CURRENT WORK WITHIN IEA HPP ANNEX 34 ON PERFORMANCE EVALUATION AND TESTING METHODS FOR THERMALLY DRIVEN HEAT PUMPS FOR HEATING AND COOLING

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Abstract: Due to a substantial growth of the market for thermally driven heat pumps (TDHP), a need for standardised performance evaluation became evident among all stakeholders. Currently, the available standards do not cover all applications of this technology. Therefore, International Energy Agency (IEA) initiated through its Heat Pump Programme (HPP) a project – Annex 34 "Thermally Driven Heat Pumps for Heating and Cooling" – in which, among other topics, the basis for a standardised testing and performance evaluation of TDHP units and systems should be elaborated. This paper gives an overview of the proposed concepts for further discussion on standardised testing and monitoring procedures, as well as subsequent performance evaluation of TDHP units and systems.

Key Words: IEA, Annex 34, thermally driven heat pumps, performance, testing

1 INTRODUCTION

The market for thermally driven heat pumps used for heating and/or cooling, primarily for residence and small to medium commercial applications, experienced a substantial growth in the recent years. Hence, the interest both from the industry and the policy makers for standardised test and performance evaluation procedures for the technology has grown. As one of the consequences, the Heat Pump Programme of the International Energy Agency started a project – Annex 34 – which addresses this among other TDHP-relevant topics.

As a part of the project, a comprehensive research on available national and international standards was carried out. The research gave an overview of the methods currently used for the evaluation of heat pump and chiller performance, both electrically and thermally driven. The standards were reviewed with a focus on testing, rating and performance evaluation procedures. Standards on safety, sound power, design etc. were not evaluated since not considered relevant for this project.

From the analysis of the collected standards, the following conclusions regarding the need for an improvement in the field of standardised performance evaluation of the TDHP units and systems could be made:

- Different definitions of performance figures and system boundaries used;
- None of the standards is fully suitable for the evaluation of the annual performance of TDHP or systems including TDHP;
- No definition of steady-state conditions the test standards analysed is applicable for cycling or batch mode operation;
- A unified method for a reliable comparison of different TDHP among each other and with other technologies is currently missing.

2 SYSTEM CLASSIFICATION

From an analysis of configurations and operation modes of TDHP systems, a reference system was developed, Figure 1. The nomenclature and the graphical display are based on the work in IEA SHC Task 38.

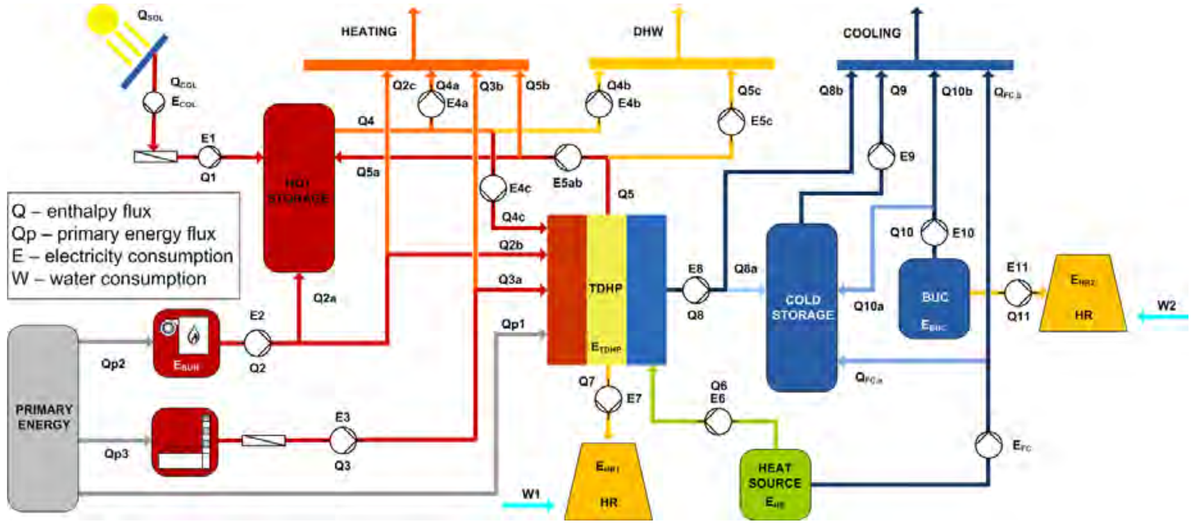

Figure 1: Reference TDHP system

System configurations – or generic system layouts – for different applications or modes of operation can be obtained from the reference system by deleting parts of it which are not used in the respective case. An example for a direct fired combined heating and cooling system is shown in Figure 4.

For TDHP systems a classification according to the type of the driving energy input and the type of the energy output was adopted, Figure 2. This classification yields six system configurations, each containing the most common features of the respective system, for which the performance figures are proposed. For the calculation of the performance figures of real systems, energy flows physically not existing have to be set to zero. If the system includes additional components not considered, the energy flows and consumptions can be easily added.

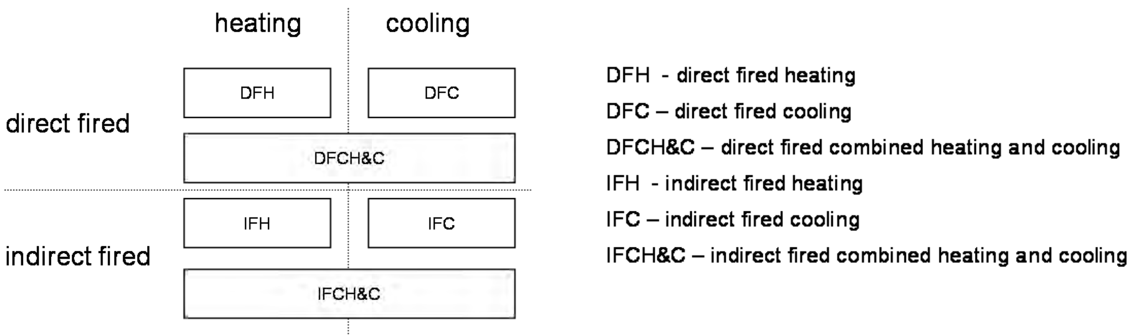

Figure 2: Classification of TDHP systems

3 DEFINITION OF PERFORMANCE FIGURES

In Annex 34, performance figures for both the heat pump unit itself, as well as for overall systems were proposed.

3.1 System boundaries and performance figures

3.1.1 General concept

The analysis of the available standards for heat pumps showed that generally three system boundaries and four performance figures are commonly used. In Figure 3, system boundaries for a reference TDHP system with the respective performance figures are shown.

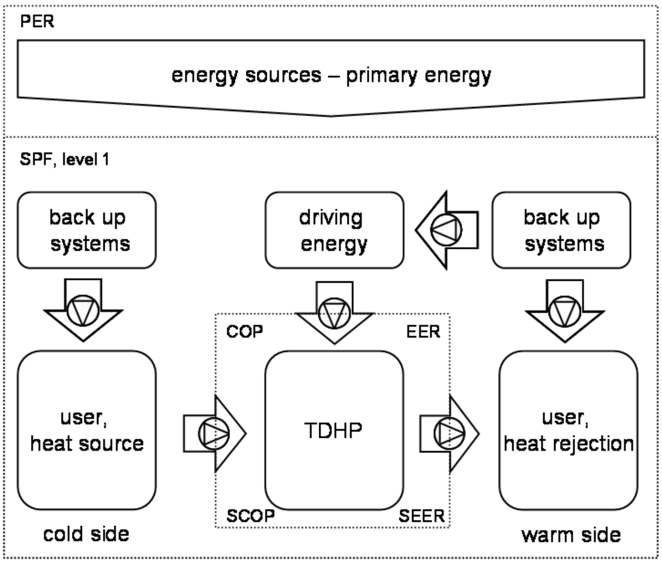

Figure 3: TDHP system boundaries for the definition of the performance figures

The system boundary for the coefficient of performance (COP) and the seasonal COP (SCOP) for heating as well as the energy efficiency ratio (EER) and the seasonal EER (SEER) for cooling are drawn around the TDHP unit. These figures take into account only the energy inputs and outputs of the heat pump itself, which are obtained from the measurements on a test rig under specified, mostly steady state operating conditions. Following the practice of CEN 2009, the energy inputs and outputs are corrected for the energy consumption of the liquid pump and its influence on the heat transfer media, respectively.

The seasonal performance factor (SPF), at its highest level (as defined in chapter 3.1.4), takes into account the total system energy outputs and inputs of all system components. The SPF, however, does not take into account the source of the consumed energy, i.e. electricity or driving heat. Finally, the primary energy ratio (PER) takes into account the type of the energy source, an important figure for the calculation of the energy costs or the emissions of the system, for example. For the SPF and the PER, the energy inputs and outputs are obtained from the monitoring results.

Unlike the electrically driven heat pumps (EDHP), the TDHP consume both electricity and thermal energy. By defining separate thermal and electrical performance figures, additional information on the unit or the system behaviour is provided.

3.1.2 COP and EER

As described in the previous chapter, the COP and the EER are obtained from the laboratory measurements under defined, steady state operation. These figures represent the efficiency of the unit at its nominal capacity under specified operating conditions. The corrections for the liquid pumps are defined in analogy to CEN 2009.

For heating:

$$COP_{th} = \frac{\dot{Q}_H \pm \dot{Q}_{lp}}{\dot{Q}_{in,unit}} \quad (1) \qquad COP_{el} = \frac{\dot{Q}_H \pm \dot{Q}_{lp}}{P_{in,unit} \pm \sum_i P_{lp,i}} \quad (2)$$

For cooling:

$$EER_{th} = \frac{\dot{Q}_C \pm \dot{Q}_{lp}}{\dot{Q}_{in,unit}} \quad (3) \qquad EER_{el} = \frac{\dot{Q}_C \pm \dot{Q}_{lp}}{P_{in,unit} \mp \sum_i P_{lp,i}} \quad (4)$$

As in CEN 2009, the upper sign (addition or subtraction) in the equations 1 through 4 applies for the case when the liquid pump is integrated into the heat pump unit, the lower sign for the case when it is not.

3.1.3 SCOP and SEER

The SCOP and the SEER represent the calculated seasonal efficiencies for a reference application and a reference climate, based on measurements on a test rig under steady state conditions at different loads. Unlike the COP and the EER, they represent the efficiency of the unit for time dependent conditions (e.g. oscillating source temperature, changing supply temperature etc.) for a certain period of time, usually a year.

Annex 34 adopted the temperature bin method already used in a number of standards (e.g. CEN 2008 and CEN 2010). The method is based on the division of the cumulative annual frequency of the outside dry-bulb temperature into temperature classes called bins. For every bin, an average operating condition is defined. It is then assumed, that the heat pump unit operates under this condition for the entire temperature range covered by the bin. A full description of the method, as well as some open questions can be found e.g. in Wemhöner and Afjei 2003 and CEN 2008.

The SCOP and the SEER are determined from equations 5 -8.

$$SCOP_{th} = \frac{(Q_H + Q_{DHW})}{\frac{Q_H}{SCOP_{on,th}} + \frac{Q_{DHW}}{PF_{DHW,on,th}} + Q_{aux}} \quad (5) \qquad SEER_{th} = \frac{Q_C}{\frac{Q_C}{SEER_{on,th}} + Q_{aux}} \quad (6)$$

$$SCOP_{el} = \frac{(Q_H + Q_{DHW})}{\frac{Q_H}{SCOP_{H,on,el}} + \frac{Q_{DHW}}{PF_{DHW,on,el}} + E_{aux}} \quad (7) \qquad SEER_{el} = \frac{Q_C}{\frac{Q_C}{SEER_{on,el}} + E_{aux}} \quad (8)$$

$SCOP_{on}$ and $SEER_{on}$ represent the efficiencies in the active heating or cooling mode, respectively. In this mode, useful energy output is delivered to the user. PF_{DHW} represents the efficiency of the unit while producing DHW. Q_{aux} and E_{aux} represent the auxiliary energy inputs like controls or the energy consumption in the stand-by mode.

$SCOP_{on}$ and $SEER_{on}$ are determined from equations 9-12. The total energy output, given in the numerator, is determined from the sum of the heating demands in every temperature bin. This figure includes energy inputs both from the TDHP and the back-up units. The terms in the denominators represent the energy input to the unit, including all auxiliaries as well as the energy consumption of the reference back-up system.

$$SCOP_{on,th} = \frac{\sum_{i=1}^{n} h_i \cdot \dot{Q}_H(T_i)}{\sum_{i=1}^{n} \left(\frac{\dot{Q}_{H,TDHP}(T_i)}{COP_{th}(T_i)} \right)} \quad (9) \qquad SEER_{on,th} = \frac{\sum_{i=1}^{n} h_i \cdot \dot{Q}_C(T_i)}{\sum_{i=1}^{n} \left(\frac{\dot{Q}_{C,TDHP}(T_i)}{EER_{th}(T_i)} \right)} \quad (10)$$

$$SCOP_{on,el} = \frac{\sum_{i=1}^{n} h_i \cdot \dot{Q}_H(T_i)}{\sum_{i=1}^{n} \left(\frac{\dot{Q}_{H,TDHP}(T_i)}{COP_{el}(T_i)} \right)} \quad (11) \qquad SEER_{on,el} = \frac{\sum_{i=1}^{n} h_i \cdot \dot{Q}_C(T_i)}{\sum_{i=1}^{n} \left(\frac{\dot{Q}_{C,TDHP}(T_i)}{EER_{el}(T_i)} \right)} \quad (12)$$

3.1.4 SPF, PER and f_{sav}

SPF is the ratio of the total useful energy output to the total energy input of a system. Like the COP, EER, SCOP and SEER, one thermal and one electrical SPF are defined for TDHP systems, equations 13 and 14. Both figures are meant to be determined from in-situ measurements of all required variables. The drawback of this definition is that systems with different shares of heating and cooling energy are hard to compare due to there different machine COPs and different exergy values of the delivered useful energies. Therefore this is part of further discussions. Beside the overall SPF, particular heating, cooling or DHW performance factors can be defined.

$$SPF_{th} = \frac{\int \left(\dot{Q}_H + \dot{Q}_{DHW} + \dot{Q}_C\right) \cdot dt}{\int \dot{Q}_{th,in,system} \cdot dt} \quad (13) \qquad SPF_{el} = \frac{\int \left(\dot{Q}_H + \dot{Q}_{DHW} + \dot{Q}_C\right) \cdot dt}{\int \dot{Q}_{el,in,system} \cdot dt} \quad (14)$$

For the definition proposal regarding seasonal performance factors, the concepts developed in IEA SHC Task 38 (Sparber et al 2008) and IEE SEPEMO-BUILD project (SEPEMO 2010) were used and adapted to the TDHP system. Three levels and four sublevels were defined which differ in complexity - the number of measured variables, required measurement equipment and thus the detailedness of the provided information about the system and its behaviour. An example for a DFCH&C system is given in the next chapter.

1. Level: At this level, the monitoring data allow basic information on operation cost and performance for the overall system. Depending on the application and system configuration, a relatively small number of measurement instruments are needed. Here, a primary energy ratio can be defined, equation (15).

$$PER = \frac{Q_H + Q_{DHW} + Q_C}{\sum_{j=1}^{m} \left(\frac{Q_{th,in,j}}{\varepsilon_j} + \frac{Q_{el,in,j}}{\varepsilon_{el}} + \frac{W}{\varepsilon_W} \right)} \quad (15)$$

2. Level: This level allows a more detailed insight into the system itself. For a better comparison of systems and subsystems, in Figure 4 four different boundaries are defined (1-4), which also define the SPF Levels 2a, 2b, 2c and 2d.

<u>Level 2a</u>: The system boundaries correspond roughly to the boundaries for the calculation of the COP and the EER. However, the correction due to the liquid pumps is not taken into account, since in most cases it is not feasible to measure the pressure drop of the secondary heat transfer medium in situ. SPFa is obtained.

<u>Level 2b</u>: The system includes the TDHP with the source (for heating applications) and the heat rejection systems (for cooling application). At this level, the SPF_b gives the possibility to compare the unit with all its necessary parts with other systems (heat sources and heat sinks) and technologies (e.g. gas boiler).

<u>Level 2c</u>: The system is extended to the storage tanks. At this level, the storage efficiency can be calculated. The seasonal performance factor is denominated with SPF_c.

<u>Level 2d</u>: The system includes all systems, excluding the distribution systems for heating, DHW and cooling. SPF_d can be used to compare entire systems among each other.

3. Level: The system performance is compared to a reference system in terms of the primary energy consumption based on the method of "fractional energy saving", equation 16. With f_{sav}, primary energy savings compared to a reference system can be quantified. A description

of this concept is given in Letz 2002. As for the PER, the primary energy conversion factors ε are dependent on the installation site, type of fuel used, its quality etc. Nominal values provided in the literature can be used for the comparison of the technologies. For the calculation of the site-related, specific values, real site-related ε values have to be used.

$$f_{sav,DFCH\&C,2} = 1 - \frac{\dfrac{Q_p}{\varepsilon_{fossil}} + \dfrac{Q_{th,in}}{\varepsilon_{fossil} \cdot \eta_{conv}} + \dfrac{E_{el}}{\varepsilon_{el}} + \dfrac{W}{\varepsilon_W}}{\dfrac{Q_{p,ref}}{\varepsilon_{fossil,ref}} + \dfrac{Q_{th,in,ref}}{\varepsilon_{fossil,ref} \cdot \eta_{conv,ref}} + \dfrac{E_{el,ref}}{\varepsilon_{el}} + \dfrac{W_{ref}}{\varepsilon_W}} \qquad (16)$$

3.2 Example: Direct fired combined heating and cooling (DFCH&C)

The concept of the SPF, PER and f_{sav} described in chapter 3.1.4 will be explained on a generic system for combined heating and cooling with a direct fired TDHP, Figure 4. It can be applied both for simultaneous heating and cooling, as well as for alternate operation.

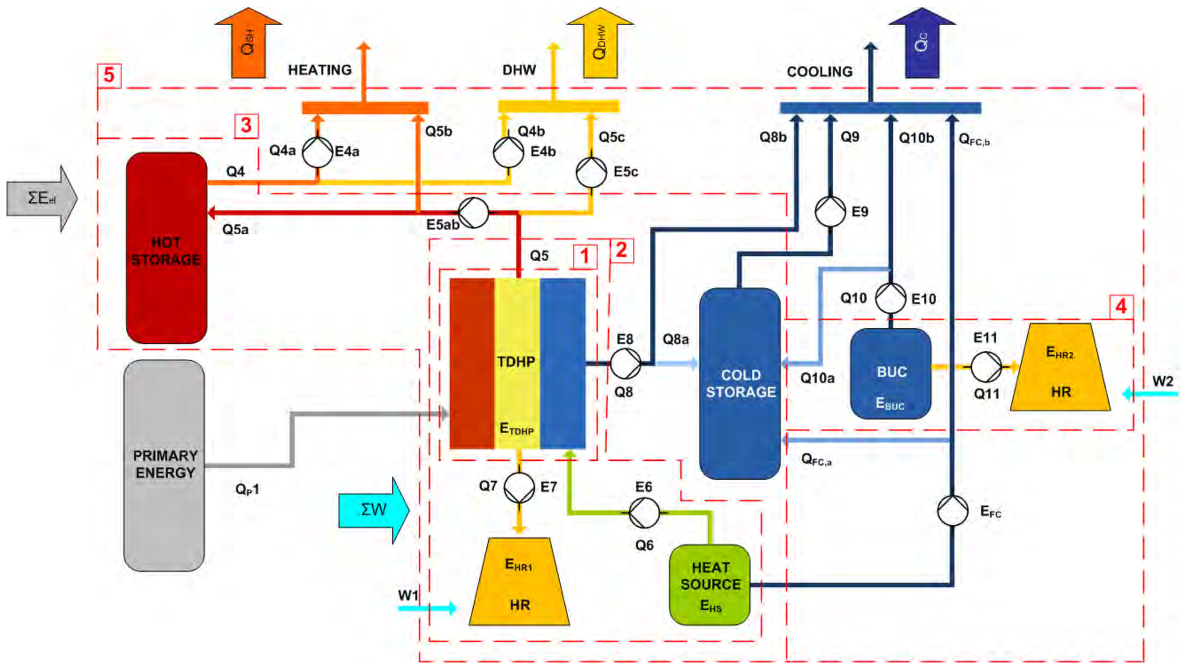

Figure 4: System configuration and boundaries for direct fired combined heating and cooling TDHP system

3.2.1 SPF Level 1

For the Level 1 SPF, the system boundary 5, Figure 4 is used. The energy output consists of overall heating, DHW and cooling energy supplied to the user. The thermal energy input is the primary energy, e.g. gas supplied to the TDHP. The electricity consumption includes the consumption of all system components. The thermal and the electrical SPF and the PER are determined from equations 17-19:

$$SPF_{th}^{DFCH\&C} = \frac{Q_{SH} + Q_{DHW} + Q_C}{Qp1} \qquad (17)$$

$$SPF_{el}^{DFCH\&C} = \frac{Q_{SH} + Q_{DHW} + Q_C}{\sum E_{el}} \qquad (18)$$

$$PER_{DFCH\&C} = \frac{Q_{SH} + Q_{DHW} + Q_C}{\frac{Qp1}{\varepsilon_{fossil}} + \frac{\sum E_{el}}{\varepsilon_{el}} + \frac{\sum W}{\varepsilon_W}} \tag{19}$$

3.2.2 SPF Level 2

Level 2a: TDHP performance

The system boundary 1 from Figure 4 is used.

$$SPF_{th,a}^{DFCH\&C} = \frac{Q5 + Q8}{Qp1} \tag{20}$$

$$SPF_{el,a}^{DFCH\&C} = \frac{Q5 + Q8}{E_{TDHP}} \tag{21}$$

Level 2b: TDHP performance including heat source and heat sink

Boundary 2 in Figure 4 applies:

$$SPF_{th,b}^{DFCH\&C} = \frac{Q5 + Q8}{Qp1} \tag{22}$$

$$SPF_{el,b}^{DFCH\&C} = \frac{Q5 + Q8}{E_{TDPH} + E6 + E7 + E_{HS} + E_{HR1}} \tag{23}$$

Level 2c: TDHP performance including heat source, heat sink and storages

For DFCH&C, level 2c differs from level 2b only in the definition of the useful energy. Instead of *Q5* and *Q8*, *Q4* and *Q9* are considered, respectively. The SPF_c takes into account the storage losses. Boundary 3 in Figure 4 applies:

$$SPF_{th,c}^{DFCH\&C} = \frac{Q4 + Q5b + Q5c + Q8b + Q9}{Qp1} \tag{24}$$

$$SPF_{el,c}^{DFCH\&C} = \frac{Q4 + Q5b + Q5c + Q8b + Q9}{E_{TDPH} + E5ab + E6 + E7 + E8 + E_{HS} + E_{HR1}} \tag{25}$$

Level 2d: System performance without circulation pumps on the user side

Boundary 3 and boundary 4 in Figure 4 apply:

$$SPF_{th,d}^{DFCH\&C} = \frac{Q_{SH} + Q_{DHW} + Q_C}{Qp1} \tag{26}$$

$$SPF_{el,d}^{DFCH\&C} = \frac{Q_{SH} + Q_{DHW} + Q_C}{\sum E_{el} - (E4a + E4b + E5b + E5c + E8b + E9 + E10b + E_{FC})} \tag{27}$$

E5b and E10b represent the electricity consumption E5ab and E10 form the Figure 4 if both direct feeding into the heating and cooling distribution systems as well as feeding from the storages are possible.

3.2.3 SPF Level 3

As a reference system for combined heating and cooling, a combination of a chiller and a fossil fuel boiler is assumed, Figure 5. The two systems deliver energy outputs without any interaction between them.

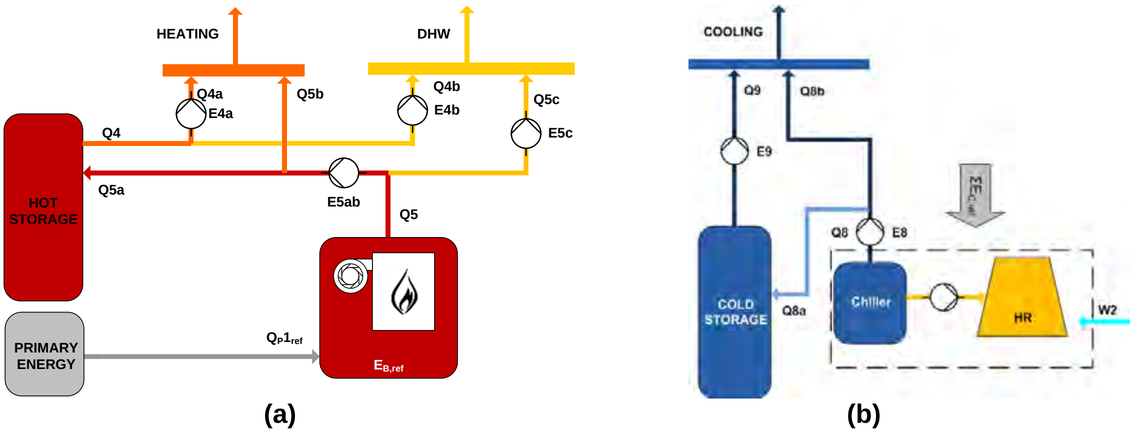

Figure 5: Reference systems for heating (a) and cooling (b)

The fractional energy saving for the reference system in Figure 5 is:

$$f_{sav,DFCH\&C} = 1 - \frac{\dfrac{Qp1}{\varepsilon_{fossil}} + \dfrac{\sum E_{el}}{\varepsilon_{el}} + \dfrac{\sum W}{\varepsilon_W}}{\dfrac{Qp1_{ref}}{\varepsilon_{fossil,ref}} + \dfrac{\sum E_{el,ref}}{\varepsilon_{el}} + \dfrac{W2_{ref}}{\varepsilon_W}} \tag{28}$$

The primary energy consumption of the reference heat production system $Qp1_{ref}$ and the total electricity consumption of the reference system can be calculated as:

$$\sum E_{el,ref} = \sum E_{el} - E_{TDHP} - E6 - E7 - \sum E_{C,ref} + E_{B,ref} + \frac{Q_C}{SPF_{C,ref}} \tag{29}$$

$$Qp1_{ref} = \frac{Q_H + Q_{DHW}}{\eta_{B,ref}} \tag{30}$$

4 TEST METHOD FOR BATCH MODE OPERATION TDHP

In order to calculate the performance figures introduced in chapters 3.1.2 and 3.1.3, laboratory measurements under specific rating conditions should be carried out. So far, the procedures for testing thermally driven heat pumps were only defined for continuous machines. However many products currently present on the market operate non-continuous. The measurement procedure should also be valid for those machines. Therefore, a methodology for testing continuous and discontinuous heat pumps at steady-state conditions has been studied within the project.

The first part of the proposed procedure concerns a new classification of the existing thermally driven heat pumps based on their working modes:

Continuous Heat Pumps (*or Chillers*) are those in which the four phases are processed continuously and by dedicated components within the machine.
Semi-Continuous Heat Pumps (*or Chillers*) are those in which the four phases are periodically shifted among the internal components (i.e. the heat exchangers) producing a discontinuous operation. The shifts, here are called “swaps”. At least two sorption/desorption devices are required for the semi-continuous operation.
Batch Mode Heat Pumps (*or Chillers*) are those in which the four phases are processed by couples and periodically shifted among the internal components like for semi-continuous heat pumps. This type of heat pump, has usually a huge amount of storage capacity embedded (usually sorbent): the couples of phases, desorption/condensation and sorption/evaporation, can be processed one at time (asynchrony). At least one module is needed.

A full sorption cycle covers the four phases: desorption, isosteric cooling, adsorption and isosteric heating). The sorption cycle of a two module heat pump consists of two half cycles.

Due to the periodical shift of the internal functions, the last two types of chillers are characterised by a cyclic operation. Representative quantities are therefore cycle time, swaps and recovery period, i.e. the time, after a swap, necessary to establish the previous working conditions

A crucial point of the proposed procedure regards the definition of steady-state conditions. The existing standards define them as ranges within which the fluctuations of the quantities under control have to lie for a certain period of time. Such conditions are inapplicable in case of discontinuous heat pumps due to their cyclic nature. Volume flow rates and outlet temperatures (see Figure 6-a) are changing rapidly during the swap. Therefore, new steady-state conditions, which take into account the heat pump’s working mode, have been set up. They make use of two subintervals – i.e. A and B – which differ in the nature of the controlled quantities’ fluctuations. In detail:

Sub-Interval A consists of all data collected during each cycle time (i.e. the time between two consecutive swaps) whose fluctuations are due both to the machine behavior and to the measuring apparatus.

Sub-Interval B consists of the data collected during each cycle time with the exception of those related to the recovery period. It contains only the data corresponding to steady-state inlet conditions to the machine; here, measured temperatures’ variations in time are due to the machine behavior (mainly for outlet temperatures) or to the measuring apparatus (fluctuations of the inlet temperatures, mass flows and pressures).

In Figure 6-b, the subintervals are represented on a discontinuous heat pump since it represents the most general case. Obviously all remarks done for discontinuous machines can be extended to the continuous ones, which can be thought as a machine characterized by a unique cycle, for which the recovery time is equal to zero and the two subintervals coincide.

In Table 1, the steady-state conditions for discontinuous heat pumps are listed on the two subintervals mentioned above. They have been fixed for the: **Cycle time**, **Inlet temperatures**, **Static Pressure Differences** and **Volume Flows**. The tolerances used refer to values suitable for some machines tested. Further tests shall be carried out on other commercial systems to provide general values. Concerning the condition given for the inlet temperatures and mass flows on sub-Interval A, an additional explanation has to be given. Since the thermal COP computed is a function of the heat supplied or removed from the heat pump along the whole cycle (sub-Interval A), the inlet temperatures as well as the mass flows have to be brought back to their set values after every swap fast enough in order to avoid that the measure is largely altered by the measuring system.

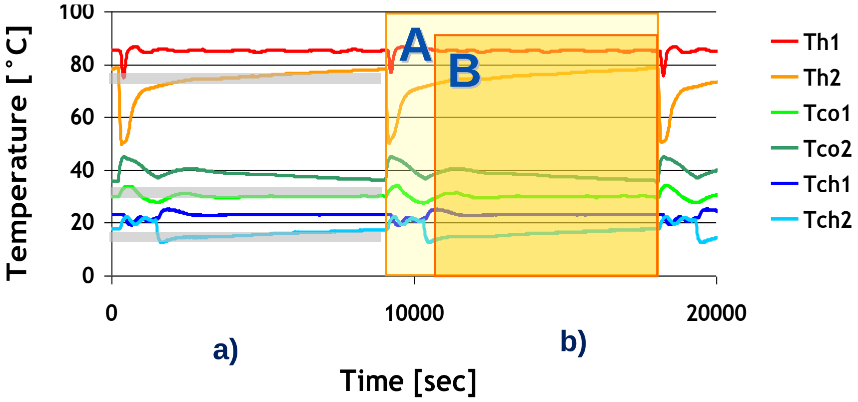

Figure 6: Temperature profiles at generator (h), condenser (co) and evaporator (ch) of a discontinuous chiller vs. time. a) Ranges used by existing standard for the steady-state condition; b) Sub-Intervals employed in the new steady-state conditions definitions

To formalise this concept it was established that, in sub-Interval A, the ratio of the temperature differences and mass flows for each circuit shall be equal or higher than 0.95. For the temperature differences, this condition can be formulated as follows:

$$R = \frac{\overline{T_{in,real} - T_{out,real}}}{T_{in,set} - \overline{T_{out,real}}} \geq 0.95 \tag{31}$$

At this point, it is possible to state that the steady-state conditions are obtained and maintained when all the monitored quantities remain constant for a minimum duration of four half cycles, with respect of tolerances given in Table 1. In this way, each component of the machine performs at least two full cycles. Moreover, four pre-conditioning half cycles shall be carried out before test to slake the influence of the former working condition on the present.

In case continuous heat pumps are tested, the steady-state conditions are obtained and maintained when all the monitored quantities remain constant for a minimum duration of one hour, with respect of the tolerances given in subinterval B of Table 1.

Once the steady-state conditions are achieved, the test can be performed. In order to acquire a dataset representative of the machine operation, the test shall last:

- 0.5 hour in case of continuous chiller,
- 4 half cycles (two full cycles) in case of discontinuous chiller.

The sample time shall be maximum 10 seconds, in order to capture the smallest meaningful data. The choice of 10 seconds is based on the swapping times of the smallest chillers available on the market.

For assessing chiller's performances, mean values shall be calculated. Here, as an example, the mean thermal power, expressed in kW, is reported. It can be obtained as the average of all instantaneous thermal powers calculated for each cycle and for each sample carried out during the test.

$$\overline{\dot{Q}} = \frac{1}{n_{total,sample}} \cdot \sum_{j=1}^{n^{\circ}_{cycle}} \cdot \sum_{i=1}^{n^{\circ}_{sample/cycle,j}} \cdot \left(\dot{m}_{i,j} \cdot c_{p,i,j} \cdot \Delta T_{i,j}\right) \tag{32}$$

Table 1: Steady-state boundary conditions for a discontinuous chiller

Measure Quantity		Permissible deviation of the arithmetic mean values from set values		Permissible deviation of individual measured values from set values	
		A	B	A	B
Generator	Inlet Temperature	R>95%	±0.2K	R>95%	±0.5K
	Mass Flow	R>95%	±2%	R>95%	±5%
	Static Pressure Diff.		-		±10%
Condenser/ Evaporator	Inlet Temperature	R>95%	±0.2K	R>95%	±0.5K
	Mass Flow	R>95%	±2%	R>95%	±5%
	Static Pressure Diff.		-		±10%
Cycle Time		±10%			

For the calculation of COP (or EER) the equations introduced in chapter 3.1.2 are used.

The proposed procedure can be applied partially for the assessment of the seasonal COP (SCOP) or seasonal EER (SEER), since, once the machine is at part load, the performance is calculated at steady-state conditions. In this case, the main obstacle to overcome is the definition of a general procedure to achieve the part load, currently under evaluation. For the calculation of SCOP (or SEER) the equation introduced in chapter 3.1.3 are used.

5 CONCLUSIONS AND FURTHER WORK

The rising market of thermally driven heat pumps generates a need for standards for their performance evaluation. A comprehensive analysis of the current standardisation documents on testing and performance evaluation showed that there is a number of different definitions of performance figures and corresponding system boundaries. Starting from the standards' review and other relevant publications, a set of performance figures for the evaluation of TDHP units and systems, based on a systematic approach, has been proposed. The figures are easy adaptable for any system configuration and appropriate for comparisons with other heating, cooling and DHW systems. The proposed methods will be tested for consistency and applicability on a number of units and systems tested and monitored within Annex 34 and are open for further discussion.

Further, the descriptions of testing procedures in available standards were found not to be applicable for discontinuous TDHP technologies. Based on an analysis of the measurement data of a TDHP unit operating in batch mode, a new definition of steady-state conditions, which can be applied to all TDHP technologies, is proposed. The boundary conditions for the tests on a specific prototype were elaborated. Further tests on other machines should yield generally applicable values.

Nomenclature

E	electrical energy, in kWh
f_{sav}	fractional energy saving
h	temperature bin hours
P	electrical power, in kW
Q	heat energy, in kWh
$\dot{Q}$	heat power in kW
T	Temperature, in K
W	mass flow rate of water, in kg/s
ε	conversion factor
η	efficiency

Subscripts

aux	auxiliary
bu	back-up
BUC	back-up chiller
C	Cooling
conv	conversion
CS	cold storage
DHW	domestic hot water
el	electrical
H	heating
HS	hot storage
lp	liquid pump
ref	reference
th	thermal
W	water

6 ACKNOWLEDGEMENTS

The authors would like to thank all participants of IEA HPP Annex 34 who contributed to the elaboration of the procedure. The financial support for the participation of the authors in the IEA HPP Annex 34 was granted by:

Austria: Federal Ministry of Transportation, Innovation and Technology supported by the Austrian Research Promotion Agency – FFG

Germany: Federal ministry of Economics and Technology (grant number FKZ 0327427B), supervised by Projekträger Jülich PTJ

Italy: Stiftung Südtiroler Sparkasse

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