Part 2: Test conditions
- Part 3: Test method
- Part 4: Requirements

According to current published version this technical standard only concerns electrically driven heat pumps (so absorption heat pumps driven either by geo-thermal energy or solar energy aren't taken into consideration)

Coefficient itself is given by produced heat divided by compressor power input.

$$COP = \frac{|\Phi|}{|P|} \tag{3.1}$$

Testing is done based on parameters set in technical standard EN 14511. In our case while testing air-to-water heat pump in heating mode, only shortened tables for air-to-water systems in heating mode are presented.

Tab. 3.1.: Environmental conditions for units designed for installation indoors

Type	Measured quantities	Rating test
Air-to-water units with duct connection on the air inlet and outlet side	Dry bulb temperature	15ºC to 30ºC
Air-to-water units without duct connection on the air inlet and outlet side	Dry bulb temperature Wet bulb temperature	15ºC to 30ºC

Tab. 3.2: Environmental conditions for units designed for installation outdoors

Type	Measured quantities	Rating test
Air-to-water units with duct connection on the air inlet and outlet side	Dry bulb temperature	See tables 3.3, 3.4, 3.5, 3.6

Tab. 3.3: Air-to-water and air-to-brine units – Heating mode (Low temperatures)

		Outdoor heat exchanger		Indoor heat exchanger Low temperature applications	
		Inlet dry bulb temperature ºC	Inlet wet bulb temperature ºC	Inlet temperature ºC	Outlet temperature ºC
Standard rating conditions	Outdoor air	7	6	30	35
	Exhaust air	20	12	30	35
Application rating conditions	Outdoor air	2	1	a	35
	Outdoor air	-7	-8	a	35
	Outdoor air	-15	-	a	35
	Outdoor air	12	11	a	35
a – The test is performed at the flow rate obtained during the test at the standard rating conditions					

Tab. 3.4: Air-to-water and air-to-brine units – Heating mode (Medium temperatures)

		Outdoor heat exchanger		Indoor heat exchanger Medium temperature applications	
		Inlet dry bulb temperature ºC	Inlet wet bulb temperature ºC	Inlet temperature ºC	Outlet temperature ºC
Standard rating conditions	Outdoor air	7	6	45	45
	Exhaust air	20	12	40	45
Application rating conditions	Outdoor air	2	1	a	45
	Outdoor air	-7	-8	a	45
	Outdoor air	-15	-	a	45
	Outdoor air	12	11	a	45
a – The test is performed at the flow rate obtained during the test at the standard rating conditions					

Tab. 3.5: Air-to-water and air-to-brine units – Heating mode (High temperatures)

		Outdoor heat exchanger		Indoor heat exchanger High temperature applications	
		Inlet dry bulb temperature ºC	Inlet wet bulb temperature ºC	Inlet temperature ºC	Outlet temperature ºC
Standard rating conditions	Outdoor air	7	6	47	55
	Exhaust air	20	12	47	55
Application rating conditions	Outdoor air	2	1	a	55
	Outdoor air	-7	-8	a	55
	Outdoor air	-15	-	a	55
	Outdoor air	12	11	a	55
	Outdoor air	7	6	a	55
	Exhaust air	20	12	a	55
a – The test is performed at the flow rate obtained during the test at the standard rating conditions					

Tab. 3.6: Air-to-water and air-to-brine units – Heating mode (Very high temperatures)

	Outdoor heat exchanger	Indoor heat exchanger Very high temperature applications

		Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Outdoor air	7	6	55	65
	Exhaust air	20	12	55	65
Application rating conditions	Outdoor air	2	1	a	65
	Outdoor air	-7	-8	a	65
	Outdoor air	-15	-	a	65
	Outdoor air	12	11	a	55
a – The test is performed at the flow rate obtained during the test at the standard rating conditions					

Further information about testing methodology can be found in bachelor thesis by Katerina Spanihelova, “Heat pumps testing methodology”available online. [22]

3.1.2 SCOP

Seasonal Coefficient of Performance

Technical standard EN 14825:2012 also specifies other SCOPs ($SCOP_{on}$ – active mode coefficient of performance, $SCOP_{net}$ – net seasonal coefficient of performance) with different input conditions. More details can be found in mentioned European Standard. Only SCOP is evaluated in this thesis.

Terms and definitions (for use of this thesis):

SCOP

Seasonal efficiency of a unit calculated for the reference annual heating demands, which is determined from mandatory conditions given in European Standard EN 14825:2012 and used for marking, comparison and certification purposes.

For calculation of SCOP, the electricity consumption of a unit is used, including the power consumption during active mode, thermostat off mode, standby mode, electricity consumption of turned on crank case heater and where required that of an additional electric back-up heater, regardless whether this back up heater is included in the unit or not.

$SCOP_{on}$

Seasonal efficiency of a unit in active heating mode which is determined from mandatory conditions given in European Standard EN14825 and used for marking, comparison and certification purposes.

For calculation of $SCOP_{on}$, the electricity consumption during active mode is used. This excludes the power consumption during thermostat off mode, standby mode, electricity consumption of turned on crank case heater. The power consumption of an electric back up heater is added for the part load conditions where the declared capacity of the unit is lower than the heating load, regardless whether this back up heater is included in the unit or not.

$SCOP_{net}$

Seasonal efficiency of a unit in active heating mode without supplementary electric heaters which is determined from mandatory conditions given in European Standard 14825 and used for marking, comparison and certification purposes.

For calculation of $SCOP_{net}$, the electricity consumption during active mode is used. This excludes the power consumption during thermostat off mode, standby mode, electricity consumption of turned on crank case heater. For the part load conditions where the declared capacity of the unit is lower than the heating load, the power consumption of a backup heater is not included.

$T_{designh}$

Temperature conditions for average (-10ºC), colder (-22ºC) and warmer (2ºC) climates

$T_{bivalent}$

Lowest outdoor temperature point at which the heat pump is declared to have a capacity able to meet 100% of the heating capacity demand. Below this point, the unit may still deliver capacity, but additional back-up heater in necessary to fulfil the heating capacity demand.

Full load – P_{design}

Cooling ($P_{designc}$) or heating ($P_{designh}$) load of the building at T_{design} conditions. It is possible to calculate the coefficients of the unit for more than one P_{design} value.

Part load

Cooling or heating load of the building which is less than the full load.

Part load ratio

Part load or full load divided by the full load

Reference heating season(s)

Representative climate profiles by temperature bins for heating corresponding to the reference design conditions for heating. There are three reference heating seasons: „A“ – average, „C“ – colder, „W“ – warmer. The climate profiles for heating are explained further.

TOL – operation limit temperature

Lowest outdoor temperature at which the heat pump can still deliver heating capacity, as declared by the manufacturer

h_j - bin hours

Sum of all hours occurring at given temperature for a specific location. The number is rounded to a whole and is derived from representative weather data over the 1982-1999 periods. For the reference heating seasons the specific locations are Strasbourg (average), Helsinki (colder), Athens (warmer).

COP_{PL}

Heating capacity at part load or full load divided by the effective power input of a unit at specific temperature conditions where applicable

Electric back-up heater – elbu

Supplementary electric heater with COP=1, considered in the calculation of SCOP and $SCOP_{on}$ regardless of whether this is supplied together with the unit

Part load conditions in heating mode

For the purpose of calculation of application SCOP the part load ratios should be based on exact part load ratios calculated from the formula in 1st column of table.

The relevant $T_{designh}$ values are defined as follows

- T_{design} „average“
  Dry bulb temperature conditions at -10ºC outdoor temperature and 20ºC indoor temperature
- T_{design} „colder“
  Dry bulb temperature conditions at -22ºC outdoor temperature and 20ºC indoor temperature
- T_{design} „warmer“
  Dry bulb temperature conditions at +2ºC outdoor temperature and 20ºC indoor temperature

Relevant $T_{bivalent}$ is defined as follows:

- For the average heating season, the dry bulb bivalent temperature is +2ºC or lower
- For the colder heating season, the dry bulb bivalent temperature is -7ºC or lower
- For the warmer heating season, the dry bulb bivalent temperature is +7ºC or lower

Tab. 3.7: Example of part load conditions air-to water units for reference SCOP, reference $SCOP_{on}$, $SCOP_{net}$ calculation of air-to-water units for medium (45ºC) temperature application for the reference heating season „A“ – average.

Average		Outdoor heat exchanger	Indoor heat exchanger	
Part load ratio	Part load ratio %	Outdoor air Inlet dry bulb (wet bulb) temperature ºC	Inlet/outlet temperatures Fixed outlet ºC	 Variable outlet ºC
(-7-16)/(Tdesignh -16)	88	-7(-8)	[a]/45	[a]/43
(+2-16)/(Tdesignh -16)	54	2(1)	[a]/45	[a]/37
(+7-16)/(Tdesignh -16)	35	7(6)	[a]/45	[a]/33
(+12-16)/(Tdesignh -16)	15	12(11)	[a]/45	[a]/28
(TOL-16)/(Tdesignh -16)		TOL	[a]/45	[a]/43-(-7-TOL)/(-7-2)x(43-37)
(Tbivalent-16)/(Tdesignh -16)		Tbivalent	[a]/45	Variable outlet shall be calculated by interpolation between the upper and lower temperatures which are closest to the bivalent temperature.
[a] – With the water flow rate as determined at the standard rating conditions given in EN 14511-2 at 40/45 conditions for units with a fixed water flow rate, and with a fixed delta T of 5 K for units with a variable flow rate [b] – For exhaust air heat pumps part load tests are performed with an outdoor heat exchanger condition according to EN 14511				

Calculation methods for reference SCOP, reference $SCOP_{on}$, $SCOP_{net}$

The reference SCOP is defined as the reference annual heating demand divided by the annual electricity consumption. This includes the power consumption during active mode, thermostat off mode, standby mode, off mode and electricity consumption of turned on crank case heater.

The calculation of the reference SCOP that applies to all types of units is given by the following formula:

$$SCOP = \frac{Q_h}{\frac{Q_h}{SCOP_{on}} + H_{TO} * P_{TO} + H_{SB} * P_{SB} + H_{CK} * P_{CK} + H_{OFF} * P_{OFF}} \tag{3.2}$$

Q_h ... The reference annual heating demand. Expressed in kWh

$H_{TO}, H_{SB}, H_{CK}, H_{OFF}$ ... The number of hours the unit is considered to work in respectively thermostat off mode, standby mode, crankcase heater mode and off mode

$P_{TO}, P_{SB}, P_{CK}, P_{OFF}$ ... the electricity consumption during respectively thermostat off mode, standby mode, crankcase heater mode and off mode expressed in kW

$$Q_h = Pdesign_h * H_{he} \tag{3.4}$$

$Pdesign_h$ ... Full load in heating

H_{he}... number of equivalent heating hours (for each heating season are based on occupancy scenarios for certain types of buildings and climate BIN method)

$$SCOP_{on} = \frac{\sum_{j=1}^{n} h_j . P_h(T_j)}{\sum_{j=1}^{n} h_j . (\frac{P_h(T_j) - elbu(T_j)}{COP_{PL}(T_j)} + elbu(T_j))} \tag{3.5}$$

$$SCOP_{net} = \frac{\sum_{j=1}^{n}(h_j * P_h(T_j) - elbu(T_j))}{\sum_{j=1}^{n} h_j * (\frac{P_h(T_j) - elbu(T_j)}{COP_{PL}(T_j)}} \tag{3.6}$$

T_j... The bin temperature

j ... The bin number

n ... The amount of bins

$P_h(T_j)$ ... the heating demand of the building for the corresponding temperature T_j expressed in kW

h_j ... The number of bin hours occurring at the corresponding temperature T_j

$COP_{PL}(T_j)$ ... The COP values of the unit for the corresponding temperature T_j

$elbu(T_j)$ … The required capacity of an electric back-up heater for the corresponding temperature T_j, expressed in kW

Calculation procedure for determination of COP_{PL} at part load conditions for air-to-water units

Fixed capacity units

For each part load conditions the COP_{PL} is calculated as follows

$$COP_{PL(A,B,C,D)} = COP_{DC}.\frac{CR}{C_c.CR.(1-C_c)} \quad (3.7)$$

COP_{DC} … The COP corresponding to the declared capacity (DC) of the unit at the same temperature conditions as for part load conditions A, B, C, D

C_c … The degradation coefficient

CR … The capacity ratio

Capacity ratio is a ratio of the total stated cooling (heating) capacity of all operating indoor units to the stated cooling (heating) capacity of the outdoor unit at the rating conditions. [15]

To see how to calculate EER_{PL} of variable capacity units see European Standard EN 14825

For air-to-water units the degradation coefficient C_c due to the pressure equalisation effect when the unit restarts can be considered as negligible. The only effect that will impact the COP at cycling is the remaining power input when the compressor is switching off.

Than the degradation coefficient C_c is determined for each part load conditions as follows:

$$C_c = 1 - \frac{measured\ power\ of\ compressor\ off\ state}{total\ power\ input\ (full\ capacity\ at\ the\ part\ load\ conditions)} \quad (3.8)$$

3.1.3 EER, SEER

For evaluation of refrigeration systems – systems working in a cooling mode, we mostly use two coefficients – EER (Energy Efficiency Ratio) and SEER (Seasonal Energy Efficiency Ratio).

Terms and definitions

$Tdesign_c$

Temperature conditions at 35ºC dry bulb (24ºC wet bulb) outdoor temperature and 27ºC dry bulb (19ºC wet bulb) indoor temperature

EER – energy efficiency ratio

Ratio of output total cooling capacity divided by total power input, measured in Watt /Watt at one point.

EER_{DC} -- energy efficiency ratio at declared capacity

Declared cooling capacity of the unit divided by the effective power input of a unit at specific temperature conditions, expressed in kW/kW

EER_{PL} – energy efficiency at part load conditions

Cooling capacity at part load or full load conditions divided by the effective power input of a unit at specific temperature conditions. The EER includes degradation losses when the declared capacity of the unit is higher than the cooling capacity demand. [15]

SEER – seasonal energy efficiency ratio

Seasonal efficiency of a unit calculated for the reference annual cooling demand, which is determined from mandatory conditions given in European Standard EN 14825 and used for marking, comparison and certification purposes.

For calculation of SEER, the electricity consumption of a unit is used, including the electricity consumption during active mode, thermostat off mode, standby mode and electricity consumption of turned on crank case heater. [5]

$SEER_{on}$ – active mode energy efficiency ratio

Seasonal efficiency of a unit in active cooling ode which is determined from mandatory conditions given in European Standard EN 14825 and used for marking, comparison and certification purposes

For calculation of $SEER_{on}$, the electricity consumption during active mode is used, excluding the electricity consumption during thermostat off mode, standby mode and electricity consumption of turned on crankcase heater. [5]

EER calculation

EER is a single-point efficiency measurement for an AC or refrigeration compressor at specific condition.

To calculate EER we use the following:

$$EER = \frac{Cooling\ capacity\ in\ Btu/hr}{Input\ power\ in\ watts} \tag{3.9}$$

Tab.3.8: Part load conditions for reference SEER and reference $SEER_{on}$ calculation air-to-water units

Part load ratio	**Part load ratio** %	**Outdoor heat exchanger**		**Indoor heat exchanger**	
				Inlet/outlet temperatures	
		Air dry bulb temperature °C	**Fixed outlet** °C	**Variable outlet** °C	**Variable outlet** °C
A: (35-16)/(Tdesignc - 16)	100	35	12/7	12/7	23/18
B: (30-16)/(Tdesignc - 16)	74	30	[a] /7	[a] /8,5	[a] /18
C: (25-16)/(Tdesignc - 16)	47	25	[a] /7	[a] /10	[a] /18
D: (20-16)/(Tdesignc - 16)	21	20	[a] /7	[a] /11,5	[a] /18

[a] – With the water flow rate as determined during „A“ test for units with a fixed water flow rate or with a fixed delta T of 5 K for units with a variable flow rate

Calculation methods for reference SEER and reference $SEER_{on}$

Reference SEER is the reference annual cooling demand divided by the annual electricity consumption. This includes the power consumption during active mode, thermostat off mode, standby mode, off mode and electricity consumption of turned on crankcase heater. [5]

$$SEER = \frac{Q_c}{\frac{Q_c}{SEER_{on}} + H_{TO}\ x\ P_{TO} + H_{SB}\ x\ P_{SB} + H_{CK} x\ P_{CK} + H_{OFF} x\ P_{OFF}} \tag{3.10}$$

$Q_c \ldots$ The reference annual cooling demand expressed in kWh

$H_{TO}, H_{SB}, H_{CK}, H_{OFF}\ \ldots$ The number of hours the unit is considered to work in respectively thermostat off mode, standby mode, crankcase heater mode and off mode

$P_{TO}, P_{SB}, P_{CK}, P_{OFF}\ \ldots$ The electricity consumption during respectively thermostat off mode, standby mode, crankcase heater mode and off mode expressed in kW

$$Q_c = Pdesign_c\ x\ H_{ce} \tag{3.11}$$

$$SEER_{on} = \frac{\sum_{j=1}^{n} h_j \,. P_c(T_j)}{\sum_{j=1}^{n} h_j . (\frac{P_c(T_j)}{EER_{PL}(T_j)}} \tag{3.12}$$

$T_j \ldots$	The bin temperature
$j \ldots$	The bin number
$n \ldots$	The amount of bins
$P_c(T_j) \ldots$	The cooling demand of the building for the corresponding temperature T_j expressed in kW
$h_j \ldots$	The number of bin hours occuring at the corresponding temperature T_j
$EER(T_j) \ldots$	The EER values of the unit for the corresponding temperature T_j

Calculation procedure for determination of EER_{PL} at part load conditions for air-to-water units

Fixed capacity units

For each part load conditions the EER_{PL} is calculated as follows

$$EER_{PL(B,C,D)} = EER_{DC} . \frac{CR}{C_c \,. CR \,. (1 - C_c)} \tag{3.13}$$

$EER_{DC} \ldots$	The EER corresponding to the declared capacity (DC) of the unit at the same temperature conditions as for part load conditions B, C and D
$C_c \ldots$	The degradation coefficient
$CR \ldots$	The capacity ratio

To see how to calculate EER_{PL} of variable capacity units see European Standard EN 14825.

While using the mentioned standard, calculating EER is a useful method of comparing different compressors and operating-condition choices. Generally speaking, compressor with higher EER could be expected to perform better in a system than one that has lower EER.

Biggest benefit of EER is that it is standardized as a measure of compressor performance, although it cannot represent how well actually the system operates. On the other hand, the biggest limitation in usage is that in any given location, the ambient temperature varies greatly from a single condensing-temperature rating point. In some regions, temperatures never even reach the ambient temperature corresponding to the EER rating-point condition.

SEER is related to EER, the efficiency rating for equipment at a particular operating point and is calculated over a range of expected external temperatures, so the relation is very relative, depending on location, because equipment performance is dependent on air temperature, humidity and atmospheric pressure.

SEER also has its limitations. It is a seasonal value, and performance at severe conditions is not heavily weighted, it is not an indicator of demand. SEER might be a good way to compare equipment, but is not an ideal indicator of energy use, neglecting the high load on hot summer days in this case.

3.1.4 AEER

While taking seasonality into account, AEER for a fixed load refrigeration system does a better job, not neglecting the high load on hot summer days. It is a weighted average performance of a refrigeration system, using varying condensing temperatures tied to the actual weather data for location. AEER is a single number that represents an average performance for whole year and leads easily to calculations of total annual power and energy cost.

There is a case study done by Emerson Climate Technologies, Inc. using EER in comparison with AEER to evaluate the efficiency of walk-in refrigeration process. The results of this study showed, that one compressor having lower EER than the other could actually be more efficient when the performance of both compressor is evaluated through the entire operating range, using the AEER analysis. [27]

3.2 Ecological factors

3.2.1 Life cycle evaluation

3.2.1.1 GWP

Global-warming potential (GWP) is a relative measure of how much heat a greenhouse gas traps in the atmosphere. It compares the amount of heat trapped by a certain mass of the gas in question to the amount of heat trapped by a similar mass of carbon dioxide. A GWP is calculated over a specific time interval, commonly 20, 100 or 500 years. GWP is expressed as a factor of carbon dioxide (whose GWP is standardized to 1). [29]

The GWP depends on the following factors:

- the absorption of infrared radiation by a given species
- the spectral location of its absorbing wavelengths
- the atmospheric lifetime of the species

Thus, a high GWP correlates with a large infrared absorption and a long atmospheric lifetime. The dependence of GWP on the wavelength of absorption is more complicated. Even if a gas absorbs radiation efficiently at a certain wavelength, this may not affect its GWP much if the atmosphere already absorbs most radiation at that wavelength. A gas has the most effect if it absorbs in a "window" of wavelengths where the atmosphere is fairly transparent. The dependence of GWP as a function of wavelength has been found empirically and published as a graph. [29]

Intro

While testing heat pump, we calculate efficiency of usage of electrical energy, that is consumed by the compressor (or the whole unit) and we compare it to the heat provided (in air conditioning, the coldness).

From ecological point of view, we are not only saving our money while using less energy than with conventional heating systems, but also helping the environment, while producing less CO_2 emissions and thus reducing CO_2 potential. We can also calculate Primary Energy reduction potential and compare it with other heat pump systems and with conventional heating systems.

Energy consumption

Energy needed to heat our houses with common systems can be calculated as following. As the AE is < 1, these systems will need more final energy than the usable energy that is delivered to the system.

$$final\ energy_{common-systems}[kWh] = \frac{usable\ energy\ [kWh]}{AE} \tag{3.14}$$

AE … Annual efficiency (in common systems it varies approximately from 0,75 to 0,86 depending on fuel type and boiler system type)

And on the other hand, while calculating a final energy consumption of a heat pump, we use following:

$$final\ energy_{heatpump}[kWh] = \frac{usable\ energy\ [kWh]}{SPF[-]} \tag{3.15}$$

With SPF being number > 1, it is obvious, that the final energy demand will be lower than the usable energy demand

Life cycle of a heat pump

During the life cycle of a heat pump, we can identify more phases, as following:

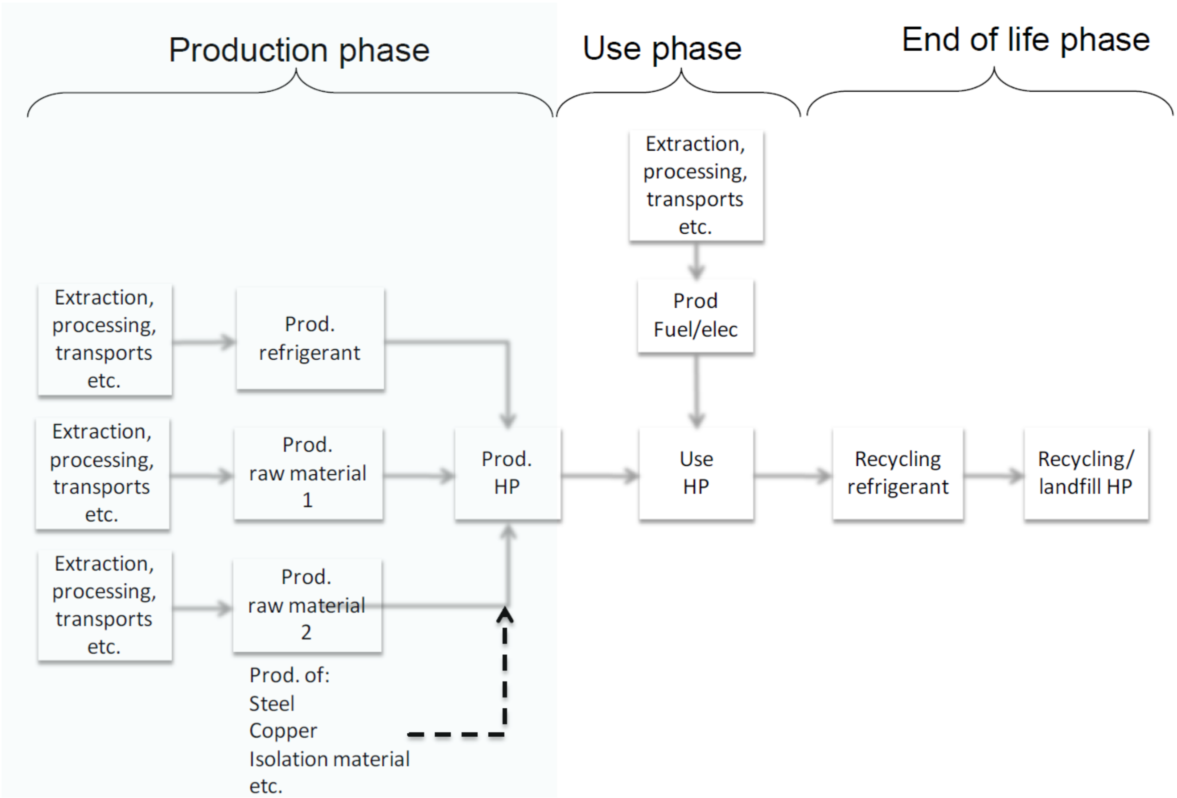


Fig. 3.1.: Life cycle of a heat pump. [26]

It was proved by other studies, that emissions related to the production and end of life phase are small in comparison with use phase emissions, also the coefficients further discussed are not taking these phases into consideration. [14,26]

3.2.2 TEWI

Terms and definitions

$AE\ [-]\ \ldots$ Annual Efficiency

$f_p\ \left[\frac{kWh}{kWh}\right]\ \ldots$ Primary energy factor

$GWP[kg\ CO_2e/kg]\ \ldots$ Global Warming Potential of refrigerant, relative to CO_2 (GWP CO_2 =1)

$TEWI\ [\frac{kg\ CO_2}{lifetime}]\ \ldots$ Total Equivalent Warming Impact

$K\ \left[kg\frac{CO_2e}{kWh}\right]\ \ldots$ CO_2e – emission coefficient

$m\ [kg]\ \ldots$ Refrigerant content of the heat pump

$L\ \left[\frac{kg\ refrigerant}{year}\right]\ \ldots$ Refrigerant losses of refrigerant per year

$n\ [year]\ \ldots$ Operation time of the system in years

$\alpha_{Recovery}\ [-]\ \ldots$ Recovery factor during the disposal of the system after life time

$E_{annual}[\frac{kWh}{year}]\ \ldots$ Annual consumption of electrical energy

TEWI

The Total Equivalent Warming Impact was developed as a comparative index of the global warming potential based on the total related emissions of greenhouse gases during the operation of the equipment and the disposal of the operating fluids at the end of a life cycle.

System boundaries for calculation the primary energy use and CO_2e emissions are presented in a [picture]. These will enable the comparison of heat pump systems by allowing for the calculation of the CO_2e emission and primary energy reduction potential. [26]

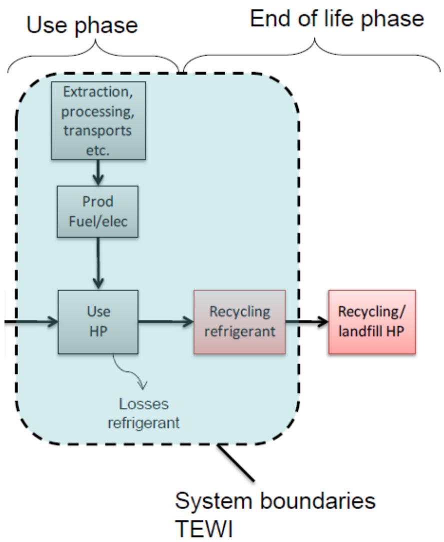


Fig. 3.2.: System boundaries for TEWI. [26]

The TEWI is calculated according to European Standard EN 378-1, resulting in easy-to-handle analysis of the different systems. It provides an estimation of the CO_2e-emissions of direct (leakage of refrigerant) and indirect (produced during the generation of electricity used to run the system). [18]

Calculation

$$TEWI = GWP_{direct} + GWP_{indirect} \tag{3.16}$$

$$TEWI = (GWP\ .L_{annual}\ .\ n) + GWP.m\,.\left(1-\alpha_{recovery}\right) + (E_{annual}\,.K\,.n) \tag{3.17}$$

Tab. 3.9: Annual leak rates by equipment class/application

Equipment class/application	Annual leak rates (% p.a.)		
	Lower	Typical	Upper
Refrigeration applications			
Centralised system (e.g. supermarket rack)	5 %	Maintained 12,5 %, otherwise 15 %	23 %
Condensing units	5 %	Maintained 12,5 %, otherwise 15 %	23 %
Road transport	20 %	30 %	40 %
Air conditioning applications			
Chillers	5 %	7 %	9 %
Chillers (using HCFC)	-	2 %	-
Rooftop packaged systems	4 %	5 %	9 %
Split systems	3 %	4 %	9 %
Window/wall units and portable	-	2 %	-

GWP values for some refrigerants

The GWP presented is the $GWP_{(100)}$, meaning that it is a global warming potential of a substance in relation to the global warming potential for CO_2 for the time of 100 years. [Table] shows this GWP as defined in European Standard EN378 of the most common used refrigerants.

Tab. 3.10: GWP values for some refrigerants

	$GWP_{(100)}$ [kg_{CO2} / $kg_{refrigerant}$]
R134a	1300
R407C	1650
R410A	1980
R404A	3780
R290	3
R744 (CO_2)	1

Recovery rate

The recovery factor shows how much refrigerant will be lost during the disposal of the system after life time. The Intergovernmental Panel of Climate Change issued practice guidelines of refrigerant recovery. Depending of equipment class, the rate varies from 70% to 95%. In systems with refrigerant charge lower than 100kg, the rate would be expected around 70%, in applications with greater charge around 90%-95%. [14]

Indirect emissions

While calculating indirect emissions, we are taking into consideration average emissions intensity of total electric sector generation of the area, state or region. It is than dependent on a fuel mix used for electricity generation in that each area.

We calculate the coefficient K as follow:

$$K = \frac{CO_2\text{−}emisions\ [g]}{E_{complete}\ [kWh]} \tag{3.18}$$

3.2.3 LCCP

LCCP

Until recently, there has been only TEWI model used for evaluation of a global warming impact of a refrigeration and air conditioning equipment. It takes into consideration both direct and indirect emissions and makes a balanced comparative index of global warming impact. The direct impact is calculated from GWP value assigned to each refrigerant and the indirect impact is calculated from CO_2 emissions associated with the electricity generation. [9]

LCCP x TEWI

As shown before, in the lifetime of a heat pump consists of more phases, that can be taken into consideration, in the table below we can find differences between TEWI and LCCP [3].

Tab. 3.11: Differences between LCCP and TEWI [14]

TEWI		**LCCP**	
Direct emissions	Refrigerant leaks (including end of life phase)	**Direct emissions**	Refrigerant leaks (including end of life phase)
			Chemical refrigerant emissions (including atmospheric reaction products, manufacturing leakage and end of life phase)
Indirect emissions	Energy generation for run of a HP	**Indirect emissions**	Energy generation for run of a HP
			Energy consumption

		from chemical production and transport, manufacturing/assembly and end of life phase

On a graphical timeline as follows:

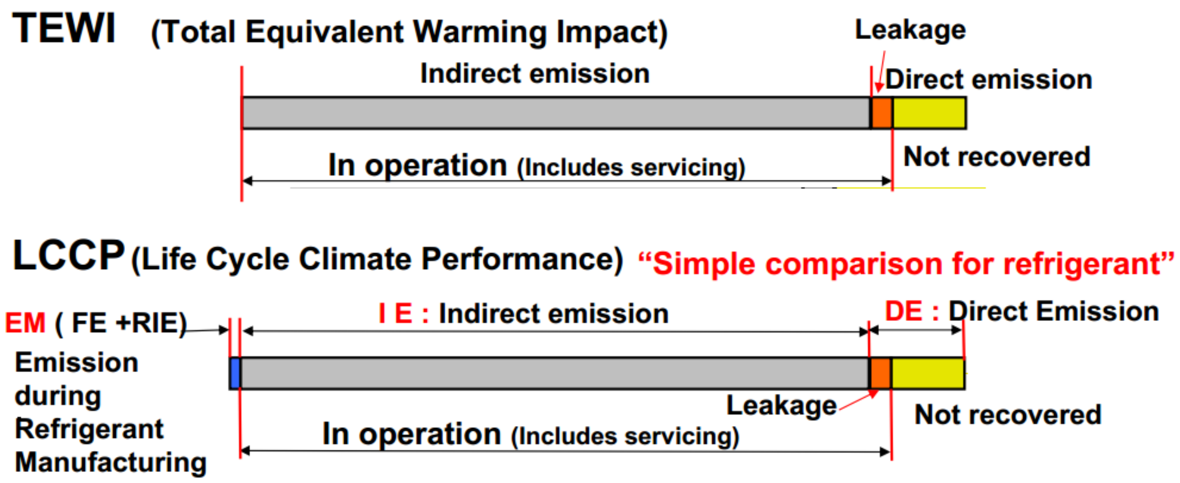


Fig. 3.3.: Difference between LCCP and TEWI. [21]

Calculation

$$LCCP = direct\ emissions + indirect\ emissions \quad (3.19)$$

$$Direct\ emissions = GWP\ .m\ .L\ .n + GWP\ .m\ .R \quad (3.20)$$

$$Indirect\ emissions = E_{annual}.K\ .n + CO_2e\ .m_m \quad (3.21)$$

$$LCCP = GWP\ .m\ .L\ .n + GWP\ .m\ .R + E_{annual}.K\ .n + CO_2e\ .m_m \quad (3.22)$$

R ... Percentage of loss when reclaim

$CO_2e\ [\frac{CO_2kg}{kg} material]$ ... CO_2 equivalent due to components manufacturing

m_m ... Mass of materials

AHRTI model

In October 2011, AHRTI (Air-conditioning, Heating and Refrigeration Technology Institute) based in the Silver Springs, USA, released a Life Cycle Climate Performance Model for Residential Heat Pumps Systems. They developed Microsoft Excel based program to simulate LCCP for residential heat pumps. It includes the direct impact of refrigerant emissions, the indirect impact of energy consumption used to operate the heat pump system and the energy to manufacture and safely dispose the system and refrigerant.

With appropriate input, the program can handle different heat pump systems, refrigerants, locations and CO_2 emission profiles of power plants. With its modular structure, the program can be easily modified to evaluate other air conditioning or refrigerant system. Although only available data are for the USA.

The flowchart of such program is presented as follows:

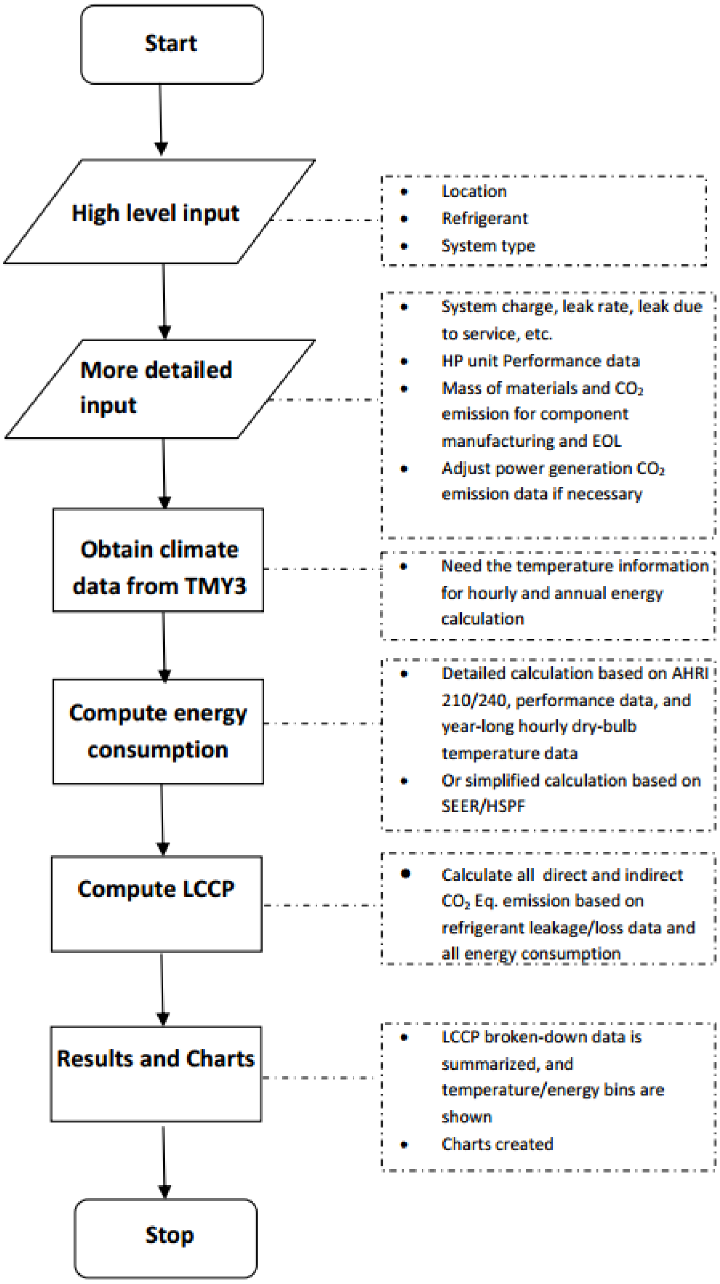


Fig 3.4.: Flowchart of AHRTI model program

Manual

There has been also a manual to a program published that is available online. [28]

3.2.4 ECOP

Ecological Coefficient of Performance is a coefficient used in modelling of thermodynamical processes:

- Irreversible Brayton cycle

- Irreversible Carnot cycle

Studies made by Turkish researchers Y. Ust from Yildiz Technical University, B. Sahin from Yildiz Technical University and A. Kodal from Istanbul Technical University have shown how to calculate COPs and ECOPs. As these calculations based on model cycles are very complex, here only basics will be presented.

Definition

ECOP is defined as the ratio of power output to the loss rate of availability

Calculation

$$ECOP = \frac{\dot{W}}{T_0.\dot{S_g}} \tag{3.23}$$

$\dot{W}$ … Power output

T_0 … Temperature of cold thermal reservoir

S_g … Entropy generation rate

Conclusion

It was suggested by above mentioned researchers, that for a practical design, the environmental effect related to entropy-generation rate, thermal efficiency and power output should be considered together. As a consequence, a design based on the maximum of the ECOP objective-function for a regenerative Brayton heat-engine represents the best compromise between the thermal efficiency, power output, investment cost and environmental impact. [23,24,25]

3.3 Field tests

SPF

Seasonal Performance Factor is a term used mainly for real installation, compared to the Coefficient of Performance, COP, or Seasonal Coefficient of Performance, SCOP, which should be evaluated in controlled lab environment according to European Standards EN 14511 and EN 14825 respectively. How do we estimate SPF depends on a situation under which we evaluate it. [20]

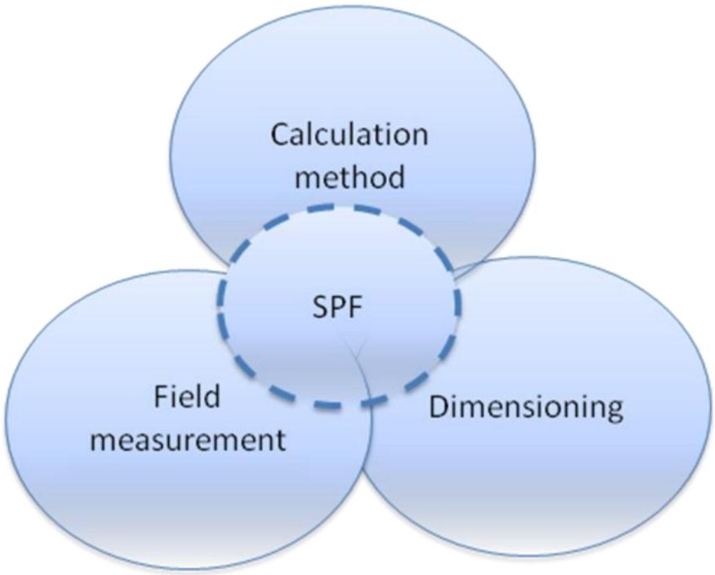


Fig. 3.5.: What is Seasonal Performance Factor. [20]

As mentioned, in order to evaluate the efficiency and performance of real installations, we also use other coefficients than COP and SCOP. The main difference is that while counting COP, we count it according to available data in each time and then average it, and on the other hand, while counting any of the SPFs, we at first make adjustment to data and then count it.

While testing heat pump, there are certain the values we need to know. While doing that, we measure power input. In following diagram we can find all measured input values. Letter „P“ marks the electrical consumer. These include the heat pump (compressor and control unit), the ventilator as well as the charge pump and the back-up heater (optional).

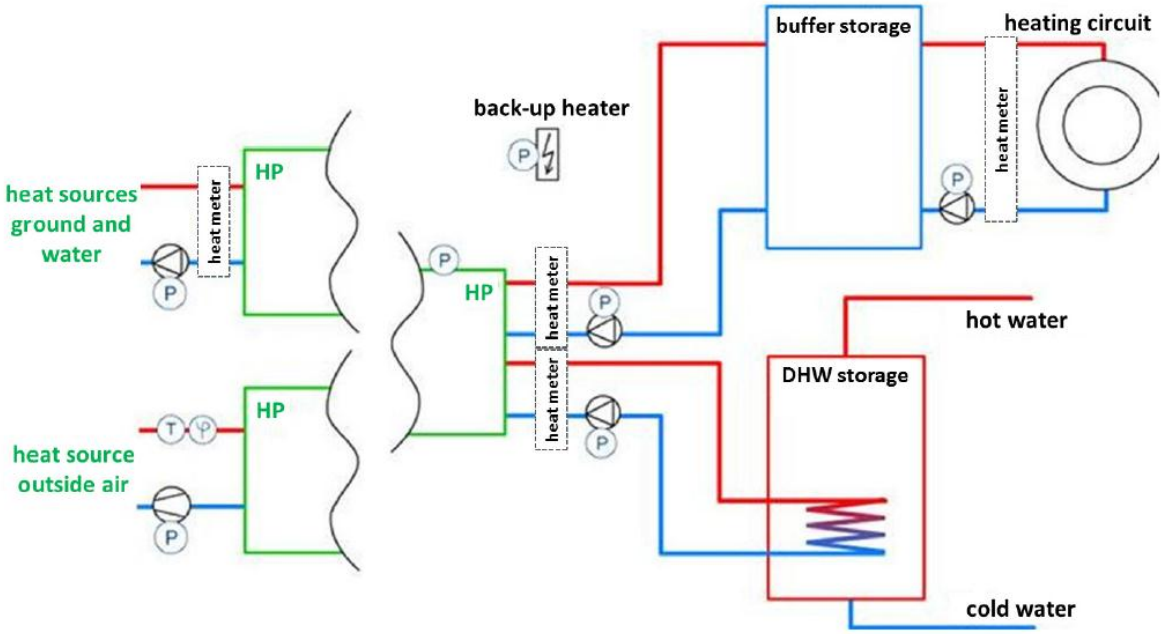


Fig. 3.5: Electricity consumers [19]

Also what can be understood from previous picture is that for one system/installation we can get different SPFs according to values that we take into consideration. In principle, there can be different system boundaries for the input as well as the output energy. In order to explain them, an illustration of a heat pump's standard hydraulic scheme is used. The red frame marks the produced thermal energy.

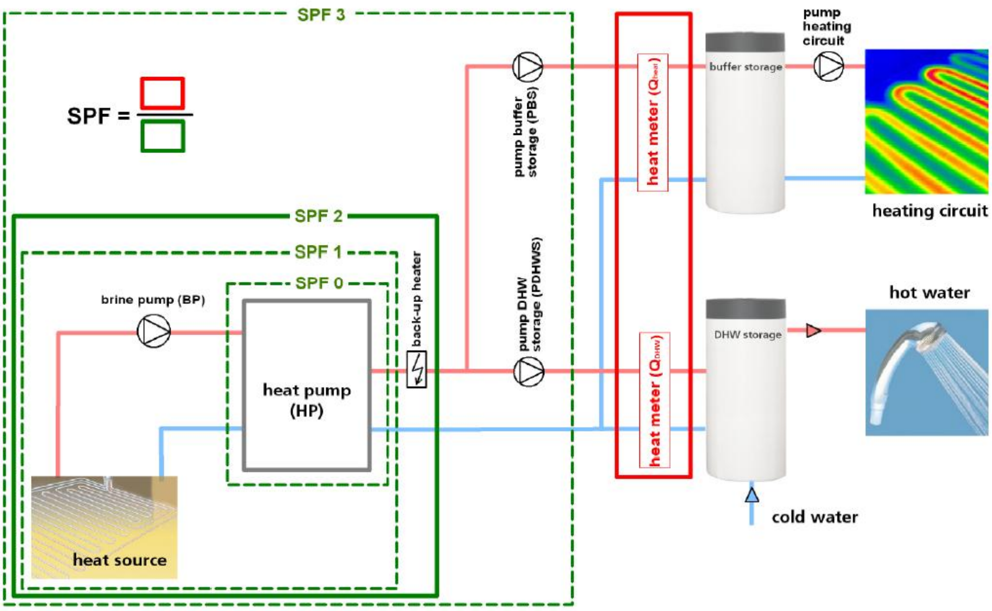


Fig. 3.6.: System boundaries according to Fraunhofer ISE study [19]

Then the equations for calculating different SPFs of a heat pump are as following [19]:

$$SPF\ 0 = \frac{Q_{heat,HP} + Q_{DHW,HP}}{W_{comp+cont}} \tag{3.23}$$

$$SPF\ 1 = \frac{Q_{heat,HP} + Q_{DHW,HP}}{W_{comp+cont} + W_{BP|Fan|WP}} \tag{3.24}$$

$$SPF\ 2 = \frac{Q_{heat,HP} + Q_{DHW,HP} + Q_{heat|DHW,back-up}}{W_{comp+cont} + W_{BP|Fan|WP} + W_{back-up}} \tag{3.25}$$

$$SPF\ 3 = \frac{Q_{heat,HP} + Q_{DHW,HP} + Q_{heat|DHW,back-up}}{W_{comp+cont} + W_{BP|Fan|WP} + W_{back-up} + W_{PDHWS+PBS}} \tag{3.26}$$

$SPF\ \ldots$	Seasonal Performance Factor
$Q_{heat,HP}\ \cdots$	Heating energy produced by heat pump
$Q_{DHW,HP} \cdots$	DHW energy produced by heat pump
$Q_{heat\|DHW,back-up} \cdots$	Energy produced by electric back-up heater
$W_{comp+cont} \cdots$	Energy consumed by compressor and control unit
$W_{BP\|Fan\|WP} \cdots$	Energy consumed by brine pump, fans or well pump
$W_{back-up} \cdots$	Energy consumed by back-up heater
$W_{PDHWS+PBS} \cdots$	Energy consumed by charge pumps

3.3.1.1 Testing methods and standards (used by SP)

SP

Technical Research Institute of Sweden is a leading international research institute. One of the major activities is testing and developing methodology of testing heat pumps according to European Standards and proposing new ones.

P-marking/SPCR130

The P-mark on a heat pump indicates that the heat pump fulfils the requirements set out in certification rules SPCR 130 . These rules have been developed in conjunction with manufacturers and relevant authorities, and specify required performance levels such as COP, documentation and quality assurance in manufacture. Certification of products entitles the manufacturer to display the P-mark on the certified products. Air-to-air, air-to-water and liquid-to-water heat pumps can be tested for approval for P-marking. [30]

SP method 0033 (Calculation of annual energy savings)

SP 0033 is a method used to calculate the seasonal performance factor of a system and the annual energy savings, in comparison with a house having direct electric heating. The data used in the calculations are based on results from a number of test points derived from EN 14511:2007 performance testing and some additional test points.

The heating demand as calculated in the program varies in accordance with a duration diagram for the particular house. Calculation is performed for the house with loss factors of 109 W/K and 199 W/K at 20 °C indoor temperature, and calculates the estimated net energy savings, excluding energy for domestic electricity and hot water. The houses are postulated as located in areas with average annual temperatures of 8.2 °C, 6.1 °C and 1.3 °C, which correspond to locations in Malmö, Borås and Luleå respectively. The lowest heat pump source temperature in the calculation is assumed to be -15 °C, and there is no heating demand above an outdoor air temperature of +17 °C.

Energy Savings Calculation SP Method 0033 is currently used for air-to-air heat pumps, air-to-water heat pumps and liquid-to-water heat pumps. [30]

EHPA Testing Regulations and EHPA Quality Label

The EHPA Quality Label scheme operates in Sweden, Switzerland, Germany and Austria. SP performs the necessary tests and inspections for the scheme. Testing includes performance, security, sound level and electrical characteristics, together with inspection of technical data, maintenance and installation instructions.

Air-to-water, water-to-water and direct-expansion heat pumps can be tested and inspected for approval under this scheme.

More information on EHPA Quality label for heat pumps can be found online. [10]

prEN 15879-1

This standard describes the testing procedure for direct-expansion ground-coupled heat pumps. The heating coil of the heat pump is assembled in a brine bath held at constant temperature during the test. A complete test includes performance and functional inspections and a review of technical data, maintenance and installation instructions. A function test includes measurement of the operating temperature range, correct operation after a power failure, and correct operation of safety devices in the event of pressure loss in the refrigerant or during failure of heat transfer medium flow.

Direct-expansion heat pumps may be tested in accordance with this method. [30]

CEN TS 14825

This test method is used to test heating or cooling mode operation under part-load conditions.

The method specifies the testing procedure, depending on whether the heat pump has constant capacity, variable capacity or incremental capacity control. This is a technical specification associated with EN 14511, and is used for testing the same types of heat pumps: air conditioners, liquid chillers, air-to-air heat pumps, water-to-air heat pumps, and water-to-water heat pumps. [30]

SP Method 2800

SP Method 2008 specifies the method of performance-testing of electrically driven exhaust air heat pumps, determining their thermal power capacity, hot water production capacity and ventilation performance. Air leakage, thermal insulation and standby power requirement are also tested. The data from the test are used to calculate the system seasonal performance factor. [30]

EN 255-3:1997 (Domestic hot water)

SP is accredited to perform sanitary water heating tests in accordance with EN 255 31. The test involves two draw offs of half the volume of the storage tank, together with a single tapping to determine the maximum quantity of useable hot water and the reference hot water temperature. The heating-up periods, energy use, standby power losses and COP of the heat pump are measured and evaluated. The test also includes verification of product marking on the heat pump.

The method is used only for electrically driven heat pumps. [30]

prEN 16147 (sound test)

This test method specifies a number of draw-offs over a 24-hour period, measuring and evaluating the heating-up periods, energy use, standby power losses and COP of the heat pump. The maximum quantity of useable hot water and the reference hot water temperature are measured during a single draw-off. The method includes testing of safety functions and inspection of technical data, maintenance and installation instructions.

The method is used only for electrically driven heat pumps. [30]

SS-EN 12102 (sound test)

SP is accredited for performing sound level tests to SS EN 12102. The method is used to measure the airborne noise to the surroundings from heat pumps, air conditioners, liquid chillers and dehumidifiers when in operation. The method is used for sound power level determination. [30]

ISO 3747:2010 (sound test)

This standard specifies a method for determination of sound power level for heat pumps. Sound power level measurements are performed at specified positions around the sound source and around a reference sound source, from which the sound power level can be calculated. [30]

Standards and methods for field measurements

SP-method 1721 (field measurement method)
SP Method 1721 is a method for measurement of the capacity and electricity consumption of air-to-air heat pumps in the field. It can be used for heat pumps in both heating and cooling operating modes. [30]

SS 2620
This standard is used when testing the performance of large heat pumps. The standard refers to field testing of heat pumps in field. It is also applicable for delivery testing of heat pumps at the factory or equivalent. The standard relates primarily to electrically driven heat pumps, but where applicable it can also be used for heat pumps with other drive motors and for refrigeration applications. The standard does not include test on electrical safety, mechanical safety, control equipment and sound power level. [30]

4 Data evaluation

4.1 Evaluated factors

Evaluated factors

This thesis was written according to needs of heat pumps manufacturer Emerson Climate Technologies Inc., local branch in Mikulov. Data that are further evaluated were also provided by the same party.

No. of working hours in each month

- To have an idea about workload during the year, graph with no. of working hours in each month will be presented.

No. of working hours versus compressor speed

- In this part, graph should be presented that show relation between working hours of a compressor and number of hours spent in each scale of compressor speed.

- That is particularly important for feedback and evaluation done for the compressor manufacturer, in meaning of determination operation envelope of a compressor and its adjustment.

No. of compressor starts in each month

- In order to evaluate the work load during the year, number of compressor starts in each month was evaluated.

No. of defrosts in each month

- Defrost of a HP occurs mainly when the heat pump is in a heating mode during cold days, e.g. when the ambient temperature gets near 0°C or below. The moisture in the outside air freezes on a heat exchanger. It reduces the air flow that is generated by the fan, reducing the efficiency of a cycle dramatically. In extreme cases, the frozen moisture can even cause damage to the outdoor unit.

- Defrost itself means, that the cycle is reversed for a while, so that the outdoor heat exchanger is used as condenser until the frozen moisture doesn’t fall off.

- How often will a defrost cycle occurs depends mainly on the outdoor temperature and the air humidity, the amount of heating load the unit is trying to deliver and the condition of the heat pump system.

Interval of data logging

- As the interval of data logging differ month to month (in some cases even during the month), for some months we have nearly 8 times more data than to others. The analysis that was performed took into account these differences, as computing weighted averages of data.

Available data from each month

- As the system is now running in test mode, it happened that from some days or part of days we don‘t have data. The amount of data available to evaluation for each month is shown in graph „Available data in each month“.

Measured COP

- As the system itself is computing COP in each data logging interval, there is a weighted average presented from each month.

Measured SCOP

- The heat pump system is also calculating SCOP. This is the name shown in data logging. „S“ means in this case seasonal and is an inaccurate word to use. This calculation of SCOP is done from system start to till the system crashes (for any reason). In our case that was quite a frequent case, as in testing mode, the system was recalculating this value every time it restarted.

Calculated SPF

- According to data available, the calculation of Seasonal Performance Factor was processed based on amount of produced heating energy and electrical energy consumed for producing it.

Comparison of coefficient‘s values

- In the end, comparison of coefficient’s values that were computed or later evaluated is presented.

4.2 Conditions

4.2.1 The subject of testing

The subject of field test is a newly developed air-to-water heat pump.

Parameters

Type – WPL 25 IK-2 IE

Refrigerant – R410A, 6kg,

Working volume

Air – 2300 m^3/h, -20ºC/+40ºC

Water – 1,1 m^3/h, +15ºC/+65ºC

Claimed performance

Point of measurement	Heat output	Power Consumption	Coefficient of Performance
A7/W35	5,82	1,21	4,81
A2/W35	7,06	1,81	3,90
A-7/W35	12,86	4,13	3,11

4.2.2 Sensors

- The logging itself was operated by Emerson employee and was not in control of author of this thesis.

In figures 4.2. and 4.3. we can see the heat pump during manufacturing and testing process in Mikulov branch of Emerson Inc.

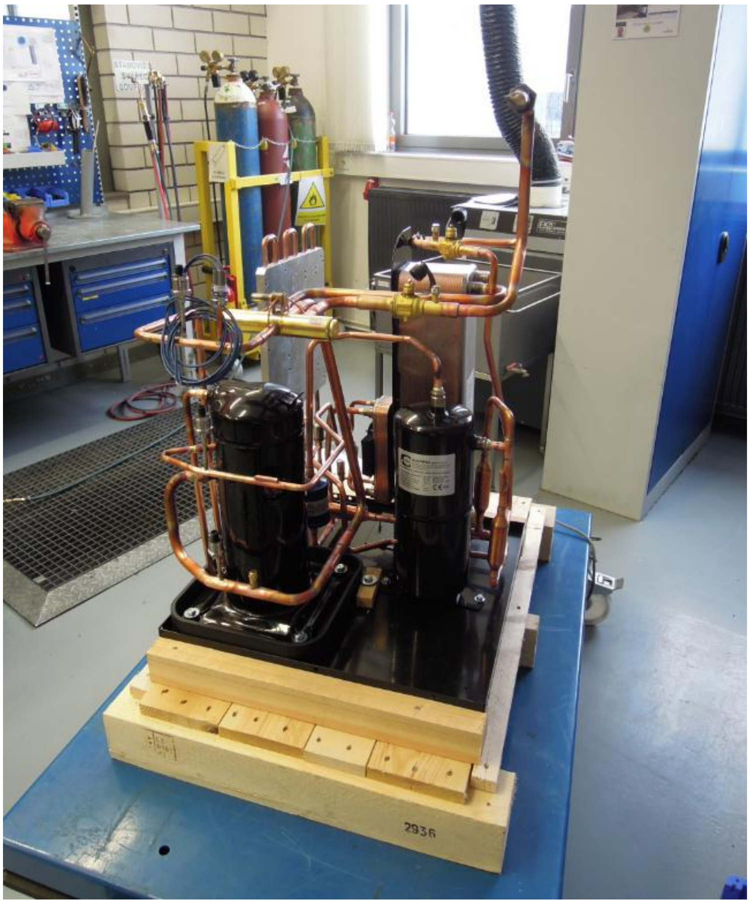


Fig 4.2 and 4.3.: Heat pump during manufacturing and testing process

4.3 Evaluation

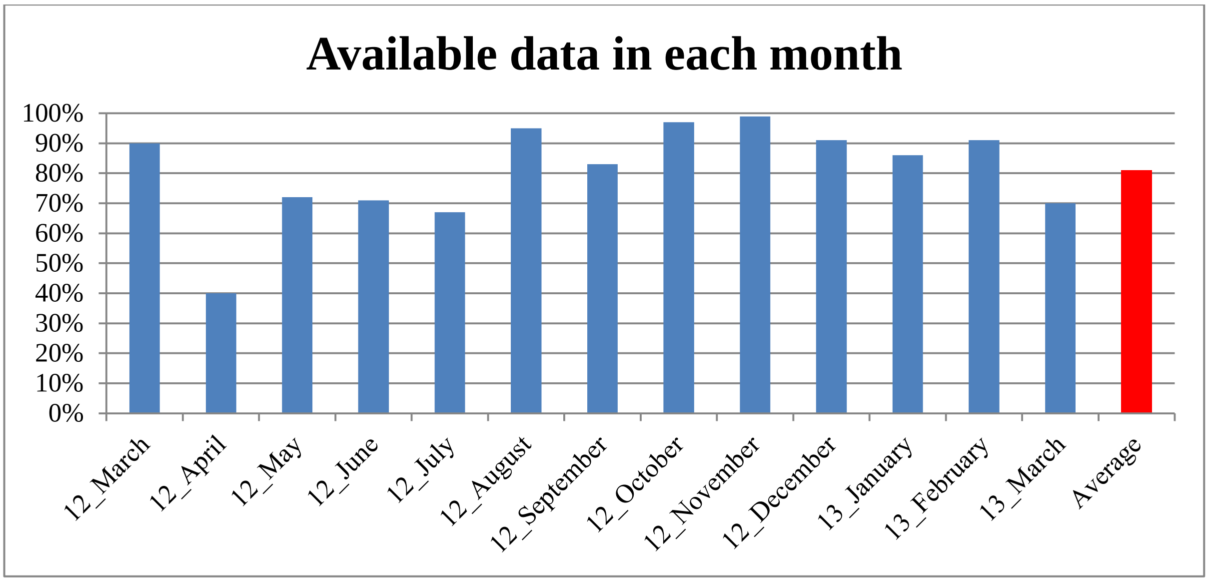


Fig. 4.4.: Available data in each month

In most months, we have at least 80% of data available. On the other hand, from e.g. April 2012, we have only 40% of data and from three other months only around 70%.

There cannot be seen any rule or correlation in amount of data from each month.

The fact of this lack of data was explained to the author as that when the system is not running (it is in „off mode“) it doesn’t send data to logging system.

It is questionable if in this case there is any power consumption, as this would certainly lower Seasonal Coefficient of Performance and Seasonal Performance Factor.

Data that are evaluated were made as a weighted average from each month, as monthly seasonality was most important for the manufacturer.

Interval of data logging also differed month to month, same as the number of data that heat pump system was sending to log. As measurement was not operated by the author of this thesis, and these inconsistences in data logging and amount of data were known, for evaluation, weighted averages were used.

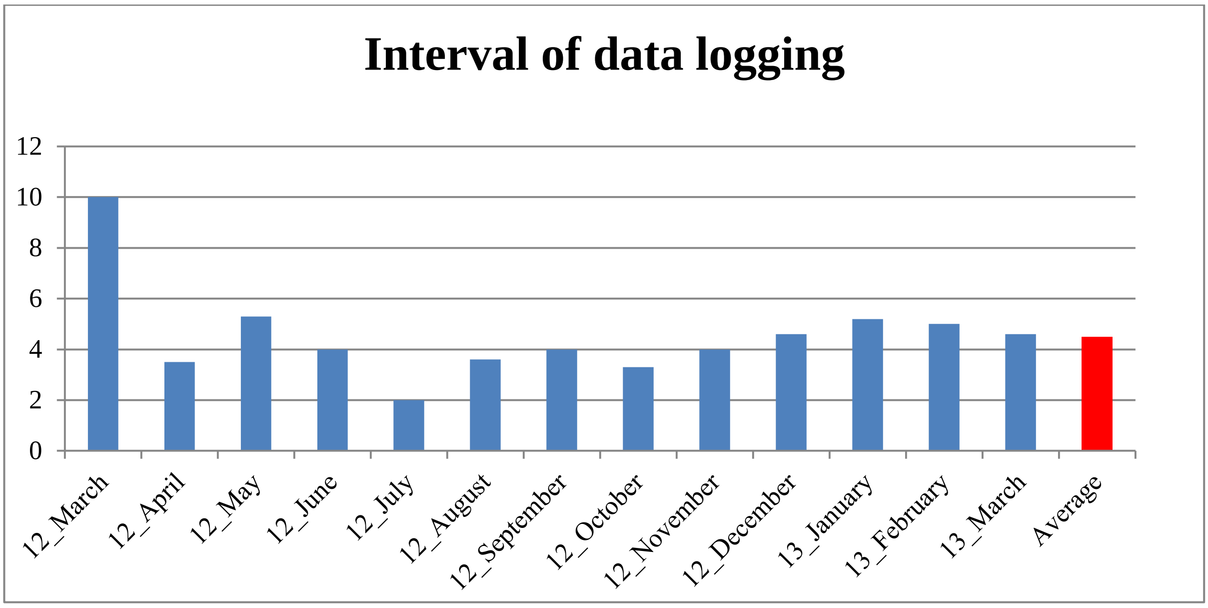


Fig. 4.5.: Interval of data logging

Number of compressor starts of mentioned heat pump system was then evaluated. It was found, that the greatest stress for compressor is during seasonal switch, during Autumn and Spring.

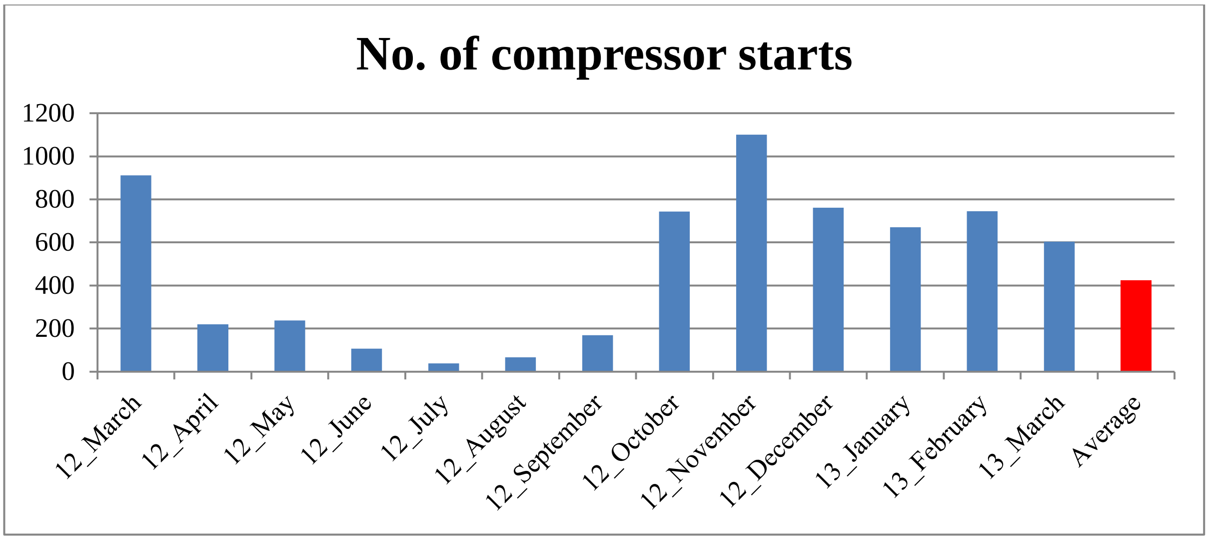


Fig. 4.6.: Number of compressor starts in each month

During heating period, the system is running in longer intervals, so then number of starts is lower, but number of working hours of compressor is higher.

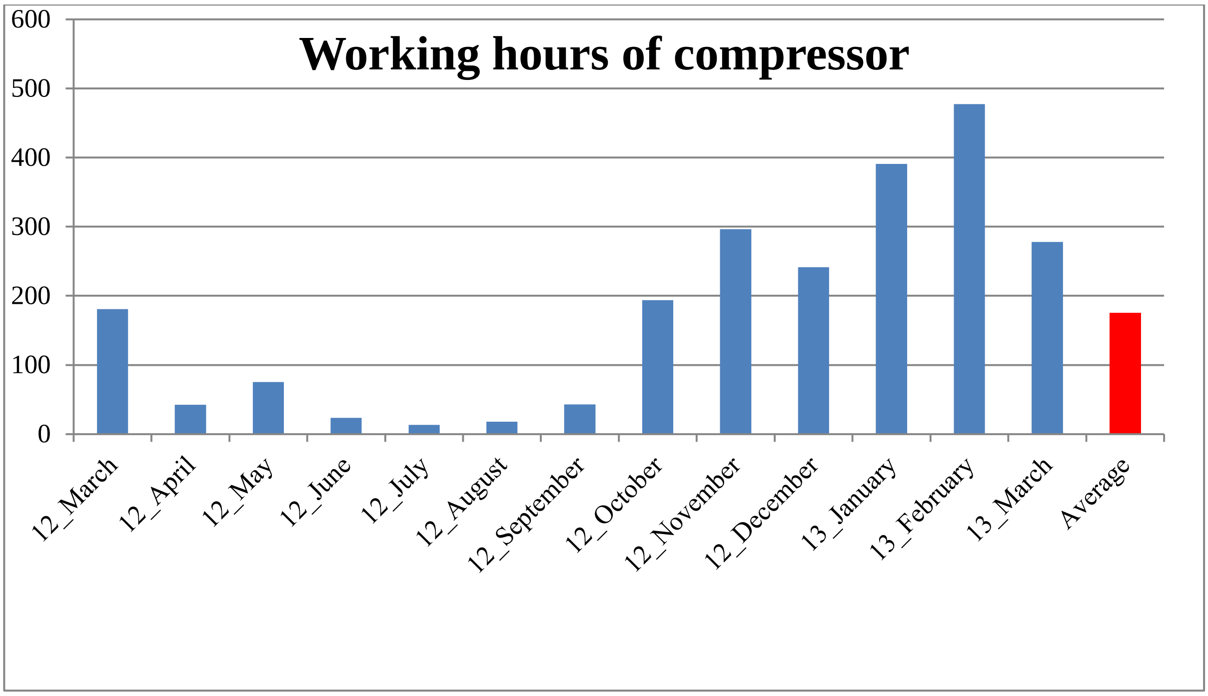


Fig. 4.7.: Number of working hours of compressor in each month

Another important factor is the number of defrosts in each month. This number is connected with temperature of outdoor air, as the moisture in outdoor air freezes in heat exchanger, it lowers a lot the exchangers‘ performance, and thus the unit of heat pump starts the defrost cycle.

The number of defrost is then highest during heating period, on the other hand, during summer period, no or very low number of defrosts was observed.

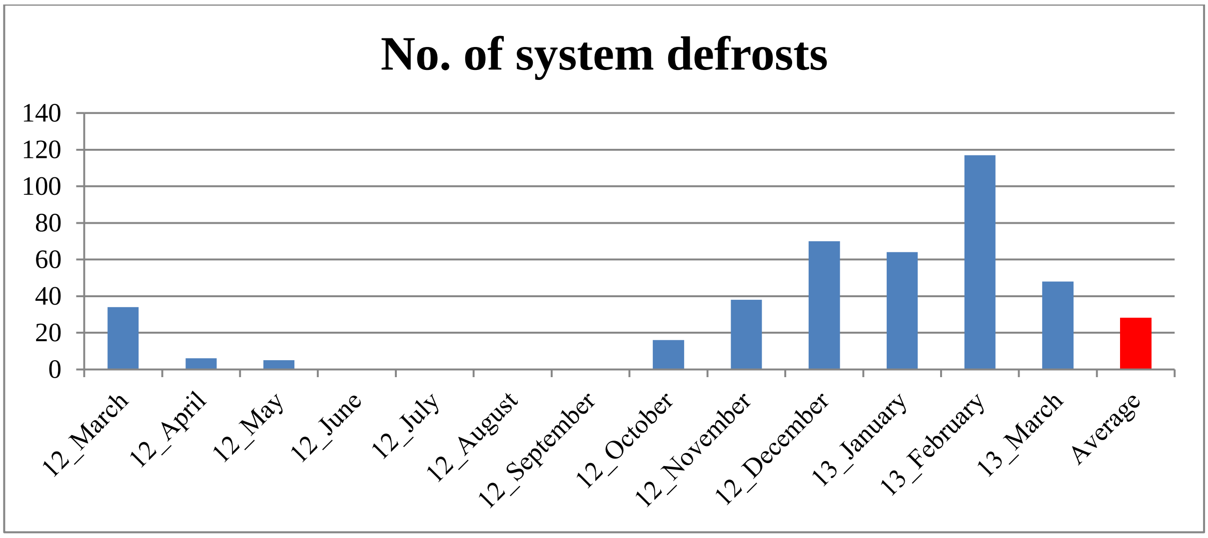
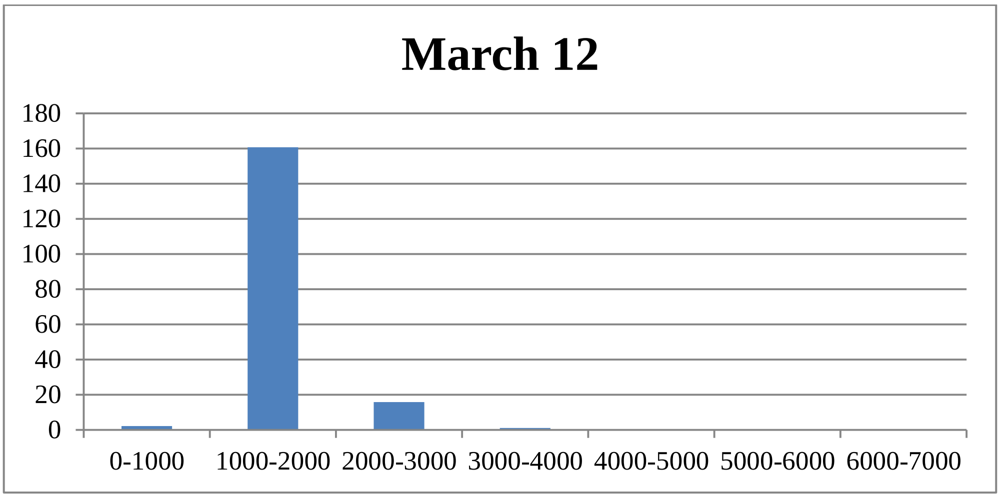
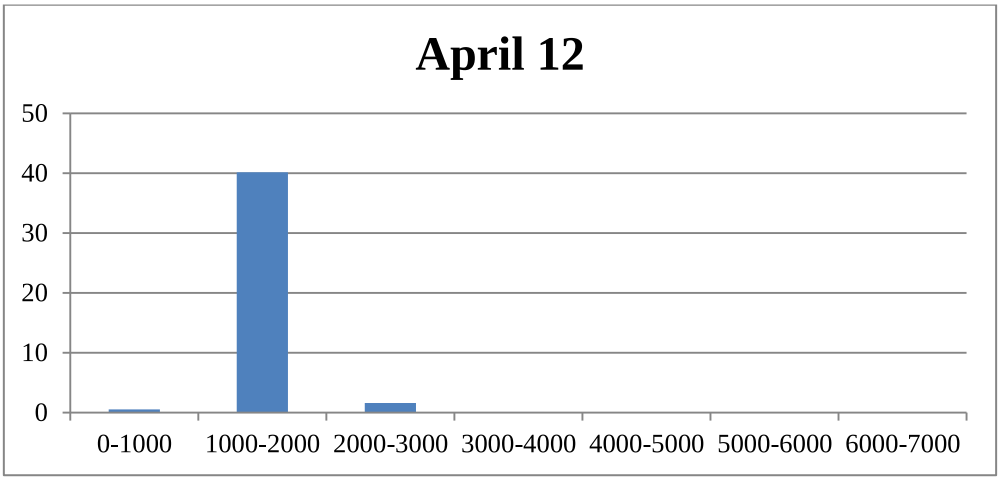
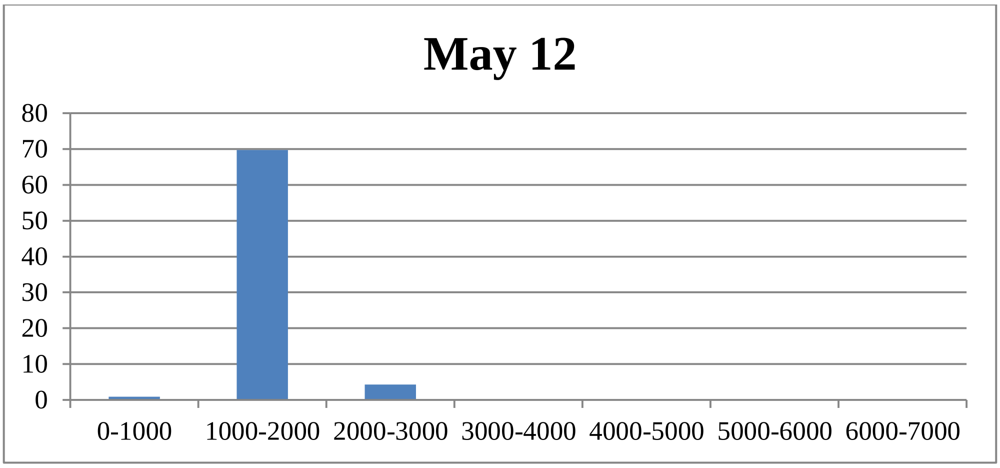


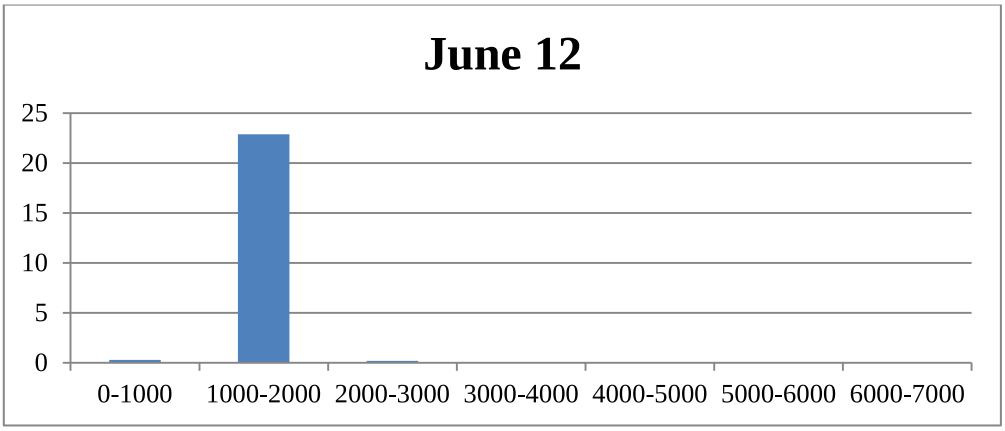
Fig. 4.8.: Number if system defrosts in each month

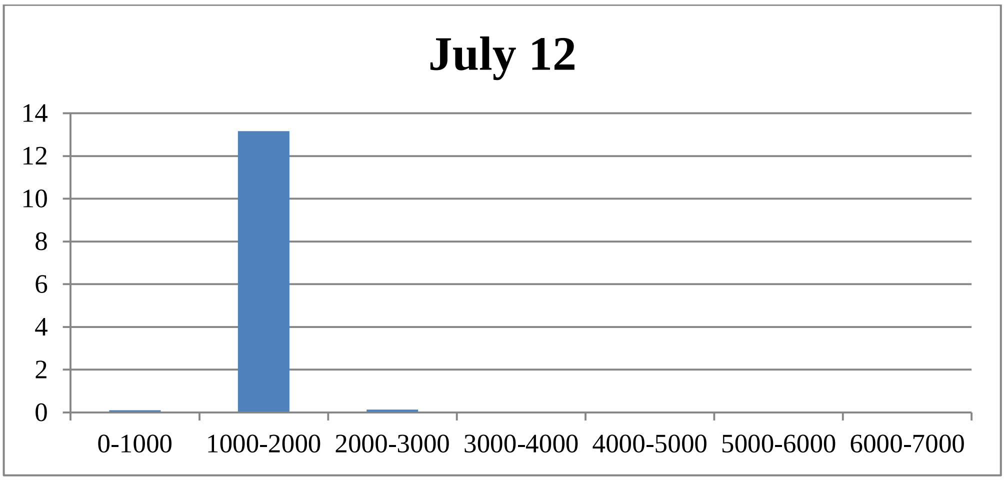
Another very important factor is the graph with compressor speed versus number of working hours spend in each. Evaluated system was equipped with variable speed engine, and further we can see how it was running during each month.

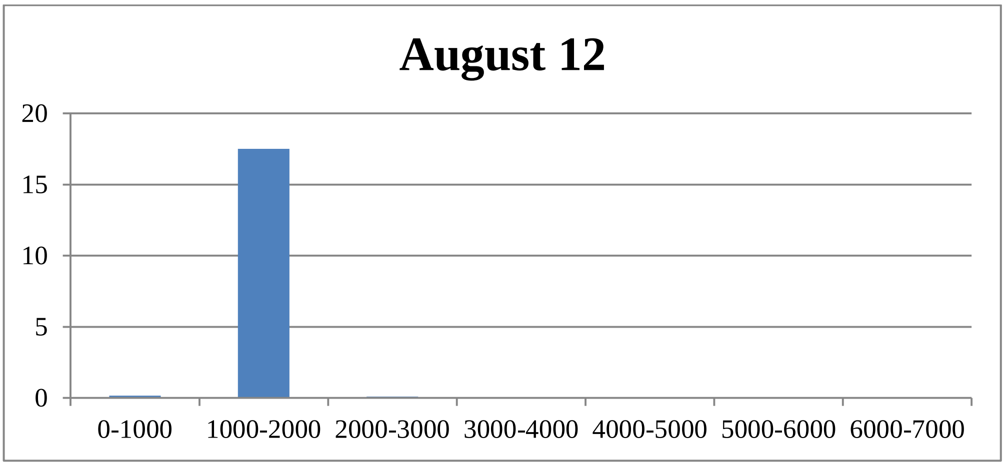


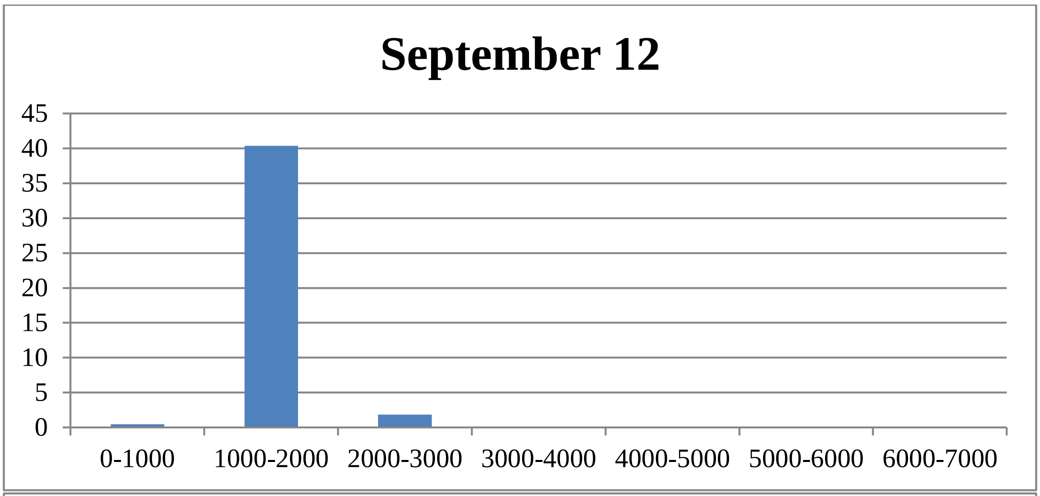


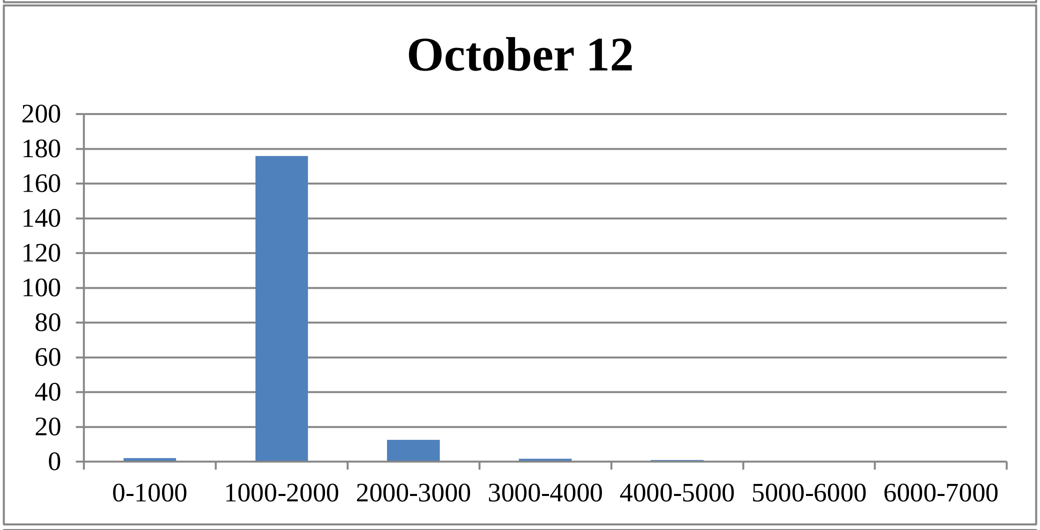


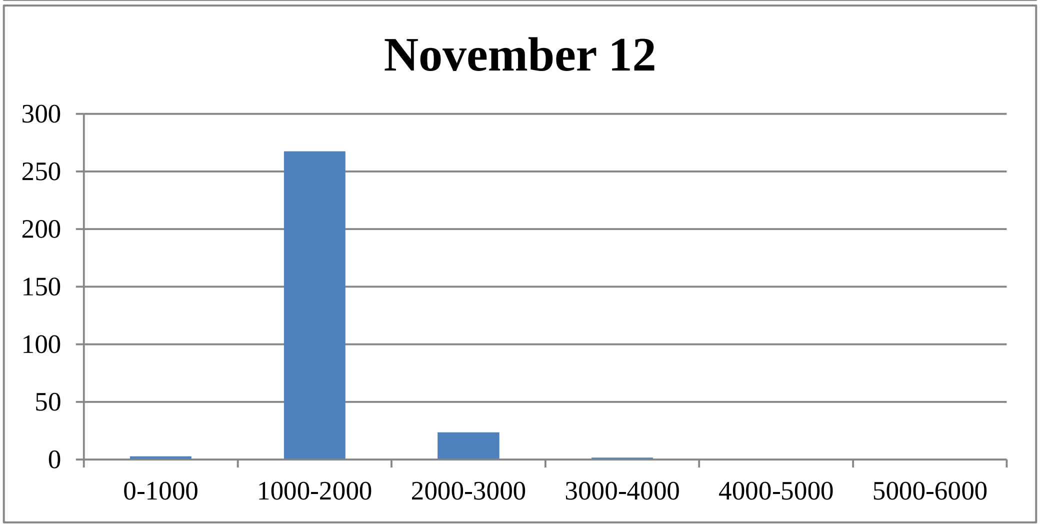

June 12
25
20
15
10
5
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000

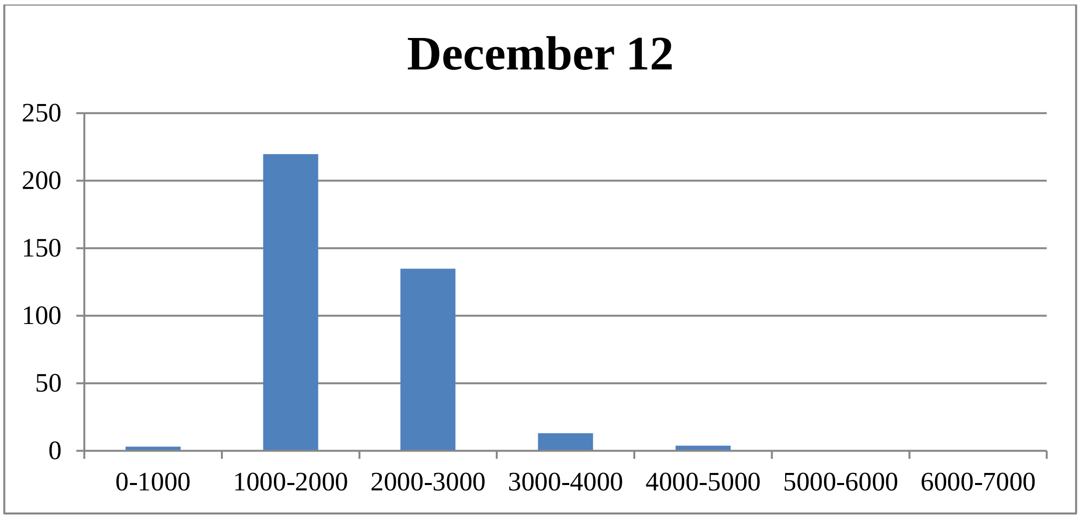

July 12
14
12
10
8
6
4
2
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000

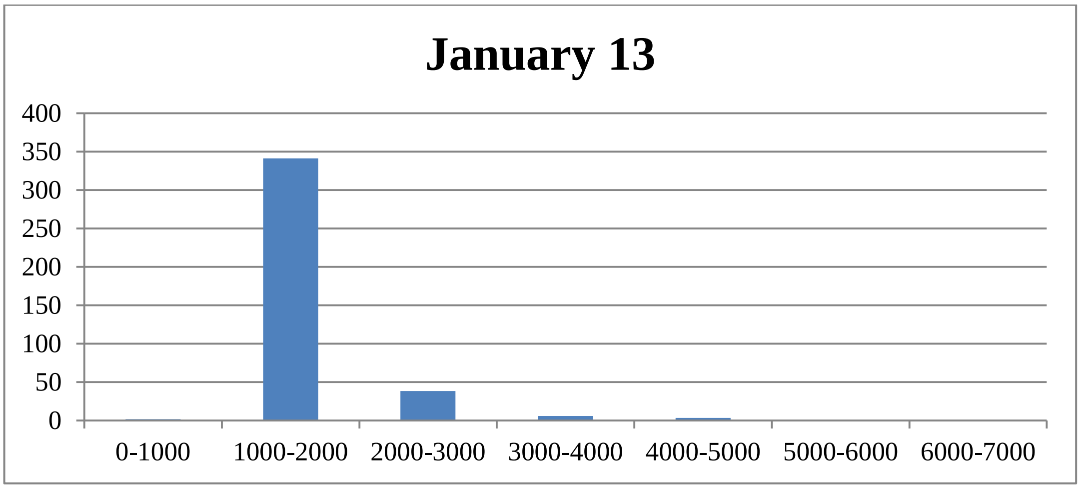

August 12
20
15
10
5
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000

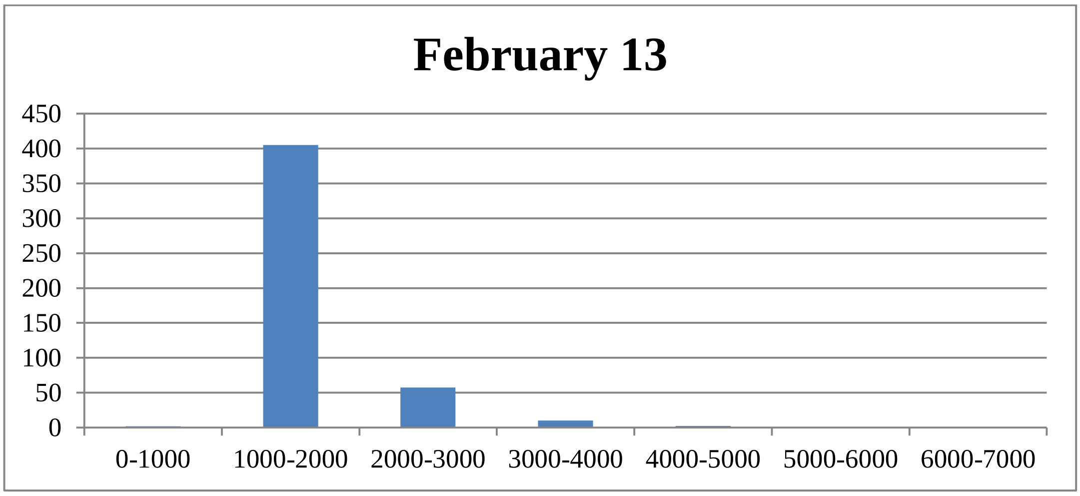

September 12
45
40
35
30
25
20
15
10
5
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000


October 12
200
180
160
140
120
100
80
60
40
20
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000


November 12
300
250
200
150
100
50
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000


December 12
250
200
150
100
50
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000


January 13
400
350
300
250
200
150
100
50
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000


February 13
450
400
350
300
250
200
150
100
50
0
0-1000
1000-2000
2000-3000
3000-4000
4000-5000
5000-6000
6000-7000

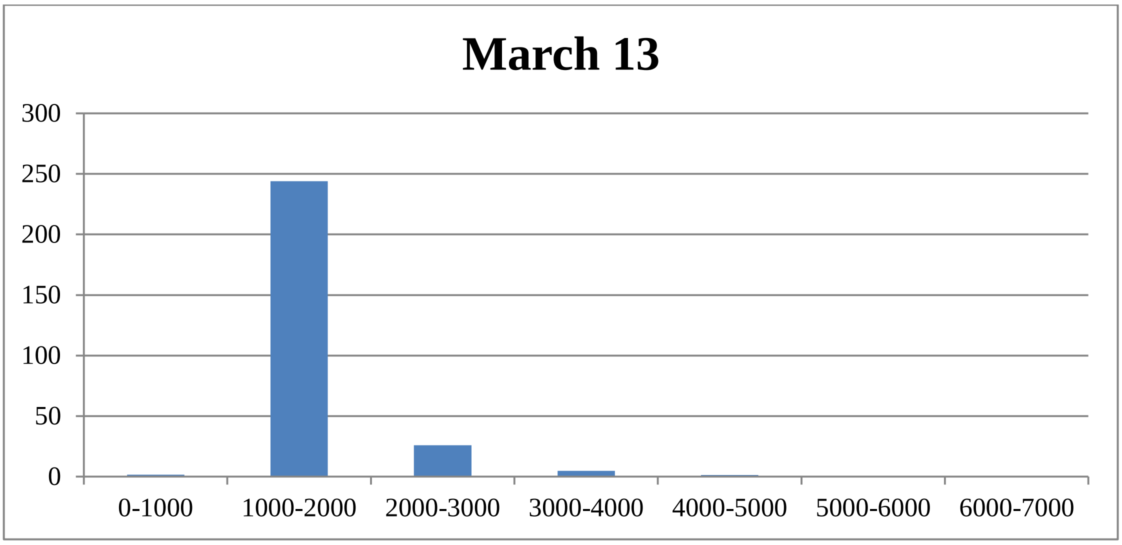


Fig. 4.9 to Fig 4.21: Compressor speed in each month

It was observed, that system never got into compressor speed higher than 5000RPM and generally was reaching the values higher than 3000 RPM only during heating period and even that not very often.

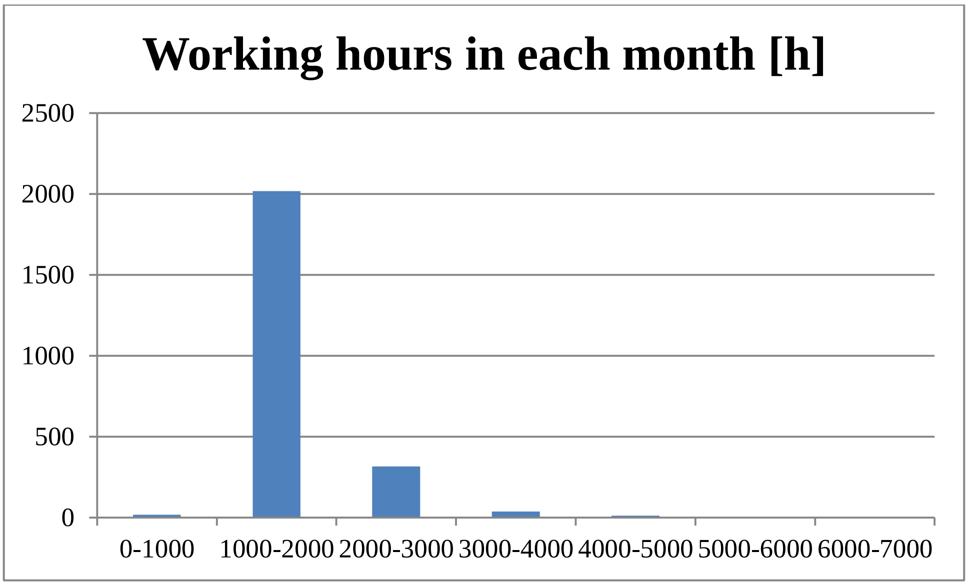


Fig. 4.22.: Number of working hours in each month

In monthly statistics, we can see that 83% of time, compressor was running at 1000-2000 RPM.

Next parameters that were evaluated are performance factors – COP, SCOP and SPF.

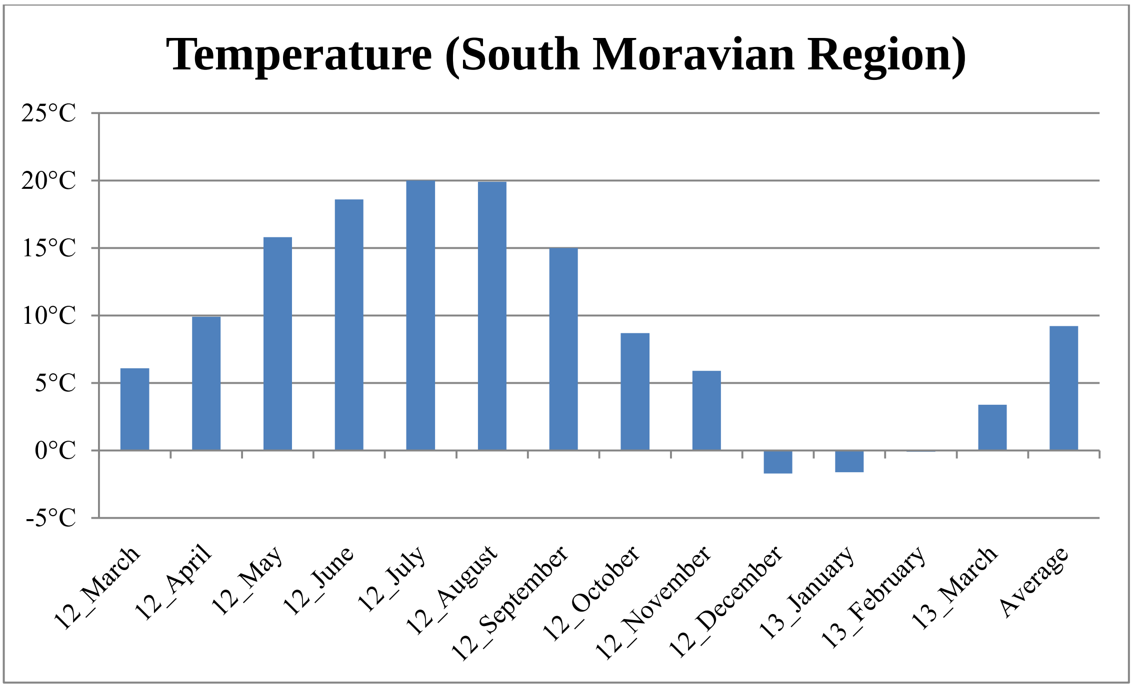


Fig. 4.23.: Average temperature in South Moravian Region

The data from CHMI (Czech Hydrometeorological Institute) about average temperature in each month in region where the tested heat pump is located are crucial for critical evaluation of outcomes that were evaluated.

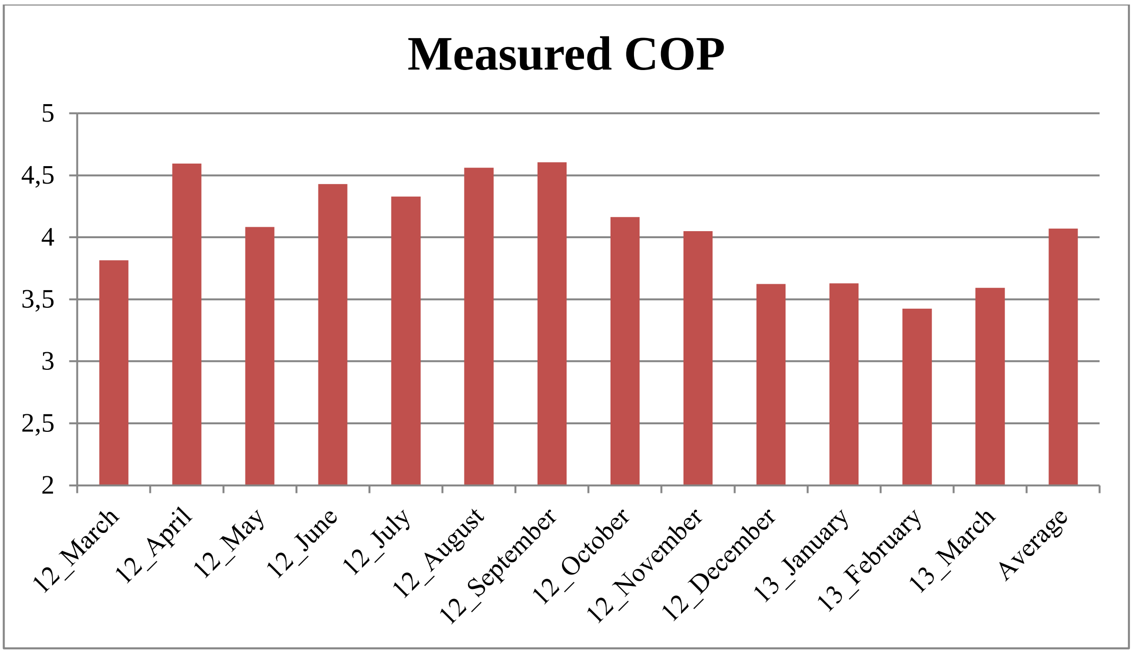


Fig. 4.24.: Measured Coefficient of Performance in each month

We can see what was expected, that COP during heating period is lower than the one during summer period. Also thanks to higher average temperature in March 2012 than the one in

March 13 we can see that also COP of a heat pump during March 2012 is higher than the one in March 2013.

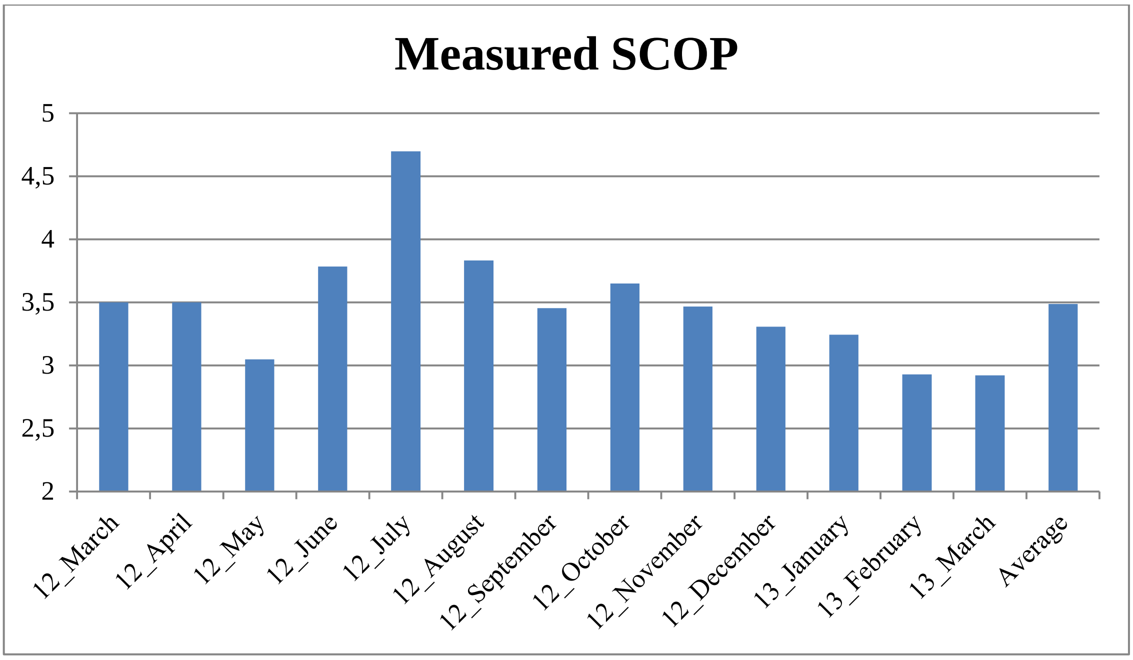


Fig. 4.25.: Measured Seasonal Coefficient of Performance in each month

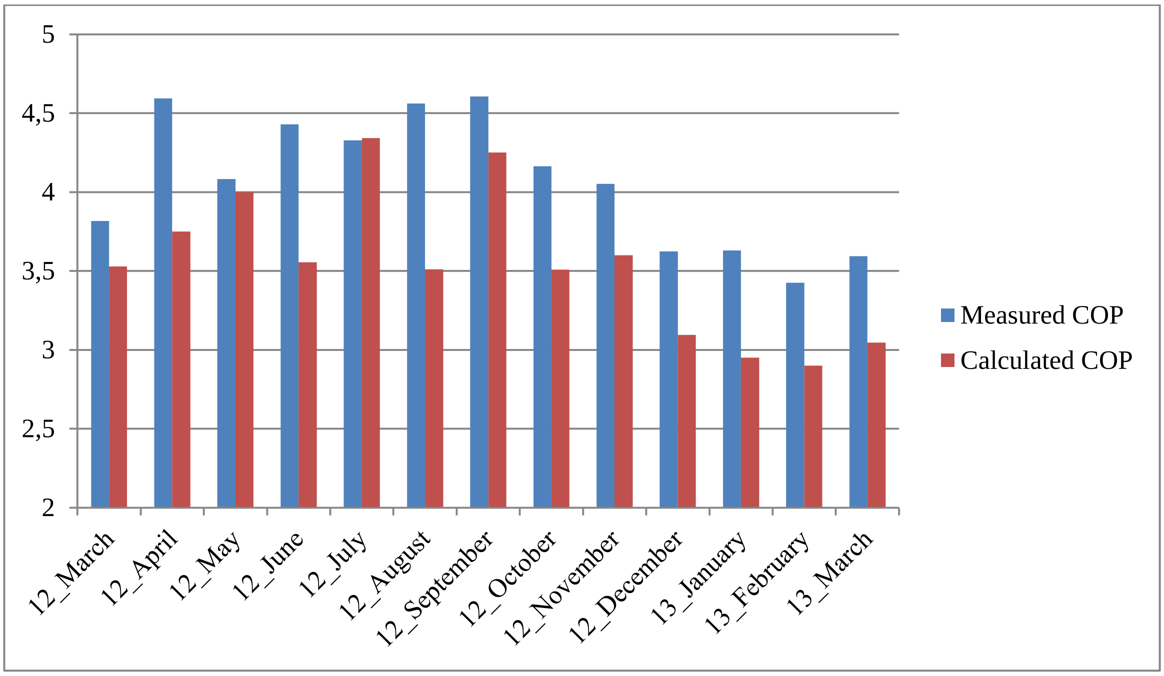


Fig. 4.26.: Difference between measured and calculated Coefficient of Performance in each month

In nearly every month, calculated value of COP differed - was lower than the value the system is logging. This difference is quite hard to explain, because we don't know exactly how the system is calculating this value and what is taken into consideration.

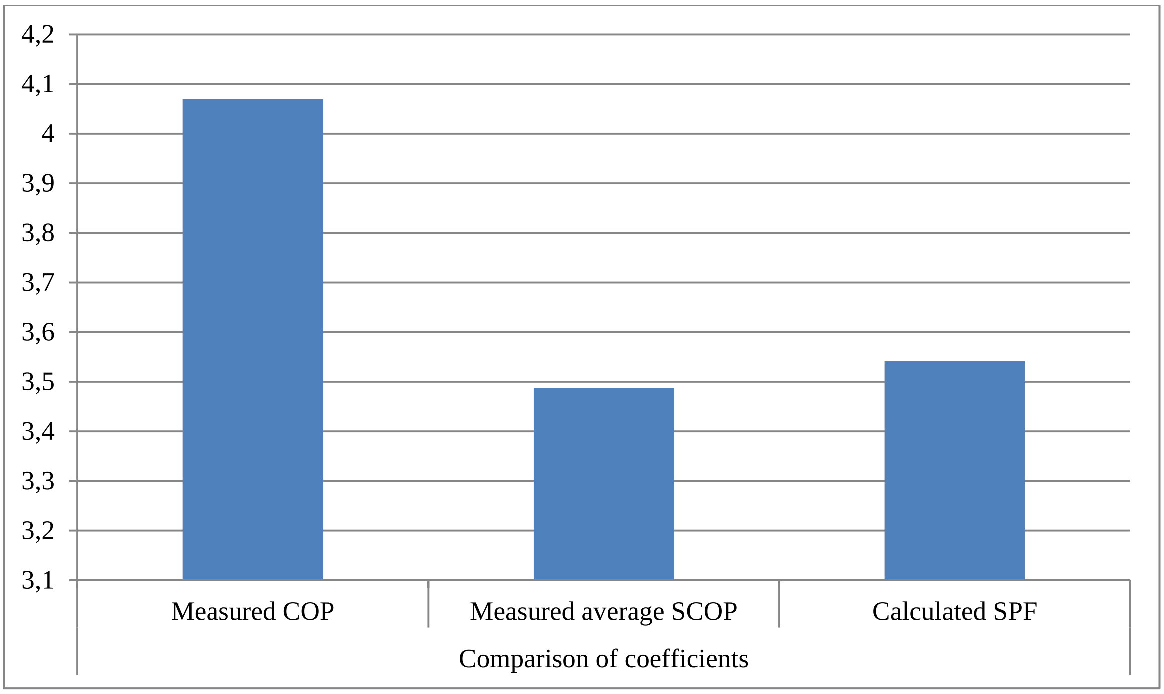


Fig. 4.27.: Difference between measured COP, measured average SCOP and calculated SPF

5 Conclusion

The research of evaluation coefficients showed new possibilities for manufacturers to test their products on different factors. This work concluded evaluation coefficients used in European Union and showes other testing methods in terms of testing performance and ecological characteristics.

The evaluation of given data shows which was expected, the performance of air-to-water heat pump in central european region is worse during heating period, thus saving less money to the owner during the period, when the heating demand is highest. Then the results of data evaluation, presented in graphical form, show the relation between outside temperature and heat pump performance.

Due to the fact, that the author was not able to manage the measurements himself, some issues with data composition were limiting for evaluation, for example the changing amount of data during each month and changing interval of data logging. In such cases, monthly weighted average was used for calculations, knowing that while evaluating of the outcomes, this should be also taken into consideration.

In a nutshell, it was found, that the heat pump is reaching the proclaimed value of Coefficient of Performance and in a yearly conclusion reaching an SPF more than 3,5 which can be considered as a good result.

There were found small differences between measured and calculated COP during run of a heat pump and is up to further discussion with the manufacturer how come these differences occured and what can be done about it.

Bibliography

[1] Thermodynamics. *Wikipedia* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://en.wikipedia.org/wiki/Termodynamics#Laws_of_thermodynamics

[2] Carnot Cycle. *Wikipedia* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://en.wikipedia.org/wiki/Carnot_cycle

[3] LIFE CYCLE CLIMATE PERFORMANCE MODEL FOR RESIDENTIAL HEAT PUMP SYSTEMS. In: ZHANG, Ming, Jan MUEHLBAUER, Vikrant AULE a Reinhard RADERMACHER. *Final Report* [online]. SilVer Spring, USA, October 2011 [cit. 6.5.2013]. Dostupné z: http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/Technical%20Results/AHRTI-Rpt-09003-01.pdf

[4] ARMAND, Paiguy, Ngouateu WOUAGFACK a Réné TCHINDA. *ECOP Criterion for Irreversible Three-Heat-Source Absorption Refrigerators* [online]. InTech, 2012 [cit. 2013-05-06]. Thermodynamics - Fundamentals and Its Application in Science. ISBN 978-953-51-0779-8.

[5] ČSN EN 14825. *Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling - Testing and rating at part load conditions and calculation of seasonal performance*. Praha: Úřad pro technickou normalizaci, metrologii a státní zkušebnictví, 2012.

[6] Absorption Heat Pumps. *ENERGY.GOV* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://energy.gov/energysaver/articles/absorption-heat-pumps

[7] Heat Pump. *Wikipedia* [online]. 2013 [cit. 2013-05-06]. Dostupné z: https://en.wikipedia.org/wiki/Heat_pump

[8] Heat Pump Refrigerant Cycle. *Vent-Axia* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://www.vent-axia.com/heatpump/how

[9] Press Release: Say Hello to Life Cycle Climate Performance. HEATCRAFT REFRIGERATION PRODUCTS. *Heatcraft Worldwide Refrigeration* [online]. 2010 [cit. 2013-05-06]. Dostupné z: http://www.heatcraftrpd.com/news/2010/life-cycle-performance/

[10] EHPA Quality Label. *SP Technical Research Institute of Sweden* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://www.sp.se/en/index/services/heatpump/EHPA/Sidor/default.aspx

[11] Heat pumps in residential and commercial buildings. *Heat Pump Centre* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://www.heatpumpcentre.org/en/aboutheatpumps/heatpumpsinresidential/Sidor/default.aspx

[12] Carnot Cycle. *HyperPhysics - Georgia State University* [online]. 2013 [cit. 2013-05-06]. Dostupné z: 6. http://hyperphysics.phy-astr.gsu.edu/hbase/thermo/carnot.html

[13] Rankine Cycle. *Wikipedia* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://en.wikipedia.org/wiki/File:Rankine_cycle_Ts.png

[14] AIRAH Methods of calculating Total Equivalent Warming Impact (TEWI) 2012. In: *Best Practice Guideline - June 2012* [online]. 2012 [cit. 6.5.2013]. Dostupné z: http://www.airah.org.au/imis15_prod/Content_Files/BestPracticeGuides/Best_Practice_Tewi_June2012.pdf

[15] ČSN EN 14511. *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling*. Praha: Úřadu pro technickou normalizaci, metrologii a státní zkušebnictví, 2012.

[16] Types of heat pumps. *Danfoss* [online]. 2013 [cit. 2013-05-06]. Dostupné z: http://www.heatpumps.danfoss.com/Content/5493D632-9953-42D3-B46C-9C40710A3138_MNU17434663_SIT403.html

[17] ARMAND, Paiguy, Ngouateu WOUAGFACK a Réné TCHINDA. *ECOP Criterion for Irreversible Three-Heat-Source Absorption Refrigerators: Thermodynamics - Fundamentals and Its Application in Science* [online]. 2012 [cit. 2013-05-06]. ISBN 978-953-51-0779-8. Dostupné z: http://www.intechopen.com/books/thermodynamics-fundamentals-and-its-application-in-science/ecop-criterion-for-irreversible-three-heat-source-absorption-refrigerators. InTech. Vedoucí práce Dr. Ricardo Morales-Rodriguez.

[18] KEMNA, René, Martijn VAN ELBURG, William LI a Rob VAN HOLSTEIJN. Technical Anaylsis (incl. System Model). In: *Eco-design of CH Boilers* [online]. Delft, 2007 [cit. 6.5.2013]. Dostupné z: http://www.ecoboiler.org/public/ecoboiler_task4_final.pdf

[19] MIARA, Marek, Danny GŰNTHER, Thomas KRAMER, Thore OLTERSDORF a Jeannette WAPLER. Analysis and Evaluation of Heat Pump Efficiency in Real-Life Conditions. In: *Heat Pump Efficiency* [online]. Freiburg, 2011 [cit. 6.5.2013]. Dostupné z: http://wp-effizienz.ise.fraunhofer.de/download/final_report_wp_effizienz_en.pdf

[20] NORDMAN, Roger, Kajsa ANDERSSON, Monica AXELL a Markus LINDAHL. *Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning* [online]. Borås, 2010 [cit. 2013-05-06]. ISBN ISBN 978-91-86319-86-1. Dostupné z: http://www.sp.se/sv/index/services/heatpump/Documents/SP-RAPP%202010-49.pdf. Bakalářská práce. Vysoké Učení Technické v Brně.

[21] ONISHI, Haruo, Ryuzaburo YAJIMA a Shotaro ITO. *LCCP OF SOME HVAC & R APPLICATIONS In Japan*. 2004. Dostupné z: http://www.epeeglobal.org/epeedocs/internet/docs/20OnishionLCCPofairco_843.pdf

[22] ŠPANIHELOVÁ, Kateřina. *METODIKA ZKOUŠENÍ TEPELNÝCH ČERPADEL* [online]. Brno, 2012 [cit. 2013-05-06]. Dostupné z:

https://dspace.vutbr.cz/bitstream/handle/11012/13244/Metodika%20zkou%C5%A1en%C3%AD%20tepeln%C3%BDch%20%C4%8Derpadel.pdf?sequence=1. Bakalářská práce. Vysoké Učení Technické v Brně. Vedoucí práce Ing. Jiří Hejčík, Ph.D.

[23] UST, Y., B SAHIN a A. KODAL. Ecological coefficient of performance (ECOP) optimization for generalized irreversible Carnot heat engines. *Journal of the Energy Institute* [online]. 2005, č. 78, s. 145-151 [cit. 2013-05-06]. ISSN 1746-0220. Dostupné z: http://www.yarbis.yildiz.edu.tr/web/userPubFiles/yust_0fc750860c7d6a75e986292713128c80.pdf

[24] UST, Yasin. Performance analysis and optimization of irreversible air refrigeration cycles based on ecological coefficient of performance criterion. *Applied Thermal Engineering* [online]. 2009, č. 29 [cit. 2013-05-06]. ISSN 1359-4311. Dostupné z: http://www.sciencedirect.com/science/article/pii/S1359431108000501

[25] UST, Yasin, Bahri SAHIN a Ali KODAL. Performance analysis of an irreversible Brayton heat engine based on ecological coefficient of performance criterion. *International Journal of Thermal Sciences* [online]. 2005, č. 45, s. 94-101 [cit. 2013-05-06]. ISSN 1290-0729. Dostupné z: http://www.yarbis.yildiz.edu.tr/web/userPubFiles/yust_56a31cca8c88ba7544c046b29b7907b8.pdf

[26] ZOTTL, Andreas, Markus LINDAHL, Roger NORDMAN, Philippe RIVIERE a Marek MIARA. *Evaluation method for comparison of heat pump systems with conventional heating systems: Concept for evaluation of CO2-reduction potential*. 31.8.2011. Sweden, 2011, 21 s. Dostupné z: http://www.sepemo.eu/uploads/media/D4_3_Concept_for_evaluation_of_CO2-reduction_potential_Final.pdf

[27] Annual Energy Efficiency Ratio: A Better Measure of Refrigeration System Performance. In: *AEER*[online]. St. Louis, USA, 2007 [cit. 6.5.2013]. Dostupné z: http://www.emersonclimate.com/Documents/6223-White%20Paper-7.pdf

[28] AHRTI LCCP HP 1.0 – Add-In User Guide. In: [online]. [cit. 2013-05-19]. Dostupné z: http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/Technical%20Results/AHRTI%20LCCP%20HP%20-%20Add-In%20User%20Guide%20v1.0.pdf

[29] Global-warming potential. [online]. [cit. 2013-05-19]. Dostupné z: http://en.wikipedia.org/wiki/Global-warming_potential

[30] Standards and methods used by SP for testing of heat pumps. SP TECHNICAL RESEARCH INSTITUTE OF SWEDEN. [online]. [cit. 2013-05-21]. Dostupné z: http://www.sp.se/en/index/services/heatpump/standards_methods_heatpumps/Sidor/default.asp

List of symbols

Quantity	Symbol	Unit
Efficiency	η	-
Temperature	T	°C
Temperature of cold reservoir	T_c	°C
Temperature of hot reservoir	T_H	°C
Entropy	S	J
Heat	Q	J
Mechanical power provided by the system per unit of time	$\dot{W}_{turbine}$	$J.s^{-1}$
Heat energy consumed by the system per unit of time	$\dot{Q}_{in}$	$J.s^{-1}$
Coefficient of Performance	COP	-
Coefficient of Performance at part load conditions	COP_{PL}	-
COP corresponding to the declared capacity (DC) of the unit at the same temperature conditions as for part load conditions A, B, C, D	COP_{DC}	-
Heat produced by a heat pump	Φ	W
Compressor power input	P	W
Seasonal Coefficient of Performance	$SCOP$	-
Seasonal Coefficient of Performance in active mode	$SCOP_{on}$	-
Net Seasonal Coefficient of Performance	$SCOP_{net}$	-
Temperature conditions for each climate	T_{design}	-
Lowest outdoor temperature point at which the heat pump is declared to have a declared capacity	$T_{bivalent}$	°C
Heating or cooling load of the building at T_{design} conditions	P_{design}	W

Operation limit temperature	TOL	°C
Bin hours	h_j	h
Heating energy	Q_h	kW
Cooling energy	Q_c	kW
Reference annual heating demand	Q_h	kWh
Number of hours the unit is considered to work in thermostat off mode	H_{TO}	h
Number of hours the unit is considered to work in standby mode	H_{SB}	h
Number of hours the unit is considered to work in	H_{CK}	h
Number of hours the unit is considered to work in	H_{OFF}	h
Electricity consumption during thermostat off mode	P_{TO}	kW
Electricity consumption during standby mode	P_{SB}	kW
Electricity consumption during crank case heater mode	P_{CK}	kW
Electricity consumption during off mode	P_{OFF}	kW
Number of equivalent heating hours	H_{he}	h
Number of equivalent cooling hours	H_{ce}	h
Bin temperature	T_j	°C
Bin number	j	-
Amount of bins	n	-
Heating demand of the building for the corresponding temperature T_j	$P_h(T_j)$	kW
Cooling demand of the building for the corresponding temperature T_j	$P_c(T_j)$	kW
number of bin hours occurring at the corresponding temperature T_j	h_j	h
COP values of the unit for the corresponding temperature T_j	$COP_{PL}(T_j)$	-
Required capacity of an electric back-up heater for the corresponding temperature T_j	$elbu(T_j)$	kW

Degradation coefficient	C_c	-
Capacity ratio	CR	-
Energy efficiency ratio	EER	-
Energy efficiency ratio at declared capacity	EER_{DC}	-
Energy efficiency at part load conditions	EER_{PL}	-
Seasonal Energy Efficiency Ratio	$SEER$	-
Active mode seasonal energy efficiency ratio	$SEER_{on}$	-
Required capacity of an electric back-up heater for the corresponding temperature T_j	$elbu(T_j)$	kW
Degradation coefficient	C_c	-
Capacity ratio	CR	-
Energy efficiency ratio	EER	-
Energy efficiency ratio at declared capacity Energy efficiency at part load conditions	EER_{DC}	-
Annual efficiency	AE	-
Seasonal Performance Factor	SPF	-
Primary energy factor	f_p	$\left[\frac{kWh}{kWh}\right]$-
Global Warming Potential of refrigerant	GWP	$kg\ CO_2e/kg$
Total Equivalent Warming Impact	$TEWI$	$\frac{kg\ CO_2}{lifetime}$
CO_2e – emission coefficient	K	$kg\ \frac{CO_2e}{kWh}$-
Refrigerant content of the heat pump	m	kg
Refrigerant losses of refrigerant per year	L	$\frac{kg\ refrigerant}{year}$
Operation time of the system in years	n	year
Recovery factor during the disposal of the system after life time	$\alpha_{Recovery}$	-
Annual consumption of electrical energy	E_{annual}	$\frac{kWh}{year}$
Percentage of loss when reclaim	R	[%]
Mass of materials	m_m	kg

Ecological Coefficient of Performance	$ECOP$	-
Global Warming Potential of refrigerant	GWP	$kg\ CO_2e/kg$
Total Equivalent Warming Impact	$TEWI$	$\frac{kg\ CO_2}{lifetime}$
Power output	W	$W.s^{-1}$
Entropy generation rate	S_g	$J.s^{-1}$
Heating energy produced by heat pump	$Q_{heat,HP}$	kW
DHW energy produced by heat pump	$Q_{DHW,HP}$	kW
Energy produced by electric back-up heater	$Q_{heat\|DHW,back-up}$	kW
Energy consumed by compressor and control unit	$W_{comp+cont}$	kW
Energy consumed by brine pump, fans or well pump	$W_{BP\|Fan\|WP}$	kW
Energy consumed by charge pumps	$W_{back-up}$	kW