

PERFORMANCE AUDITS OF HEAT PUMPS— PROCEDURES AND UNCERTAINTIES

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ABSTRACT

Heat pumps are used for climate control of buildings and processes. Experience indicates that their real-world efficiency often is considerably below the theoretical potential. This has been noted by the European Union and hence the Directive on Energy Performance of Buildings includes the status of air-conditioning and heating systems.

Performance audits can detect deviations and indicate ways of upgrading the performance. A survey indicates a shortage of proven, cost-effective methods but a Nordic work has resulted in two in situ test methods. One is based on traditional *external* measurements of energy inputs and outputs. The other relies on *internal* measurements of temperatures and pressures inside the refrigerant system. Both methods include requirements on accuracy and stability and methods to compare stated performance with measured data.

The *internal* measuring principle has been validated by means of laboratory measurements and sensitivity analysis. Deviations between the *internal* and *external* methods were typically less than 5 %. One particular advantage of the *internal* principle is the possibility to disclose a deviation from expected performance and simultaneously provide information regarding the cause.

Key words: *audit, commissioning, field test, heat pump, in situ, method, performance*

1 INTRODUCTION

Air-conditioning equipments based on heat pumping technologies use a large and growing part of the world's electricity supply. They also play an ever more important role in providing a suitable climate for people, products and processes. Furthermore, the environmental effects of air-conditioning equipment have been in the focus of public concern. Typically, effects related to the electric drive power supply dominate over other effects such as refrigerant emissions etc. (Heikkilä and Fahlén 2003). Hence the efficiency, operation and availability are of great concern for society. One indicator of this importance is the new European Directive on the Energy Performance of Buildings that specifically addresses the operation of air-conditioning equipment with a capacity exceeding 12 kW.

Field investigations of heat pumps and refrigeration equipment generally show that performance is well below the possibilities of a good installation. Hence there are plenty of ECOs (Energy Conservation Opportunities) in existing as well as future plants. However, in order to assess the potential in a specific installation the current status of the equipment must be determined. Even though the ECOs provide possibilities of substantial economic benefits, the performance assessment is rarely made for various reasons. One important reason has been the lack of suitable procedures for this type of audit, in particular regarding small and medium sized equipment. The emergence of CEN and ISO standards has not altered this situation since these standards only concern laboratory testing. Another reason is that in most applications a requirement on indoor climate is the driving force for commissioning (e.g. comfort cooling in office buildings, storage of chilled and frozen food etc.). Hence, the primary task is to ascertain a compliance with requirements on indoor climate and equipment efficiency is a secondary issue. Heat pumps used in heating applications, on the other hand, have as their basic motivation an improved heating

economy. Therefore the cost, and hence also the efficiency of operation, become fundamental issues and it is only natural that most of the work on performance auditing has been carried out in this application.

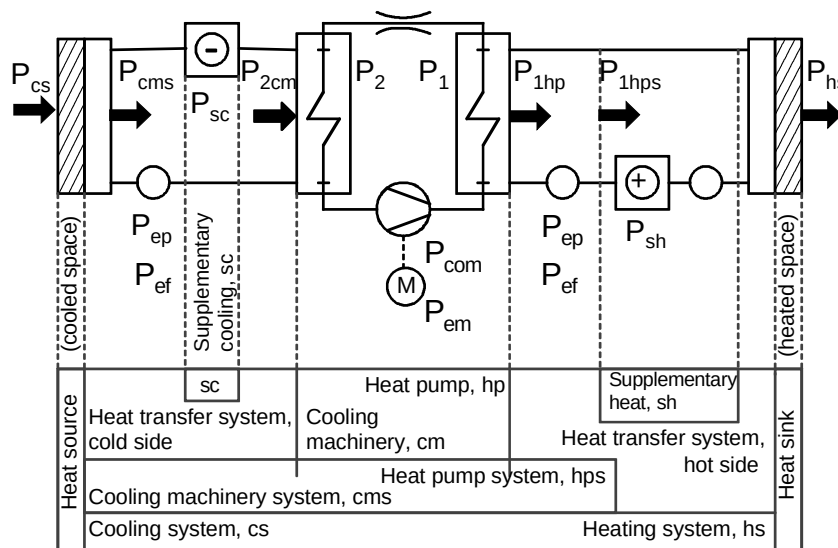
Obviously, to ascertain the proper functioning of installed air-conditioning equipment, there is a need for suitable methods for commissioning and performance checking. A Nordic project on this topic has resulted in two in situ methods. One is based on *external* measurements of the energy inputs and outputs of the equipment. The other is based on *internal* measurements of temperatures and pressures inside the refrigeration system. The Nordic work comprises: 1. *General conditions regarding field testing and presentation of performance* (NT VVS 115 1997); 2. *Check-ups and performance data inferred from measurements in the refrigerant system* (NT VVS 116 1997); and 3. *Large heat pumps - Field testing and presentation of performance* (NT VVS 076 1989). The principles of the methods have been used extensively, NT VVS 076 since 1982, and they can be used for both heating and cooling applications.

2 COMMISSIONING AND PERFORMANCE AUDITS

Commissioning is a process to verify that a technical system performs according to specified functional requirements. To avoid disagreement between client and contractor after the commissioning, the functions and where they apply, as well as the permissible deviations and the consequences of excessive deviations, must all be well specified. Therefore, relevant *system boundaries* must always be defined and the same boundaries used consistently in the specifications as well as during the evaluation. Commissioning provides information for important decisions. Hence, in the *comparison* of specified and measured performance at the specified system boundary, the *uncertainty* of performance data must always be adapted to the consequences of deviations. In case of a heat pump, important functional requirements may concern *operating range*, *capacity range* (part load operation in case of capacity control), heating (P_1) or cooling (P_2) *capacity*, and heating (COP_1) or cooling (COP_2) *coefficient of performance* at full load and, possibly, part load. Use of recognized standards simplifies the life of contracting parties and substantially reduces the bulk of paper work to specify contractual prerequisites.

2.1 Prerequisites: System Boundaries and Stability of Operation

System boundaries define points of delivery of functions in commissioning tasks. Figure 1 provides an example from the Nordtest standards (NT VVS 115 1997).



The original Swedish indices have been replaced in the diagram.

Fig. 1. System boundaries based on NT VVS 076, 115 and 116.

Obviously, measuring points in performance audits should harmonize with the relevant system boundaries. When this is not practical, corrections must be made. It must be clear if components such as pumps or fans and losses from pipes or ducts are included or not in the delivery. For instance, if the performance of a heat pump system (c.f. Fig. 1) has been guaranteed but thermometer wells are only available according to Fig. 2, then the heating contribution from the pump must be considered. The thermometer wells to the right of the pump cannot be used because they are not in the same flow circuit as the flow meter is. Using incompatible measuring positions can be the cause of large errors.

Irrespective of the type of performance audit, be it laboratory testing or on-site evaluation, there must be sufficient stability of operation to give a relevant result. In this context, one often considers more or less cyclic undulations but the most difficult case during on-site commissioning tests is probably a systematic temperature glide. In this case there will be no averaging effect and a systematic deviation will follow. Another situation is when a unit is operating on-off at part load. Then there is always a charging or discharging of energy in the system. To minimize the effect, it is advisable to measure at positions that provide the smallest possible thermal mass within the system boundary. One can estimate the transient effects as indicated by Eq. 1-Eq 4. Using designations according to Fig. 2 we have:

$$P_1 = M_1 \cdot c_{p1} \cdot \frac{\partial t_1}{\partial \tau} + P_{1hp} \quad [\text{Eq. 1}]$$

(storage of heat in the condenser)

$$P_{1hp} + \xi \cdot P_{ep} = M_{hw1} \cdot c_{phw} \cdot \frac{\partial t_{hw}}{\partial \tau} + P_{1hps} \quad [\text{Eq. 2}]$$

(storage of heat in the heating water system and heating input from the heating water pump)

$$P_{1hps} + \eta_{sh} \cdot P_{sh} = M_{hw2} \cdot c_{phw} \cdot \frac{\partial t_{hw}}{\partial \tau} + P_{hs} \quad [\text{Eq. 3}]$$

(storage of heat in the heating water system and additional heat from the supplementary heating system)

$$P_{hs} + \xi \cdot P_{e,p,rh} = M_{rh} \cdot c_{p,rh} \cdot \frac{\partial t_{rh}}{\partial \tau} + P_{rh} \quad [\text{Eq. 4}]$$

(storage of heat in the room heating system and input from the heating water pump)

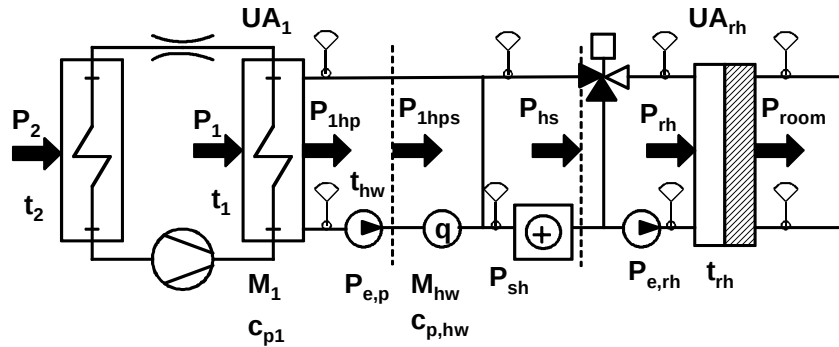


Fig. 2. An example of system boundaries and points of measurement (Fahlén 2001).

Thus, dynamic errors, such as those caused by a temperature glide, are related to the product of thermal mass and rate of temperature change. Partial loads result in short operating times and large temporal temperature derivatives. Measurements directly at the heat pump connections result in the smallest included thermal mass. Even so, the dynamic error may be considerable. On a small residential unit, for instance, the rate of temperature change may be of the order of 0.1 K/s (6 K/min) during autumn and spring. An estimate of the thermal mass of the condenser and its ancillary components provides an expected error due to the storage effect of the order -5 to -10 %.

Naturally, cyclic variations will also cause errors and uncertainties. The undulations yield an uncertainty in calculated mean values. The magnitude of this uncertainty will depend on the amplitude and frequency of the variation as well as on the measuring frequency and the dynamic characteristics of the sensors. The rise times of sensors will put a limit to the shortest measuring interval that is meaningful to use. If the sensors are slow they will provide apparently accurate mean values. However, since they do not deliver any relevant information regarding the actual variations, the real uncertainty may be severely underestimated. Furthermore, calculated quantities such as thermal power, $P = \rho \cdot c_p \cdot q \cdot \Delta t$, may have an error even if the individual mean values are correct. The product of two mean values is not necessarily the same as the mean value of their products. It is better to have sensors with short response times and calculate derived quantities for each set of measurements. In this way the uncertainty will also be less. The worst situation occurs when the frequency of variation coincides with the measuring frequency.

2.2 Performance Audits and Related Uncertainties

Nordtest provides field measuring methods suitable for commissioning of heat pumps and refrigerating plants. Irrespective of the measuring principle, however, it is absolutely vital to use a transparent and recognized method of estimating uncertainties of individual measurands, how they propagate to derived results and the chosen confidence level. This can be presented by means of uncertainty budgets (Fahlén 1994). Without a specified method, uncertainty figures will be ambiguous. For instance, an uncertainty of 5 % at a confidence level of 95 % (approx. 2 standard deviations) may yield a more reliable result than 2.5 % at a level of 60 % (approx. 1 standard deviation). Also, the overall uncertainty of a derived result will appear to be much larger if a worst-case approach is taken for the combined uncertainty than if the probable value is estimated. Measurements are important tools in decision-making. Hence, one should allocate resources to ascertain their quality in relation to the consequences of their use leading to a wrong decision, e.g. approving an under par heat pump installation (buyer's risk) or rejecting a correct unit (vendor's risk; c.f. 1.3).

Preferably, ISO GUM (ISO 1993) should be practised. It uses the principle of aggregation of variances to find the combined uncertainty u of a derived result. To raise the confidence, the standard deviation of a result is multiplied by a coverage factor k to yield the expanded uncertainty U , i.e. $U = k \cdot u$. The resulting range will cover the results with an associated confidence level. ISO recommends $k = 2$ (equivalent to 95 % confidence). Fahlén (1994) describes in detail how to use the ISO principles in heat pump and refrigeration applications. The Nordtest methods use the principles recommended by ISO with a coverage factor $k = 2.5$ (a confidence level equivalent to 99 %; the ISO Guide was not available when NT VVS 076 was conceived). The Nordtest methods cover three basic levels of evaluation with approximate levels of expanded uncertainty of ± 5 % (level 1), ± 10 % (level 2) and ± 15 % (level 3). These levels are only to be taken as indicators. The real uncertainty can of course be lower or higher depending on the conditions of a specific audit.

2.3 Comparison of Measured and Specified Performance

Having determined actual performance at the relevant system boundary, there is still a need for methods to compare measured and specified performance. A contract must also specify permissible deviations and what happens if there is an excessive deviation. In this situation it is important to compare measured and specified performance at the same physical locations (system boundaries). Equally important is to compare data for the same operating conditions. Commissioning rarely provides the facilities of a testing laboratory to control temperatures and flow rates to specified values. Therefore, in addition to the uncertainties of measured performance, one has to consider the uncertainty of the specified performance induced by not knowing the exact operating conditions for the comparison.

Figure 3a illustrates how an uncertainty U_t of the heating water outlet temperature during testing translates to an uncertainty U_p of the specified capacity. A manufacturer's calculation program will obviously yield a different capacity for comparison at $t_{wo} + U_t$ than at $t_{wo} - U_t$ °C. Figure 3b shows that if the true values of the operating conditions (t_{bi}^* , t_{wo}^*) differ from the measured values (t_{bi} , t_{wo}), then the assumed reference value of the capacity ($P_{1,r}$) will differ from the relevant value ($P_{1,r}^*$) and there will be an error $\Delta P_{1,op}$ of the reference capacity in addition to the error of the measured capacity $\Delta P_{1,meas}$ ($t_{bi}^* < t_{bi} < t_{bi}^* + U_t$).

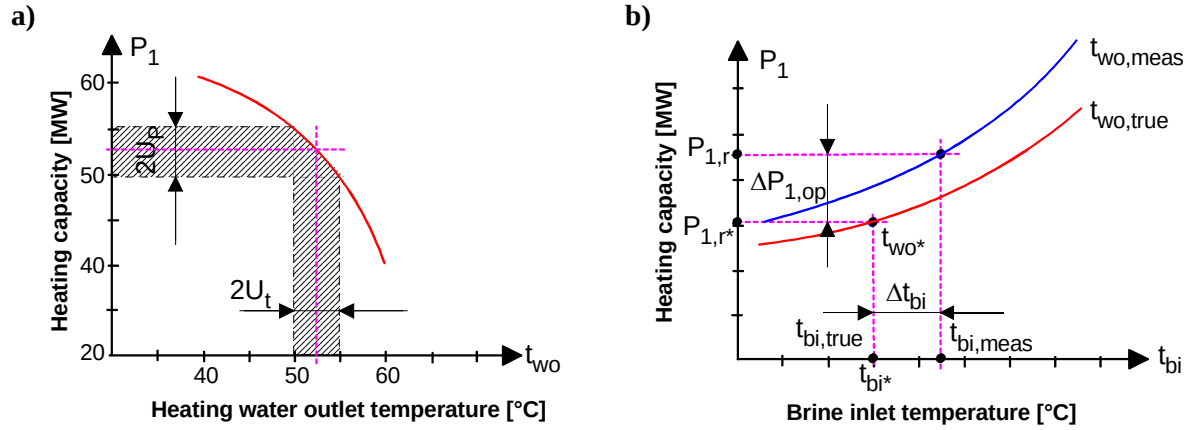


Fig. 3. Comparison of measured and specified capacity. Uncertainties in measured temperatures influence both the measured and the specified capacities.

2.4 Operating Range

The operating range can in some instances be an economically more important performance characteristic than capacity or *COP*. As an example, large heat pumps using lake water have to reduce the capacity in winter to avoid freezing. The economic benefit from being able to operate with full capacity at a lower water temperature with one heat pump may be worth more than a higher *COP* of another. High *COP* and a large operating range are, of course, by no means mutually exclusive. Still, one has to consider both. Figure 4a illustrates one method of describing the operating range. Line 1 relates to maximum operating pressure, 2 to maximum temperature, 3 to structural strength, 4 to motor load, 5 to various component capacities and 6 to the risk of freezing. A simplified alternative in 4b only limits the outlet temperatures from condenser and evaporator.

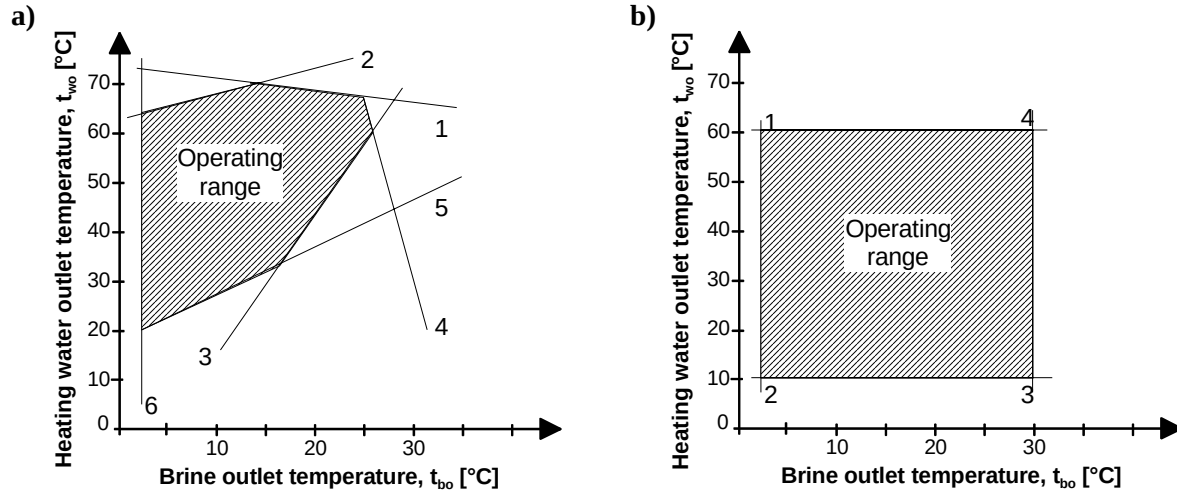


Fig. 4. Alternative presentations of operating range (NT VVS 115 1997).

3 METHODS FOR PERFORMANCE AUDITS OF HEAT PUMPS

The interface between heat pump and the external heat transfer media provides two alternative methods to measure performance. One is to measure on the *external* heat transfer medium side of the condenser or evaporator and the other to measure on the *internal*, refrigerant side. The external and internal methods are both indirect methods in the sense that the final results, capacity or *COP*, have to be calculated from a number of individual measurements and thermophysical property data.

3.1 External Methods

The traditional external method, e.g. NT VVS 076, requires measured values of flowrate, inlet and outlet temperatures and property data for density and specific heat capacity of the heat transfer fluid according to:

$$P = q_v \cdot \rho \cdot c_p \cdot (t_o - t_i) \quad [\text{Eq. 5}]$$

and the expanded uncertainty U_P of the capacity is given by:

$$\frac{U_P}{P} = \left(\left(\frac{U_{q_v}}{q_v} \right)^2 + \left(\frac{U_\rho}{\rho} \right)^2 + \left(\frac{U_{c_p}}{c_p} \right)^2 + \left(\frac{U_{\Delta t}}{t_o - t_i} \right)^2 \right)^{\frac{1}{2}} \quad [\text{Eq. 6}]$$

Experience from laboratory testing (Fahlén 1994), including Round Robin tests, indicates that it is possible to measure heating capacity with an expanded uncertainty U_P less than 1-2 %. This requires high-class equipment and lots of work. In practical installations one can rarely expect a lower uncertainty than 5 % (c.f. NT VVS 076 level 1). However, in some large installations it is possible to achieve sufficiently stable conditions and the economic incentives are large enough to motivate the effort to achieve accuracies on a par with laboratory levels. The limiting factor is the possibility to calibrate very large flowmeters at relevant flows and temperatures.

3.2 Internal Methods

The internal method NT VVS 116 relies on the mapped heat pump cycle in an h - p diagram (Fig. 5a). If the heat loss from the compressor is expressed as a fraction of the electric input, then $P_{loss} = f \cdot P_{em}$ and the COP can be derived from an energy balance (Fig. 5b) of the compressor:

$$COP_1 = \frac{P_1}{P_{em}} = \frac{q_{Rm} \cdot (h_3 - h_5)}{P_{em}} = \frac{P_{em} \cdot (1 - f) \cdot (h_3 - h_5)}{(h_o - h_i) \cdot P_{em}} \quad [\text{Eq. 7}]$$

and

$$COP_1 = \frac{(1 - f) \cdot (h_3 - h_5)}{(h_3 - h_7)} \quad [\text{Eq. 8}]$$

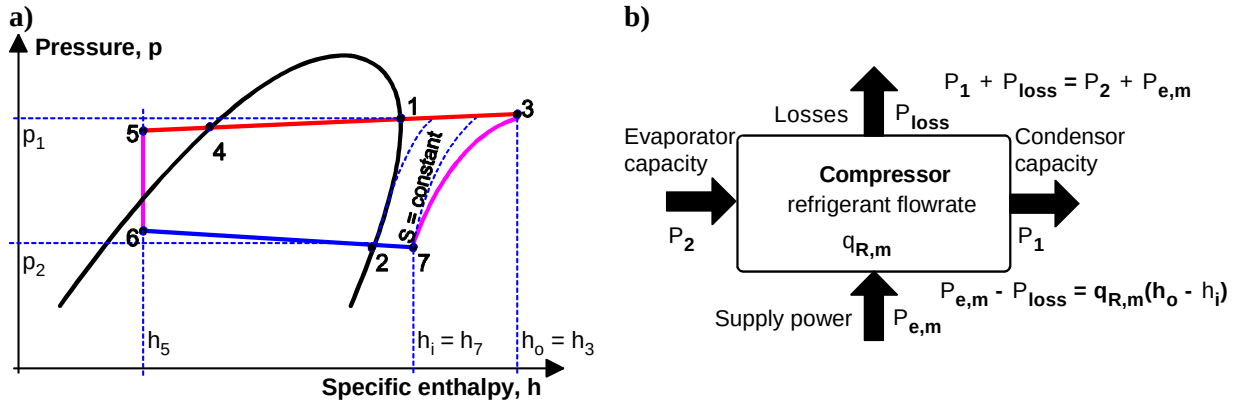


Fig. 5. The principle of using the h - p refrigerant diagram with assumed compressor losses to derive COP according to NT VVS 116.

COP_2 can be similarly derived. If the supply power is measured, then the thermal capacity, P_1 or P_2 , can be calculated according to $P = P_{em} \cdot COP$. The uncertainty of the calculated COP can be estimated by differentiating Eq. 8:

$$\frac{\Delta COP_1}{COP_1} = -\frac{\Delta f}{1 - f} - \left(1 - \frac{1 - f}{COP_1}\right) \cdot \frac{\Delta h_3}{(h_3 - h_7)} - \frac{\Delta h_5}{(h_3 - h_5)} + \frac{\Delta h_7}{(h_3 - h_7)} \quad [\text{Eq. 9}]$$

and then we have:

$$\frac{U_{COP1}}{COP_1} = \left(\left(\frac{U_f}{1 - f} \right)^2 + \left(\left(1 - \frac{1 - f}{COP_1} \right) \cdot \left(\frac{U_{h3}}{h_3 - h_7} \right) \right)^2 + \left(\frac{U_{h5}}{h_3 - h_5} \right)^2 + \left(\frac{U_{h7}}{h_3 - h_7} \right)^2 \right)^{\frac{1}{2}} \quad [\text{Eq. 10}]$$

Using tables, computer software or algorithms (Fahlén and Johansson 1989, NT VVS 116 1997) for the relation between specific enthalpy and measured temperatures and pressures, the three U_h in Eq. 10 can be replaced by corresponding measuring uncertainties for pressure, U_p , and temperature, U_t . With

$h = h(p, t)$ we have $\Delta h = \frac{\partial h}{\partial p} \cdot \Delta p + \frac{\partial h}{\partial t} \cdot \Delta t$ and hence

$$U_h = \left(\left(\frac{\partial h}{\partial p} \cdot U_p \right)^2 + \left(\frac{\partial h}{\partial t} \cdot U_t \right)^2 \right)^{\frac{1}{2}} \quad [\text{Eq. 11}]$$

Since isobars in the h - p diagram are almost vertical, $\partial h / \partial p$ is small and therefore quite large values of U_p are acceptable (c.f. table 1). Of the temperatures, t_3 is usually the most difficult to measure accurately and

thus the most critical. Table 1 gives examples of the maximum permissible measuring uncertainties of pressures and temperatures for some common types of refrigerants. The values are based on the relations regarding propagation of uncertainties given in Eq. 10 and Eq.11, assuming equal contributions of $\frac{15}{6} = 2.5 < 3 \%$ from each of the 6 contributors to a required total uncertainty of 15 % (Fahlén 1996).

Table 1. Examples of required component measuring uncertainties.

Type of refrigerant	Maximum permissible uncertainty ($\pm$)					
	$\frac{\Delta p_1}{p_1} [\%]$	$\frac{\Delta p_2}{p_2} [\%]$	$\Delta t_3 [K]$	$\Delta t_4 [K]$	$\Delta t_5 [K]$	$\frac{\Delta f}{f} [\%]$
R12	19	34	3.7	2.2	5.1	35
R22	19	30	3.4	2.8	5.3	35
R502	9	22	2.0	1.8	3.7	35
R717	78	50	8.0	5.5	9.7	35

4 RESULTS

Measuring uncertainty (Fahlén and Johansson 1989, NT VVS 116 1997) has been investigated theoretically and validated in laboratory as well as in field measurements. There is also ample experience from 20 years of using the principles of both the external NT VVS 076 (1989) and the internal method NT VVS 116 (Fahlén and Johansson 1989, NT VVS 116 1997).

4.1 External Methods

Commissioning of large heat pumps is normally conducted to decide on contractual bonuses or penalties. Hence, the client is prepared to put in a substantial effort to get reliable results and accuracy in field tests hardly gets any better than this. For instance, commissioning tests (Fahlén and Lagerkvist 1987) of 27 heat pumps with capacities ranging from 10 to 40 MW showed calculated uncertainties ranging between 1.4-4.7 % with a mean value of 2.8 %.

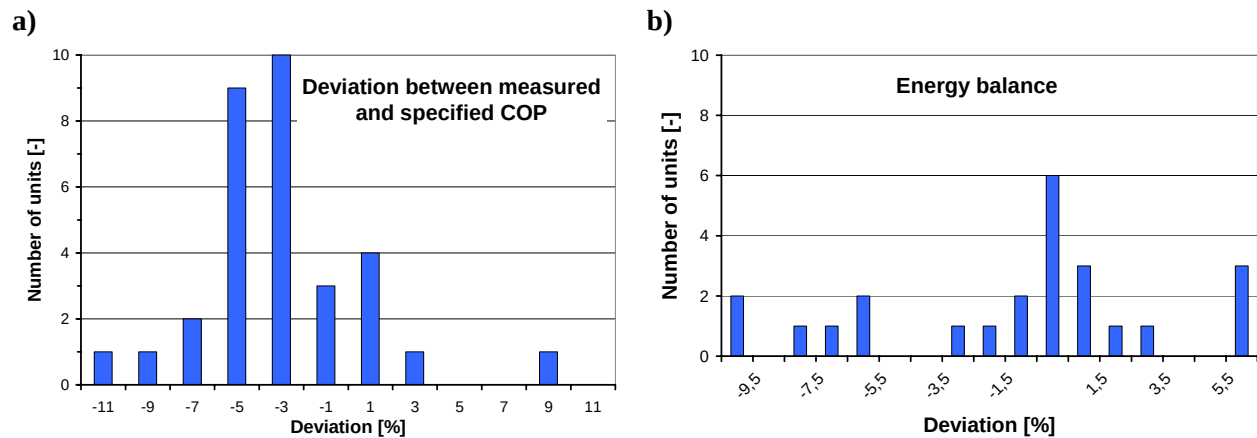


Fig. 6. Number of tests with a certain contractual deviation (a) and energy balance (b).

Figure 6a indicates typical deviations from guaranteed performance and 6b provides results of the energy balance for installations with measuring equipment on both sides of the heat pump. Where the instrumental quality is the same on both sides, the mean deviation is -0.3 % with a largest deviation of -3.2 %. Theoretically, the energy balance EB should be negative from the definition

$$EB = 100 \cdot \frac{P_1 - (P_2 + P_{em})}{P_1} = -100 \cdot \frac{P_{loss}}{P_1} [\%].$$

4.2 Internal Methods

Fahlén and Johansson (1989) and Munter (1989) provide examples of successfully conducted evaluations using the refrigerant state method. The report by Fahlén contains more than 110 tests on 13 heat pumps, 5 of which were tested in a laboratory environment and 8 were tested in situ. The laboratory tests include investigations on the effects of changing condensing and evaporating pressures, outlet, super heat and sub-cooling temperatures and the heat loss from the compressor. In these investigations the internally determined COP was in no case outside $\pm 15\%$ of the externally measured COP . Indeed, when it was possible to mount the sensors properly, the difference was less than $\pm 5\%$. With the right conditions, the internal method can generate quite accurate results. However, for both the external and internal methods, the quality of results relies heavily on the competence of the auditor.

Figure 7 shows how variations in heat loss affect the deviation between the external and internal methods. Tests were made by operating two heat pumps with (SL) or without (ML) their cabinets at a number of different operating conditions. The losses were further increased by means of forced convection (LL) and only then did the deviation exceed $\pm 5\%$. In the calculations, f was maintained constant at 8 %. As Eq. 9 predicts, the deviation changes in a positive direction when the assumed value of f underestimates the losses.

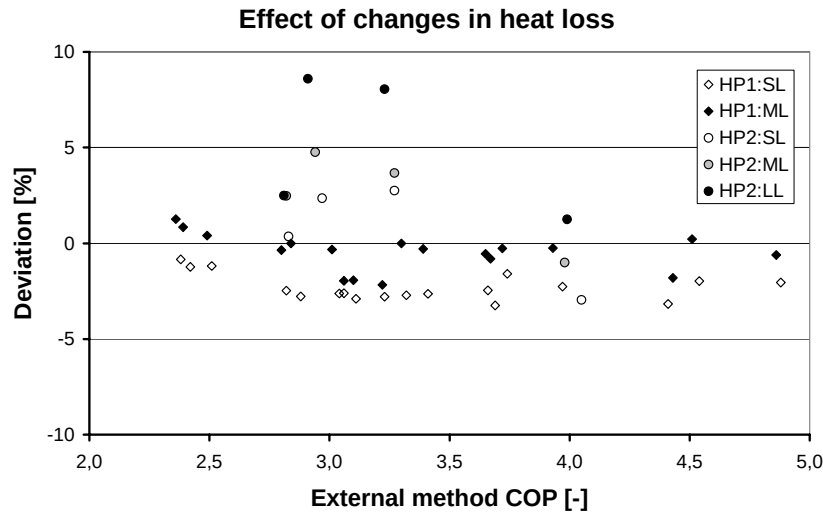


Fig. 7. Effect of increasing the heat losses. Comparison of internal and external methods on two heat pumps. On HP2, losses were both reduced (HP2:SL) and increased (HP2:LL) from the normal (SL, ML and LL = small, medium and large losses).

For small ($P_{e,m} < 5$ kW) hermetic compressors placed indoors and normally insulated, a typical value of the loss factor is 7-8 % and the reported range 5-15 %. The same type of unit placed outdoors, and/or poorly insulated, could have a loss factor of 15 % with a reported range of 10-25 %.

5 DISCUSSION AND CONCLUSIONS

Commissioning requires definitions and procedures for system boundaries, presentation of stated performance data, performance testing and the related uncertainty, recalculation of measured data to specified conditions, and finally a method for comparison of stated and measured data and the uncertainty of the comparison. ASHRAE (Phelan, Brandmuehl and Krarti 1996) and Elforsk (Fahlén 2001) have conducted extensive reviews of existing on-site measuring methods and the Nordtest methods seem to be the most developed and validated. NT VVS 115 (1997) provides general information regarding the prerequisites for performance testing of heat pumps, NT VVS 076 (1989) provides specific information on large units including methods of testing, uncertainty and comparison of measured and stated performance while NT VVS 116 (1997) gives the details of an alternative, less costly, method of measurement. There is also an ISO standard (1968) for testing of refrigerating systems.

The presented Nordic methods include preparation, methods of measurement and uncertainty analysis, and requirements on accuracy and stability. Extensive laboratory and field measurements, including sensitivity analysis from validation programs for both the *external* and *internal* measuring principles, indicate that generally achievable measuring uncertainties are less than 2 and 5 % respectively. The actual uncertainty must always be derived for each test and will depend not only on the measuring equipment but also on the conditions of installation and operation. One particular advantage of the *internal* principle is that the method not only discloses a deviation from expected performance but it also provides information as to why. This can be put to good use in expert FDD-systems if heat pumps come equipped with permanently installed sensor for internal measurement of *COP*.

The economic consequences of deviations in large district heating heat pump installations may serve to illustrate the importance of assessing measuring uncertainties. In such a plant, a temperature uncertainty of 0.1 K may be equivalent to an annual operating cost of more than 100 000 Euro. Hence there is a price connected to uncertainty and this should reflect on the care and resources used in planning, instrumentation and evaluation of the commissioning test in question.

6 NOMENCLATURE

Roman symbols

A	area
c	specific heat capacity
COP	Coefficient of Performance
f	compressor loss factor
h	specific enthalpy
k	coverage factor
M	mass
P	power (capacity)
q	flowrate
Q	heat (thermal energy)
t	temperature (Celsius)
T	temperature (absolute)
u	combined uncertainty
U	expanded uncertainty
U	thermal transmittance
W	work (mechanical or electric)

Units

$[m^2]$
$[J/kg/K]$
$[W_{thermal}/W_{electric}]$
$[J/K]$
$[W/K]$
--
$[kg]$
$[W]$
$[m^3/s], [kg/s]$
$[W]$
$[^{\circ}C]$
$[K]$
[relevant unit]
[relevant unit]
$[W/m^2/K]$
$[J]$

Subscripts

b	brine
c	cooling
com	compressor
e	electric
f	fan
h	heating
hp	heat pump
hps	heat pump system
hs	heating system
i	inlet (of heat pump)
m	mean; motor; mass
o	outlet (of heat pump)
op	operating condition
p	pump; pressure
P	power
r	reference
R	refrigerant
rh	room heater
$room$	room
sh	supplementary heating
t	temperature

Greek symbols

ρ	density	$[kg/m^3]$
ξ	fraction of drive power to pump or fan that heats the heating fluid	$[W_{thermal}/W_{electric}]$

Abbreviations

CEN	Comité Européen de Normalisation (European Committee for Standardization)
Elforsk	Swedish Electrical Utilities' R&D Company
FDD	Fault Detection and Diagnosis
Nordtest	Organization for Nordic test methods

Subscripts (continued)

v	volume
w	water
1	condenser (heating)
2	evaporator (cooling)

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