A review of the fault behavior of heat pumps and measurements, detection and diagnosis methods including virtual sensors

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ABSTRACT

Heat pumps are increasingly being installed in domestic buildings, which is an opportunity to reduce the energy needed to heat and cool buildings. However, this reduction will only be possible if the equipment is maintained correctly. Fault detection and diagnosis (FDD) systems can improve the cost of operating and maintaining heating and air conditioning systems, specifically heat pump, while keeping their performance. This article first briefly introduces the methodologies used in FDD systems for heat pumps and compare their performance. Then, an accurate description of the common faults and their effects on field vapor compression systems is made. After that, a compilation of which faults and how are emulated in both laboratory conditions and virtual environments are extensively described. The measurements needed to perform the diagnosis are analyzed along with instrumentation needed for FDD systems. Finally, the virtual sensors applied to heat pumps to reduce the cost associated with FDD implementation are described. This article aims to establish a criterion for selecting which faults can be tested under laboratory conditions or by simulation with a virtual model and to determine the features that identify those faults. Finally, several areas of improvement for the aspects reviewed have been identified: increase the use of performance indicators for FDD, new and updated studies about the health status of field heat pumps, testing methods that take into account the gradual and probabilistic nature of heat pump faults and further research in the use of virtual sensors in FDD systems.

Nomenclature

AHUs	Air handling units
AFDD	Automated fault detection and diagnosis
COP	Coefficient of performance
E	Electrical energy
f	Compressor frequency
FDD	Fault detection and diagnosis
FI	Indoor unit fouling
FO	Outdoor unit fouling
FXO	Fixed orifice
I	Current
IMC	Indoor unit mechanical component failure
LL	Liquid line restriction
LMTD	Log mean temperature difference
NILM	Non-intrusive load monitoring
NON	Non-condensable gas
Q	Reactive power
OMC	Outdoor unit mechanical component failure
RTU	Rooftop unit
RO	Refrigerant overcharge
RU	Refrigerant undercharge
SEN	Sensor error
SHR	Sensible heat ratio
TXV	Thermostatic expansion valve
UA	Overall heat transfer coefficient (U) multiplied by surface area (A)
V	Voltage
VE	Evaporator valve position
VL	Valve leakage
W	Compressor electrical power consumption
h_{aie}	Enthalpy inlet air
h_{aoe}	Enthalpy outlet air
m_{condr}	Water flow rate at condenser
m_{ref}	Refrigerant flow rate
m_r	Water flowrate
P_{dis}	Refrigerant discharge line pressure
P_{ll}	Refrigerant liquid line pressure
P_{suc}	Refrigerant suction line pressure

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PO_{feed}	Pressure oil feed
R_f	Fouling factor
rated	Subscript: manufacturers' rating conditions are used
T_{ai}	Inlet air temperature
T_{amb}	Ambient air temperature
T_{ao}	Outlet air temperature
T_c	Refrigerant condensing temperature
T_{condro}	Water temperature at condenser outlet
T_{dis}	Refrigerant discharge line temperature
T_e	Refrigerant evaporating temperature
T_{euapri}	Refrigerant temperature at evaporator inlet
T_{euapro}	Refrigerant temperature at evaporator outlet
T_{euapro}	Water temperature at evaporator outlet
T_{indoor}	Indoor air temperature
T_{ll}	Refrigerant liquid line temperature
T_{suc}	Refrigerant suction line temperature
T_{ri}	Water inlet temperature
T_{ro}	Water outlet temperature
α	Degradation coefficient
Φ_{aie}	Inlet air relative humidity
Φ_{aoe}	Outlet air humidity

1. Introduction

Space heating and cooling, and water heating represent 79% of the energy used by households in the European Union (EU) [1]. Although most spaces and water are heated by gas, cooling is produced largely by electricity [1] Under these conditions, heat pump technology provides an efficient and sustainable solution for heating and cooling buildings.

The European heat pump market has been growing each year, with a 12% increase in 2018 [2]. This increasing tendency is expected to contribute significantly to the mitigation of CO_2 emissions because of energy savings. Predictions under different scenarios yield total reductions in CO_2 emissions ranging from 34% to 46% by 2030, the result of heat pumps being extensively integrated into new and retrofitted buildings [3].

Even though the introduction of heat pumps improves classical heating and cooling applications, there is evidence to suggest that in most installations there is a lack of maintenance that leads to significant operational inefficiencies [4]. Measured performance data of heat pumps surveyed in buildings show that a high percentage of applications need servicing due to major compressor repair and heat transfer degradation [5]. Field surveys point to significant losses in the performance of heat pumps installed in buildings, with 20-50% of them operating at 70-80% efficiency or lower than their design efficiency [6], with faulty operation contributing an additional 40% of energy consumption [7]. The loss of heat pump efficiency means significant energy losses and maintenance costs [8,9]. Likewise, degraded efficiency leads to higher peak demand levels in periods of high energy usage, such as the cooling season in the summer [10]. Under a global warming scenario, maintenance of heat pump efficiency has been shown to be an adaptive measure to compensate for the increase in electricity consumption associated with the higher cooling demand during hot periods [11].

The degradation of heat pump efficiency can be prevented with automated fault detection and diagnosis techniques that identify faults in the system components, which leads to savings in energy, service and operating costs [12,13]. Besides, reliable comfort of occupants is one of the co-benefits of the better maintenance of heat pump installations [10]. At the building level, the irruption of affordable data mining processes has increased the interest in FDD [14]. FDD in buildings helps building engineers and facility managers to improve preventive maintenance actions and to reduce maintenance costs [15]. The detection of faulty behavior that usually would pass undetected for long periods prevents the performance degradation of HVAC systems and the increase of energy consumption. The diagnosis identify which components are not working properly thus reducing the maintenance cost and the downtime of the equipment.

The aim of the present article is to make a review of the common faults and features that need to be taken into account when developing an FDD and it is structured in seven sections. First, section 2 presents the methodology used in this review. Section 3 of the article compares the different reviews in the FDD area, identifying that little focus has been placed in how to reproduce heat pump faults in a laboratory or by simulation, or which are the useful features for feeding the FDD algorithm. The same section does a comparison of the FDD performance with quantitative values as this has been identified as not sufficiently addressed in available reviews. Section 4.1 described the different faults and their effects on vapor compression system as an introduction for the next sections. The health of field heat pumps is reviewed to prioritize each fault by its occurrence in section 4.2. Section 5 describes how to emulate those faults in a laboratory or a simulation environment as part of the validation process of an FDD. Finally, section 6 focuses on feature selection and instrumentation needed for FDD systems. As one of the main disadvantages of applying FDD to domestic heat pumps is the cost of adding additional sensors, section 6 describes the use of virtual sensors as a way of overcoming this disadvantage. Finally, the article concludes identifying several areas of improvement regarding methodological aspects for FDD methods in heat pumps, including instrumentation with virtual sensors and simulation techniques for FDD methods.

2. Methodology

The methodology adopted for the realization of this review consisted of the broadly described next steps:

- Review research: Using Scopus database, the main reviews of the area were searched. For the search, keywords as fault detection, compression cycle and FDD were used. The reviews found were classified in general fault detection and diagnosis methods without any specific application, FDD applied to building systems and FDD applied to specific HVAC equipment. The latest are the basis for the present work.
- Subtopic selection: After a thorough reading, most of the reviews focused on the FDD method itself. Some topics where covered superficially despite the importance in the development and validation of the FDD. Those subjects where selected as the subtopics of the present review: common faults in heat pumps, experimental fault emulation, measurements needed for FDD and the application of virtual sensors.

- Article search and filter: A new article search about those subtopics was made and handled by a reference manager. They were classified by different criteria as the device used (heat pumps, refrigerators, air handling units…) or the faults tested. The articles that focused, or which content can be applied to domestic heat pumps, were selected.
- Subtopic description, analysis and synthesis: For each of the subtopics of the present review a literary description is made. The outcomes of each article are synthesized and compared with quantitative data in tables. Each of the sections concludes with a brief description of the found issues. These issues are pointed out as possible research lines in the general conclusions.

3. Review analysis and performance comparison of FDD methods

Fault Detection and Diagnosis for heat pumps has been extensively investigated in the last three decades [16,17], and several reviews have been made. This section analyses the most relevant ones and a quantitative comparison of the efficiency of different FDD methods is made.

Venkatasubramanian authored a series of three review papers [18–20] on the general FDD process. The first article of the series discusses the desirable characteristics of an FDD. These characteristics range from robustness to measurement noise to the detection of simultaneous faults. It also explains how to extract useful features from measurements. The term feature is used in FDD literature to express a measurement made by the equipment that can provide information useful for performing FDD. The same article and the next two [19,20] give an explanation of the different FDD methods classified by the a priori knowledge about the system to perform the FDD. This classification is adapted to building systems by Katipamula and Brambley in two review articles [4,15]. The first [15] gives an accurate description of the use of FDD in systems such as refrigerators, air conditioners, heat pump, chillers and air handling units (AHUs). The methods are classified on the basis of the type of knowledge used to perform the diagnosis. There are three main groups: quantitative model-based, qualitative model-based and process history based. Quantitative model-based methods use models obtained from basic principles to perform the diagnosis. Qualitative model-based methods rely less on the accuracy of the data and uses ranges of values to extract qualitative values from the features. This group consists of methods based on a series of if-then-else rules (rule-based) and applies expert knowledge (expert systems). The group of process history based methods relies on the empirical relation between output and input data. This group is divided into two subgroups: black box models, which do not need any knowledge about the process and perform the diagnosis through the relationship between inputs and outputs of the system, and grey-box models, which need some knowledge of the system studied. The second article [4] describes the faults, measurement and FDD method used for each type of building system. Some of these FDD are statistical rule-based methods [21] under steady state and transient conditions [5], limit-and-trend checking and physical model-based approaches [22]. The review by Katipamula and Brambley [15] is the basis for a later review by Kim and Katipamula [14] of FDD for building systems, which focuses on the publication trends of each of the methods described in Katipamula's classification between 2005 and 2018. Despite this review does not focus on the performance of the different methods, the main conclusion is that there is an increasing tendency to use process history-based methods to perform FDD. The simplicity of the black box model and the increasing process capacity of electronics are the main reasons for this. This tendency becomes evident in the review of Zhao et al. [23] who authored a review on artificial intelligence-based FDD for buildings. Zhao et al. pointed out that there is a lack of articles on sensor faults and physical component faults in the same FDD. This increase in the use of process history-based methods means that new methodologies are applied to FDD in HVAC systems. So, Yu et al. [24] propose a new classification of methods alternative to Katipamula's. FDDs applied to AHUs were revised and methods such as Artificial Neural Networks (ANN), principal component analysis (PCA) and machine learning were described.

A recent review by Rogers et al. [10] focused on advances in the development of FDD methods for residential air-to-air conditioners. In contrast to the other reviews that describe the method to perform the FDD and describe briefly the faults, Rogers defined the common processes of an FDD for air conditioning systems: feature selection, steady state detection, fault-free modeling, fault detection and fault diagnosis. This point of view is necessary to use the FDD as a complete product ready for the market, rather than a method still in development. They described each of the parts and gave examples of some faults tested, steady state detectors and fault detection methods. The article concludes with some comments on the potential of using smart thermostat data for fault detection and the use of virtual sensors as features.

While there are many reviews that classify and explain the different methods to perform an FDD, none has been found which compares the method used with their performance. Table 1 shows the comparison of quantitative performance parameters of different FDDs found in the literature highlighting the type of method implemented. In order to compare different methods, the performance parameters must first be defined. Yuill and Braun [25] defined no response, the false alarm rate, the misdiagnosis rate and missed detection as the parameters for comparing FDD methods. Shi and O'Brien [26] used a more statistical point of view, and identified the false positive rate (false alarm), false negative rate (missed detection), precision (accuracy) and detection time as the parameters for validating FDD performance. However, few authors use these factors to determine the performance of their FDD. Li and Braun [27] used a rule-based FDD system with a decoupling method, obtaining low rates of false alarms. Zogg et al. [28] obtained different false alarms rates depending on the number of training and validation data sets for their grey-box model. Wang et al. [29] compared the accuracies of different types of fault diagnosis techniques with Bayesian networks. Yuill and Braun [30] developed a tool to evaluate the performance of different FDD protocols and obtaining different performance parameters as false alarm, misdiagnosis and missed detection rates. Li et al. [31] obtained good accuracy results for their tree-based FDD that uses virtual sensors. In Table 1 accuracy and false alarm rate have been used as performance parameters as they are the most common ones used in the reviewed paper allowing direct comparison. However, it is difficult to extract conclusion from the results when the faults emulated or the fault intensity are not the same for each method. In some cases, the authors have tested conditions quite different from those of the training data, whereas others used data similar to the training phase, so the performance values of the latter may be better.

Table 1 – Performance comparison between different FDD methods. CH: chiller, HP: heat pump, m: air flow rate sensor, MS: multi-split, P: pressure sensor, RTU: rooftop unit, T: temperature sensor and VS: virtual sensor. The superheating and subcooling temperatures are obtained from pressure sensors and condensing and evaporating temperatures from temperature sensors.

Category	Method	Equipment	Data	No. Sensors	Accuracy	False alarm rate	Ref.
Qualitative	Rule-based	HP	Real	4T,3P and 2m		1.6%	[27]
History-based	Grey-box	HP	Real	4-5T		3%	[28]
History-based	Black-box	CH	Real	7T, 3P	93%		[29]
Qualitative	Rule-based	RTU	Simulated	Unspecified		35%	[30]
Qualitative	Rule-based	RTU	Simulated	Unspecified		55%	[30]
Qualitative	Rule-based	RTU	Simulated	Unspecified		15%	[30]
History-based	Tree-based	MS	Real	28T	94%		[31]
History-based	Tree-based	MS	Simulated	28T	80.55%		[31]

4. Common faults and their effects on vapor compression systems

4.1. Types of fault in vapor compression systems

Research into common faults and their detection is more prolific for larger systems [10] such as climatization for buildings [24]. This is because the cost impact of increasing the number of sensors in these systems is considerably lower than for domestic units. Also, the cost of repairing a fault and the down time on such systems are more significant [32]. Nevertheless, although this section focuses on larger heating or ventilation systems, some faults can be found in all vapor compression systems.

The key variables affected by those faults are the coefficient of performance (COP), the power consumption of the compressor, the condenser inlet saturation temperature (which depends on the discharge line pressure), the evaporator exit saturation temperature (which depends on the suction line pressure), subcooling (which depends on the pressure in the liquid line) and superheating (which depends on the pressure at the evaporator outlet). Because the effect of these faults depends on the type of heat pump and whether the system is in cooling or heating mode, we have used an air-to-air heat pump working in cooling mode to describe the effect of the faults on the variables.

4.1.1. Outdoor unit fouling and outdoor mechanical component failure: FO & OMC

In real installations the outdoor unit is normally exposed to dirt, debris and other components (fallen leaves) that can accumulate in the filters or gaps in the heat exchanger. This fouling can accumulate at the air inlet or between the internal gaps. Both types of fouling reduce the flow of the external stream (air or water). The first type can also increase heat resistance.

Another fault that can affect the outdoor unit is the failure of mechanical components (OMC). The fan or pumps may not work correctly due to a mechanical or a control problem. In this case, the main repercussion is the decrease in external stream.

In field studies, outdoor unit fouling fault (FO) does not significantly affect indoor comfort levels [5]. Only 14% of outdoor unit fouling faults have been found to reduce the comfort in the building [5]. This is supported by Mehrabi and Yuill [33], who show that the fouling effect of field heat pumps has a nearly negligible effect on performance (0.2% reduction) and capacity.

In the literature, to test the fouling fault, the authors reproduced this fault by blocking the air inlet, or reducing the water or air flow. In some studies on particulate fouling, the impact on coil heat transfer is small, and the main effect is an increase in the air-side pressure drop [34]. Therefore, the effects explained below will consider only the reduction in air flow and not the increase in heat resistance, which is more relevant in virtual simulations.

For the levels tested in the literature, the reduction in airflow involves an increase in condensing temperature and pressure [35]. The temperature of the discharge line in the compressor and condenser inlet saturation temperature are higher [5,36]. The COP of the unit is lower, because the power consumption of the compressor increases and the thermal capacity decreases [5,35,37–39].

4.1.2. Indoor unit fouling and indoor mechanical component failure: FI & IMC

Even though the indoor unit is not affected by the same type of dirt and debris as the outdoor unit, it can be affected by fouling because particulates and fibers create deposits that obstruct air circulation or increase heat transfer resistance. These deposits can also act as substrate for bacteria growth [40], which worsens the air quality. As with the outdoor unit, the fans, pumps or controls of the indoor unit can also have mechanical faults. And for systems with fan-coils, the functioning of the fan or the size of the duct can reduce the air flow rate as well.

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Although some authors think that the fouling of the indoor heat exchanger can reduce the heat transfer of the device, it has been shown that this reduction is negligible [41]. Fouling can even increase heat transfer [41], leaving the decrease in the external stream flow rate as the main repercussion of fouling. Like the outdoor unit, the reduction in external stream flow rate is the only explanation considered for the effect of those faults.

Low air flow in the evaporator decreases the saturation temperature at the evaporator exit, while the air temperature drop in the evaporator increases [5,35,36]. This means that the sensible heat ratio (SHR) decreases as well, which improves moisture removal [38]. This fault has a greater effect on capacity than on COP [5].

4.1.3. Valve leakage: VL

In the literature, the valve leakage fault refers to a leakage of refrigerant from the discharge line to the suction line. This leakage is normally associated with the valves at the inlet (suction line) and outlet (discharge line) of the compressor. These valve seals can break which, in turn, leads to leakage. Leakage can be also found in the 4-way valve, but because the consequences are the same, they are both classified in the same group [42].

If the valve in the suction line is loose, when the compressor is in the compression phase, high pressure refrigerant may enter the suction line. If the discharge valve is loose, when the compressor is in the suction phase, high pressure refrigerant from the discharge line may enter the compressor. For the 4-way valve, there is also a refrigerant bypass between the suction and discharge line. Therefore, the high pressure vapor enters the suction line, which decreases the mass flow rate of refrigerant.

The refrigerant saturation temperature at the condenser inlet decreases while the saturation temperature at the evaporator exit increases [36]. For heat pumps equipped with a thermostatic expansion valve (TXV), the actuation of the valve decreases the mass flow rate in response to the increase in the saturation temperature at the outlet of the evaporator [35]. This fault increases the SHR and affects capacity and COP [5]. In some studies it has considerably degraded COP [38].

4.1.4. Non-condensable gas: NON

During the commissioning, before the refrigerant is charged, the circuit must be purged to remove all the air and moisture inside. When the vacuum pump does not work correctly, some air or moisture will remain in the system. The presence of non-condensable gases may raise the system pressure, reducing the efficiency and even causing a TXV malfunction.

Non-condensable gases settle in the condenser, so the condenser pressure is increased by the partial pressure of the non-condensables [36]. This decreases capacity and increases compressor power consumption, and has been identified as the fault that has the largest impact on system performance [37].

4.1.5. Refrigerant overcharge: RO

During the commissioning, the technician may apply a higher refrigerant charge than is specified by the manufacturer. This fault increases the subcooling value [36] and degrades the COP [37,39]. In some cases, for fault levels below 20%, it can increase the capacity of the heat pump [38], but this may depend on the specific characteristics of each heat pump.

4.1.6. Refrigerant undercharge: RU

During the commissioning, the technician may apply a lower refrigerant charge than is specified by the manufacturer. Also, any leakage of refrigerant may decrease refrigerant mass so the optimal working point will be lost. In this case there will be a continuous loss of refrigerant, so the fault intensity will increase over time whereas in the first case, the fault intensity remains the same.

Mehrabi and Yuill [43] found that, for devices with TXV, when the level of undercharge is below 20%, the effects are nearly negligible.

The features that are most sensitive to this fault are subcooling and superheating [37,38]. The speed of the variable compressors and the refrigerant flow rate increases, which reduces the range of outdoor operating conditions [39]. Breuker and Braun [5] found that this fault has a greater effect on thermal capacity than on COP for equipment with fixed orifices (FXO), but Du et al. [38] found that this was one of the faults that most degraded COP in a heat pump with TXV.

4.1.7. Liquid line restriction: LL

Normally, a filter or dryer is installed in the liquid line to remove solid particles from the refrigerant. These particles may enter the circuit if a technician does not follow good refrigerant charging practices or if the material is rusty as a consequence of inefficient tube joinery. This debris will clog the filter/dryer and increment the refrigerant flow restriction. As a consequence, there will be a reduction in refrigerant mass flow rate, an increase in suction superheat and a reduction in cooling capacity [35]. However, for devices with TXV, the fault effect is negligible for fault levels below 10% [35,36] and has a greater effect on thermal capacity than COP [5].

4.1.8. Sensor error: SEN

All FDD methods use sensor measurements for detection and diagnosis. The sensors are affected by ageing, which can lead to a loss of precision [44]. For methods that use large quantities of sensor data for data mining, the detection of a drift or bias in measurements is especially important [45]. Sensor accuracy will affect not only FDD performance but also the efficiency of the heat pump itself. If a faulty sensor is used for the internal control of the device, the performance and the comfort of the occupants can be affected [46]. Therefore, the sensors measuring the heat pump or FDD operation must be monitored to ensure they are providing correct data.

4.1.9. Other errors

The errors explained above have been discussed frequently in the literature, but other errors are less frequent or do not occur in current equipment. Some of these faults are: compressor liquid ingestion [47], expansion valve malfunction [48] and excess of oil in the refrigerant circuit [49].

In general, faults have much less effect on heat pumps equipped with a TXV than pumps with fixed orifices as expansion devices [50]. The same occurs for systems with variable speed compressors [51]. Table 2 shows a comparison between the quantifiable effects of the faults mentioned on COP and thermal capacity. The table only takes into account the references that give a quantitative value of the effect and fault levels of around 10%. Note that for the FO, IMC and FI faults, the authors talk about fouling or faulty fans, but they are all reproduced by reducing the external stream flow rate.

Table 2 – Summary of the quantifiable effects on thermal capacity and COP of common faults for different types of heat pump. For changes below 0.5% the parameter has been assumed to remain the same. FS: fixed speed, VS: variable speed.

Fault type	Type of pump	Compressor	Mode	Effect	Ref
FO	Air-to-Air	FS	Cooling	COP decreases 9%, Capacity decreases 14%	[52]
	Air-to-Water	VS	Heating	COP decreases 0.77%	[53]
	Air-to-Air	FS	Heating	COP decreases 1.34%, Capacity remains	[54]
IMC	Air-to-Air	FS	Cooling	COP increases 33%, Capacity decreases 13%	[52]
FI	Air-to-Water	VS	Heating	COP remains	[53]
	Air-to-Air	FS	Heating	COP and Capacity remains	[54]
VL	Air-to-Air	FS	Cooling	COP decreases 15%, Capacity decreases 24%	[52]
	Air-to-Air	FS	Heating	COP decreases 4.4%, Capacity decreases 4.7%	[54]
RO	Air-to-Air	FS	Cooling	COP increases 22%, Capacity decreases 36%	[52]
	Water-to-Water	VS	Cooling	COP decreases 2%, Capacity remains	[55]
	Water-to-Water	VS	Heating	COP and Capacity remains	[55]
	Air-to-Water	VS	Heating	COP decreases 3.4%	[53]
	Air-to-Air	FS	Heating	COP decreases 2.7%, Capacity increases 1.5%	[54]
RU	Air-to-Air	FS	Cooling	COP increases 18%, Capacity decreases 29%	[52]
	Water-to-Water	VS	Cooling	COP decreases 4.6%, Capacity decreases 6.6%	[55]
	Water-to-water	VS	Heating	COP decreases 10.6%, Capacity decreases 15.7%	[55]
	Air-to-Water	VS	Heating	COP increases 1.2%	[53]
LL	Air-to-Air	FS	Cooling	COP increases 8%, Capacity decreases 20%	[52]
	Air-to-Air	FS	Heating	COP and Capacity remains the same	[54]

4.2. Common faults found in field studies

In general, there is a lack of data on heat pump faults resulting from ageing in real installations. Repair services are the main source of data on field installations. These services are normally requested when there is a problem that compromises the comfort of the equipment owners. In these cases, the repair report only reflects the cause of the problem, but not the conditions that led to the fault. Thus, the data provided will be about hard faults, whereas soft faults have been the main focus of FDD literature. Hard faults appear suddenly and can stop the system from working or make it impossible to reach comfortable conditions. For this reason, hard faults are easy to detect once they occur and are easy to diagnose. On the other hand, soft faults normally allow the system to continue operating but degrade performance and may get worse over time. Soft faults are difficult to detect and, if not repaired, may lead to hard faults. Breuker and Braun [5] focused on the common faults of rooftop air conditioners. To this end, they analyzed a database with 6000 separate faults between 1989 and 1995. The faults that led to inadequate building comfort, in decreasing order of number of occurrences, were: control errors, electrical problems, refrigerant leaks, condenser-related faults, air handling, evaporator-related faults, compressor problems, cooling water loop problems, plugged filters, personnel error, expansion devices and others that were not classified. However, motor failures were classified with associated equipment (e.g. faults in evaporator motor fans are classified as evaporator related). Compressor faults were related to electrical faults (short to ground and open windings), mechanical faults (locked rotor and broken compressor internals) and compressor valve leakage or other leakage paths. Another cause of early compressor failure was high compressor temperature, which may be caused by low air flow in the condenser, liquid line restriction or a low refrigerant charge. For the condenser, the faults were related to mechanical and electrical problems with the fan, and fouling of the condenser. The evaporator faults were fouling and coil rupture/damage. If the repair cost is analyzed, compressor, control and condenser faults were the most expensive. Brownell et al. [56] published an article about malfunctions in a large refrigeration system. Those malfunctions were: plugged liquid line filter, low refrigerant charge and refrigeration system control not working properly. They concluded that changing the control system can significantly reduce the operational cost of the device. Ahn et al. [40] focused on the fouling of fin-and-tube heat exchangers of air conditioners, and they collected operational data from heat pumps installed in inns, offices and restaurants during their opening hours. The cooling capacity and the indoor air quality were negatively affected. The type and concentration of pollutants and the structure of the heat exchanger influenced the degree of fouling. Madani and Rocatello [57] examined the repair orders of Swedish heat pump producers during the warranty period, focusing on four different types of heat pump: Air/Air, Air/Water, Brine/Water and Exhaust Air/Water. They concluded that control and electronics, and temperature sensor faults were among the most common and costly faults.

The studies that focused on the health status of field heat pumps may not be representative of the actual market situation. Some of them [5,56] studied equipment assembled before the year 2000. Since then, heat pump production has changed dramatically, in particular everything to do with electronics and controls, devices with TXV or EEV *(Electronic Expansion Valve)* and compressors with variable speed [58]. Madani and Rocatello's study [57], despite being relatively recent, focuses only on repair orders during warranty periods. So it can be assumed that the user has noticed a malfunction and that most faults will be hard faults. Also, because it is limited to the warranty period, faults due to the ageing of the components are hardly represented. In conclusion, there is a need for more field studies about the health status of field heat pumps to update the data about which faults occur and with which frequency.

5. Faults reproduced under laboratory testing and simulation environments

5.1. Faults reproduced in the laboratory

Table 3 shows a compilation of the different faults tested on real equipment under laboratory conditions. An explanation of the emulation techniques for each fault is given. Rossi et al. [21] wrote one of the first articles on reproducing faults under laboratory conditions. They emulated condenser fouling, evaporator fouling, leaky compressor valves, liquid line restriction and refrigerant leakage. These are the same faults used by Breuker and Braun [5]. In both articles, condenser fouling corresponds to the fault FO. This fault was reproduced by partially blocking the finned area of the coils with paper. Evaporator fouling (FI) was reproduced by placing paper on the air-side filter. VL was emulated by opening a hot gas bypass line controlled by a manual valve. LL was implemented by including a valve in the liquid line to increase the pressure loss. Refrigerant leakage (RU) was reproduced by removing the charge from the refrigerant circuit. Chen and Braun [50] included non-condensable gas in the refrigerant circuit and refrigerant overcharge. To emulate the NON fault, a fixed quantity of dry nitrogen was added to the refrigerant circuit. The RO was implemented by adding charge to the system. Working with a water-to-water system, Comstock et al. [37] studied reduced condenser water flow and reduced evaporator water flow, which was reproduced by reducing the speed of the circulating pumps. Excess oil and defective expansion valves were also emulated. Armstrong et al. [47] based their research of FDD on the transitory electrical signals of a rooftop cooling unit. They tested some of the above-mentioned faults and added mechanical imbalance of fans and compressor liquid ingestion. Weights were placed on the tips of the fan blades to create imbalance and the suction port of the semi-hermetic compressor was injected with a mass of liquid to emulate liquid ingestion.

Xiao [59] and Hou et al. [60] focused on sensor faults in HVAC systems by applying bias in the measurement of critical sensors for the operation of the equipment. Luo et al. [45] differentiated between sensor bias, drift, precision degradation and complete failure. Namdeo et al. [61] focused on valve leakages, but differentiated between leakage at the suction or the discharge valve. In their study with heat pumps, Kim et al. [42] implemented compressor valve leakage but also included as the same type of fault a leakage in the 4-way valve. This is because the consequences of leakage at the compressor valves or at the 4-way valves are the same (hot gas to the suction line). Bonvini et al. [62] worked with a model of a real chiller plant and considered a more general description of faults, such as higher energy consumption than expected, a decrease in chiller efficiency or occluded condenser water valves.

5.2. Faults reproduced in virtual environments

The laboratory testing of real equipment is costly and time-consuming. In some cases, the faults tested can permanently reduce the performance of the heat pump or even leave it inoperable. Models for heat pumps and fault modeling are increasingly being used [63]. Fault modeling is used to validate FDD techniques under a variety of fault free and faulty conditions, including concurrent fault occurrence. Simulations can also provide greater insight into the overall effect of faults on the operation of a system, the interaction between concurrent faults the energy consumption and thermal comfort of buildings [63]. Several studies have engaged in FDD by applying heat pump simulations modeled with a grey-box and white-box approach [64–66].

For FO, FI, OMC and IMC faults, they use grey-box and white-box models to simulate the reduction of the air-flow. Whereas grey-boxes are used to reduce the airflow only across the heat exchanger [64,65], white-boxes consider the effect on the term UA, which is the product of the overall heat transfer coefficient (U) and the coil heat transfer surface area (A), when introducing a fouling factor [66] for FI and FO faults. Additionally, white-box models simulate fouling by formulating the degradation of the UA factor [67] or by modifying of the log mean temperature difference (LMTD) term in the heat transfer equation [68].

The VL fault is commonly simulated by reducing the refrigerant mass flow rate and the effect of mixing during suction on the refrigerant's thermodynamic properties is also accounted for [64]. The LL fault is simulated with an additional pressure drop in the liquid line [63,64]. Even though some authors consider that the NON fault accumulates within the condenser only [69], Cheung and Braun [64] assumed that it accumulates in the hot gas line and the vapor section of the condenser. Therefore, the fault is simulated by calculating the refrigerant and the non-condensable partial pressure and then solving the rest of the heat pump model by considering this influence [64].

Control sensor faults in HVAC systems are frequently modeled by considering both positive and negative offset values, particularly on readings of temperature sensors [68]. The fault modeling methods applied in the literature are summarized in Table 4.

Table 3- Summary of faults emulated under laboratory testing on real equipment.

Fault type	Emulation technique	Quantification	Fault level	Ref.
FO	Fan/pump speed reduced	% of air flow reduced	10-50	[39,53]
			30	[70]
			Unspecified	[71]
	Blocking the area of the heat exchanger with paper or cardboard	% of area blocked	14-39	[47]
			3-16	[27]
			10-30	[54]
			5-35	[42]
			10-40	[50]
OMC	Attaching weights to the fan to create mechanical imbalance	Unspecified	Unspecified	[47]
FI	Fan/pump speed reduced	% of air flow reduced	40	[70]
			10-30	[54]
			6.82-27.28	[50]
			10-50	[28]
	Blocking the area of the heat exchanger with paper or cardboard	% of area blocked	10-100	[47]
			5-31	[27]
IMC	Fan switched off			[28]
FI&IMC	Fan/pump speed reduced		5-33	[42]
VL	Opening the hot gas bypass valve a specific amount	% Reduction of refrigerant mass flow rate	8-56	[27]
			4-19	[54]
			3-11	[42]
			10-40	[50]
			Unspecified	[47,61,70,71]
NON	Adding dry nitrogen to the refrigerant circuit	% of the amount of nitrogen in the circuit at atmospheric pressure	20-50	[42]
		% of the total refrigerant mass	0.03-0.17	[50]
RO	Increase the refrigerant charge	% of refrigerant with respect to the nominal amount	20	[39,47]
			10-20	[55]
			10	[39,70]
			11-32	[27]
			10-30	[42,54]
			5-30	[50]
			Unspecified	[71]
RU	Decrease the refrigerant charge	% of refrigerant with respect to the nominal amount	20	[47]
			10-30	[42,54,55]
			10-50	[39]
			15	[70]
			11-32	[27]
			30-50	[39]
			5-30	[50]
			Unspecified	[71]
LL	Partially closing the needle valve on the liquid line	% of pressure drop increased in liquid line	5-19	[27]
			8-49	[54]
			1-20	[42]
			4.75-18.66	[50]
SEN	Adding a bias to the measurement value of a sensor	ºC added	-4 to 4	[46]
			-0.75 to 0.75	[45]
			1 to 8	[72]
			0.5	[59]
			-1.5 to 1	[60]
			Unspecified	[62]
	Adding a drift in the measurement value of a sensor	ºC/hour added	-0.1 to 0.1	[45]
Others	Compressor switched off			[71]
	Expansion valve malfunction	% of superheat temperature increased	40-130	[48]
	Short cycling			[47]
	Compressor liquid ingestion			[47]

Tables 3 and 4 show the fault levels used for fault testing and simulations. However, these levels may not be representative of real situations. While the literature assumes that the average reduction in airflow by fouling of the outdoor unit (FO) is 26%, the actual average for devices in the field is 1.3% [73]. The same occurs for NON fault, where the fault levels do not exceed 5% in field studies [36]. The tables also show that there is a lack of standardization of some of the faults, particularly for FO. However, in recent years, authors such as Yuill [35] have made an effort to homogenize fault emulation. Li and O'Neill [63] stated that faults can be simulated by using either a deterministic or a stochastic approach. Deterministic fault simulation assumes that faults take place throughout the simulation period while the stochastic approach considers only the probability that faults will occur.

Table 4 – Summary of fault modeling techniques for heat pumps. Percent values are defined with respect to fault free conditions. HX: heat exchanger. R*f*: fouling factor. UA: overall heat transfer coefficient times area.

Fault type	Simulation technique	Fault levels	Ref.
RO&RU	Charge variation	70-130%	[64,75]
FI&FO air side	*Grey-box:* Reduction of air flow rate by increasing friction factor or reducing fan speed. *White-box:*	0-55%	
	1) Modeling of UA introducing R_f	-	[66,76]
	2) Modeling UA degradation with degradation coefficient α	α Є [0,1]	[67]
	3) Regressions on air-side effective heat transfer and air pressure drop	-	[41]
	4) Reduction of UA based on changes of LMTD value	-	[68]
FI&FO water-side	*White-box:*	-	[66]
	same approaches 1) 2) and 4) of air-side HX fouling		
IMC&OMC	Reduction in fan speed	-	[67]
	Reduction in fan efficiency	-	
	Change of fan performance curve	-	
	Change of pump performance curve	-	[63]
LL	Increase in pressure drop across liquid line	0-3500%	[64]
NON	Change in partial and total pressures	0-20%	[64]
VL	Reduction of refrigerant mass flow	0-50%	[64,75]
	Effect of mixing on suction enthalpy		
SEN	Addition of offset	±0.5 K	[68]

In most studies, the researchers used a deterministic approach even though it is sometimes unrealistic. In the case of the RU due to a leakage of refrigerant, the fault level will increase with time, but the fault is emulated at a fixed level or in increasing steps. Otto et al. [74] proposed a stochastic method for studying the distribution of faults by using a probability density function. However, there is a lack of information on the evolution of those gradual faults and their incidence on heat pump systems.

Another key fact is the occurrence of simultaneous faults. This section has only considered isolated faults. Other authors have studied the option of simultaneous faults [27,70,77]. However, there are not enough data to determine which faults occur simultaneously in field equipment.

6. Instrumentation and feature selection for fault detection and diagnosis

6.1. Direct measurements and feature selection

In the development of FDD systems, a critical stage is the selection of sensors and features for analysis. A minimum number of direct measurements for domestic and non-domestic vapor compressor systems should be selected to reduce the cost of the instrumentation. Some studies in the literature use a significant number of measurements (in some cases almost one hundred sensors [60]) while other studies use very few (for example non intrusive load monitoring (NILM) [47]). Some studies have focused on the optimal number of sensors [78] using a genetic algorithm technique that maximizes FDD accuracy. One such study concluded that from an initial number of 64 sensors the optimal number was 8 and 24, with and without taking cost into account, respectively. Likewise, after analyzing the impact of different faults, Namburu et al. [79] concluded that the optimal set was 15 sensors from an initial set of 48 measurements. To select sensors, Han et al. [49] used a mutual information base method that discriminated between minimally redundant sensors.Then they used a genetic algorithm technique to obtain the optimal set of sensors by optimizing fault classification. The sensor set obtained by these authors for a water-to-water heat pump is shown in Table 5. In previous FDD studies, it is common to find a minimum of 7 direct measurements, among which are the refrigerant suction and discharge pressures and temperatures, the refrigerant temperatures at the inlet or outlet of the evaporator and at the liquid line, and the electrical power consumption of the compressor [55,80,81]. Most of these measurements were discussed in the study by Kim et al. [36] to derive a series of features that were sensitive to each of the faults mentioned in section 4.1. The most common features in FDD are the saturation temperatures at the condenser and evaporator, subcooling and superheating, electrical power consumption, the compressor discharge temperature and the coefficient of performance [36,39]. However, features are commonly sensitive to more than one fault. Decoupling techniques can avoid the use of features that are sensitive to different faults [80]. Decoupling requires the use of virtual sensors because many of these features cannot be measured directly [80].

Table 5 - Direct measurement sensors selected for fault testing studies in heat pumps. Thermo. + electr.: method with thermodynamic and electrical sensors.

Method	Heat pump	Sensors number	Direct measurement sensors	Ref.
Feature selection	water-to-water	7	T_{evapwo}, T_{condwo}, T_{dis}, T_c, PO_{feed}, m_{condw}, VE	[49]
Thermo.+ electr.	water-to-water	8	T_{wi}, T_{wo}, m_w, T_{dis}, T_{suc}, P_{dis}, P_{suc}, T_{ll}, T_{evapro}, W	[55]
Thermo.+ electr.	air-to-air	8	T_{ai}, T_{ao}, T_{dis}, T_{suc}, P_{dis}, P_{suc}, T_{ll}, T_{evapro}, W	[80]
Thermo.+ electr.	air-to-water	12	T_{suc}, P_{dis}, P_{suc}, P_{dis}, T_{ll}, Te_{vapri}, T_c, T_e, T_{indoor}, T_{amb}, W,I,V, E	[81]
NILM	air-to-air	5	Q, W, f, V, I	[47]

6.2. Virtual sensors

The implementation of virtual sensors enables the instrumentation costs of fault detection and diagnosis systems to be reduced. Virtual sensors are frequently based on low-cost direct measurements to characterize equipment operation and estimate virtual measurements indirectly. Most virtual sensors that have been proposed to monitor the fault behavior of building systems are based on first-principle and grey models [82]. The aim of these virtual sensorsis to provide information on system operation by indirect measurements (e.g. virtual sensors for refrigerant mass flow rate and pressures), while others are indicators of the state of the system in terms of faulty operation (e.g. virtual sensors representing the fouling level).

This section summarizes the main virtual sensors that are used in vapor-compression systems for the estimation of system state operation and performance. Table 6 shows a summary of the various virtual sensors that are reviewed in this section and the methodology they are based on.

Table 6 - Methods for virtual sensors in fault and diagnosis systems of heat pump units

Virtual sensor	Methodology	Input sensors needed	Validation	Ref.
Refrigerant charge	Charge variation	Tc, Te, $Tsuc$, Tll	<15 %	[84]
Refrigerant mass flow	Compressor map 6 or 10 coef. polynomial	Tc, Te, $Tsuc$, $Psuc$, f	3-19 % RMS Not recommended for: compressor failure	[85,86]
Refrigerant mass flow	Compressor map based on volumetric efficiency	$Pdis$, $Psuc$, $Tsuc$, $Tamb$	± 7%	[80]
Refrigerant mass flow	Energy balance	W, $Tdis$, $Pdis$, $Tsuc$, $Psuc$, f	<5 %RMS	[85]
Refrigerant mass flow	TXV device model	$Tsuc$, $Psuc$	1-10 % RMS	[85]
Refrigerant mass flow	EEV device model	EEV motor step, Pc, Pe, Tc	<6 %	[85]
Pressure sensors	Saturation pressures and pressure drops	Te, Tc	-5-9%	[90]
Compressor power	Compressor map ARI 10 coef. polynomial	$Psuc$, $Pdis$	± 5%	[80]
Compressor power	Compressor map 6 coef. polynomial	Tc, Te, f	± 4-8%	[86]
Compressor frequency	Frequency correction factors	$\dot{m}ref$	6-10% RMS	[86]
Air flow rate	Energy balance	$\dot{m}ref$, $Tsuc$, $Psuc$, Tll, Pll, $Taie$, $Taoe$, Φaie, Φaoe	-	[80]

6.2.1. Virtual refrigerant charge sensor

The levels of refrigerant charge are important for heat pumps to run efficiently. Field studies indicate that more than 50% of air conditioning units present significant deviations from optimal charges as a result of inadequate commissioning service and leakage [83].

The virtual refrigerant charge sensor proposed by Li and Braun [84] allows non-intrusive estimation of the refrigerant mass in the units. This virtual measurement of refrigerant mass (m_{total}) is obtained using four surface temperature measurements (condensing, liquid-line, evaporating, and suction-line temperatures), according to the following formula:

$$\frac{m_{total}-m_{total,rated}}{m_{total,rated}} = \frac{1}{k_{ch}}\left(T_{sc} - T_{sc,rated}\right) - \frac{k_{sh}}{k_{sc}}\left(T_{sh} - T_{sh,rated}\right) \quad (1)$$

where k_{sc} and k_{sh} are constants that depend on the condenser and evaporator geometries, respectively, T_{sc} is the subcooling temperature and T_{sh} is the superheating temperature. The subscript *rated* means that the values are the ones at the manufacturers' rating conditions. This virtual sensor method was validated by Li and Braun [84] by comparing refrigerant charge readings with those obtained with the virtual sensor. The validation yielded good accuracy under several normal and faulty operating conditions, with errors below 15%.

6.2.2. Virtual refrigerant mass flow meter

Although measuring the refrigerant mass flow rate is useful for system monitoring and fault detection, including a mass flow meter device in the in-built sensor system of a heat pump is not generally feasible because of associated costs and operational issues.

To address this limitation several refrigerant flow virtual sensor methods have been proposed in previous studies. Most of these methods are based on three techniques: 1) estimation method using manufacturer's data and polynomial regressions 2) an energy balance method based on the compressor map for power consumption, and 3) empirical correlations based on the operation of the expansion device.

The first approach, which is based on the ANSI/ARI Standard 540-2015, estimates the refrigerant mass flow rate with a 10-coefficient polynomial equation. Kim and Braun [85] proposed an equation to calculate the refrigerant mass flow rate ($\dot{m}_{ref}$) with the input of the evaporating saturation temperature (T_e), the condensing saturation temperature (T_c) and the density at the inlet of the compressor (ρ_{suc}). For fixed speed compressor heat pumps, this equation is given by:

$$\dot{m}_{ref} = \rho_{suc}(a_0 + a_1 T_c + a_2 T_c + a_3 T_e^2 + a_4 T_e^2 + a_5 T_c T_c + a_6 T_e^3 + a_7 T_e^3 + a_8 T_e^2 T_c + a_9 T_c^2 T_e) \tag{2}$$

Similarly, Kim et al. [86] proposed an expression for obtaining the mass flow rate with a 6-coefficient second order polynomial equation.
This method, when applied to a variable speed compressor heat pump takes the form of a second-order polynomial that accounts for the frequency, as given by the following expression:

$$\dot{m}_{ref} = \rho_{suc}(c_1(f - f_{rated})^2 + c_2(f - f_{rated}) + c_3)(b_0 + b_1 T_c + b_2 T_c + b_3 T_e^2 + b_4 T_e^2 + b_5 T_c T_c)_{rated} \tag{3}$$

where f is the compressor frequency and the subscript *rated* indicates rating conditions.
An alternative method using manufacturer's data for calculating the refrigerant mass flow rate is based on the volumetric efficiency [80]:

$$\dot{m}_{ref} = \eta_v \frac{NV}{v_{suc}} \tag{4}$$

where η_v is the volumetric efficiency, V indicates the displacement volume, v_{suc} is the refrigerant specific volume in the suction line and N is the number of suction strokes per unit time. In this approach, the volumetric efficiency is estimated from compressor map data. To do this, the 10-coefficient polynomial in the ANSI/ARI Standard 540-2015 [87] can be used, but other expressions for estimating the volumetric efficiency with the discharge and suction temperatures and pressures as input are also available [88].

A second method for calculating the refrigerant mass flow rate applies an energy balance that is valid for both fixed speed and variable speed compressors. It is given by:

$$\dot{m}_{ref} = \frac{W(1-\alpha_{loss})}{h_{dis}(T_{dis}, P_{dis}) - h_{suc}(T_{suc}, P_{suc})} \tag{5}$$

where α_{loss} is the compressor heat loss ratio, $h_{dis}(T_{dis},P_{dis})$ and $h_{suc}(T_{suc},P_{suc})$ are the enthalpies at the discharge line and suction line, respectively, and W is the power consumption of the compressor. The heat loss ratio can present a value below 5% for fixed speed heat pumps but it may be higher in faulty conditions or for variable speed compressors at low speed. Li and Braun [80] proposed a set of empirical expressions to calculate heat losses under a variety of operating conditions, using the discharge and suction pressures and temperatures and the compressor frequency as input.

A third method for calculating the flow rate uses the operation principle of expansion devices to control the refrigerant mass flow rate and decrease the refrigerant pressure. With this method, the mass flow rate is calculated using semi-empirical models for TXV and EEV devices. In the case of a TXV, an empirical model is proposed that assumes that the mass flow rate is a linear function of the area at the valve opening. This leads to the following equation [85]:

$$\dot{m}_{ref} = [a_3(P_{sat}(T_{suc})^2 - P_{suc})^2 + a_4(P_{sat}(T_{suc}) - P_{suc}) + a_5]m_{max} \tag{6}$$

where m_{max} is the mass flow rate of refrigerant for the expansion device in full open position and $P_{sat}(T_{suc})$ is the saturation pressure for the temperature of the suction line. The maximum flow rate depends on the valve pressure drop and orifice size and can be determined empirically [89].

For the case of an EEV device, Li and Braun [84] used the same principle to propose a semi-empirical model that assumes that the mass flow rate is proportional to the flow area, which is dependent on the needle position controlled by the motor of the expansion device.

Kim and Braun [85] analyzed the three refrigerant mass flow virtual sensor methods described above by comparison with experimental data and fault-free and faulty conditions. In the first method, based on polynomial equations and manufacturer's data, the RMS error was generally below 3% for normal and faulty operation except when there was a fault with the compressor in which case the RMS error was higher (19%). Li and Braun [80] showed that this method may not be applicable if the compressor has already deteriorated with respect to the initial manufacturer's data. Validating the method with the energy balance equation, showed RMS errors below 3-10% in normal and faulty conditions for both fixed and variable speed compressor heat pumps [85]. As far as the method based on the operation principle of expansion devices is concerned, overall RMS errors are about 1-3% of the actual mass flow rate for the TXV-based model with significant errors of about 10% in the case of refrigerant undercharge. For the case of the virtual flow sensor based on the EEV device, the RMS error was found to be lower than 6% for fixed and variable speed compressor heat pumps [85].

6.2.3. Refrigerant circuit pressures

Measuring the pressure at different points in the refrigerant circuit is useful for monitoring the performance of the vapor compression system. Nevertheless, pressure sensors are costly and they are difficult to install as the refrigerant circuit needs to be evacuated and refilled. Also, the installation of pressure sensors at the service ports of the circuit may cause a refrigerant leak over time [90]. It is therefore desirable that pressure sensors be replaced with virtual ones based on low-cost direct measurements of temperature [90].

With this aim, Li and Braun [90] developed a method to estimate the main pressure points in the circuit (i.e. saturation pressures at condenser and evaporator, discharge and suction pressures, and liquid line pressure) using surface temperature measurements only. They estimate the evaporation and condensation pressures from the direct saturation temperature measurements. The accuracy of these measurements is critical to the method and the surface sensors should be located carefully. These sensors should be placed in direct contact with the surface at points where the refrigerant is sure to be in saturation state. The saturation pressures are then determined from these temperature measurements using refrigerant state correlations. Once the saturation pressures are known, pressures at other locations of the refrigerant circuit are estimated by calculating the pressure drop across the circuit. Under the assumption that the pressure drop in the heat exchangers is caused mainly by friction, the limits of the pressure drop are given by [90]:

$$\alpha^2 \Delta P_{rated} < \Delta P < \beta^2 \Delta P_{rated} \quad (7)$$

where α and β are limit coefficients for the pressure drop in rating conditions, ΔP is the actual pressure drop and ΔP_{rated} is the pressure drop in rating conditions. Common values for the coefficients α and β are 0.7 and 1.23, respectively, for high volumetric efficiency compressors. For low volumetric efficiency compressors, the values of these coefficients are estimated to be 0.41 and 1.35, respectively [90].

The comparison of estimations from virtual pressure sensors and real pressure measurements demonstrates that the virtual sensors exhibit accuracies that are similar to the direct sensors and that they can be used for detecting faults [90]. Error measurements have been found to be between ±1.6% for the pressure in the suction line and ±4.4% for the liquid line and discharge pressures in air conditioning systems. Higher errors of -5 to 9%, however, have been found in a heat pump system [90]. When applied to a fault diagnosis system, these sensors did not cause a loss in sensitivity nor false alarms when studying valve leakage faults. On the other hand, outdoor heat exchanger fouling revealed 8% of missed diagnosis and 1% false alarms [90]. This shows that some improvement is also necessary for virtual pressure sensors, which is still a relatively robust method that can be integrated into a fault and diagnosis system.

6.2.4. Compressor power consumption

The measurement of electrical power consumption is critical if the performance and the efficiency of heat pumps is to be monitored but it may require the installation of relatively expensive sensors. A well-known virtual method for estimating power consumption uses compressor map manufacturer data, as defined in the ANSI/ARI Standard 540-2015 [87], to calculate power consumption using a 10-coefficient polynomial that depends on discharge and suction dew point temperatures. Although the coefficients used in this method are for rating conditions, it has been shown that the approach is still valid when the compressor deviates from them [80] and even when the compressor has degraded as a result of compressor valve leakage. A comparison of calculated and measured power consumption by a vapor compression system under faulty operation (compressor leakage, heat exchanger fouling, liquid-line restriction, and refrigerant undercharge) indicates that there is a 5% difference between measurements and estimations, which demonstrates that this method is still valid under fault conditions.

Kim et al. [86] proposed an alternative 6-coefficient polynomial expression to estimate the electrical power consumption from the evaporating and condensing temperatures. To use this expression, a correction is applied for compressor speeds other than rated conditions as in equation 3. Deviations between estimations with this method and direct measurements have been found to be between 4 and 8% in heating and cooling modes [86].

6.2.5. Outdoor heat exchanger air flow sensor

The result of a faulty fan and fouling of the outdoor heat exchanger of a heat pump reduces the air flow. To monitor this effect, Li and Braun [80] proposed a virtual air flow meter based on the application of an energy balance for the evaporator in heating mode such as:

$$\dot{V}_{ea} = \frac{\nu_{ea}\dot{m}_{ref}\left(h_{suc}(P_{suc},T_{suc}) - h_{ll}(P_{ll},T_{ll})\right)}{h_{aie}(T_{aie},\phi_{aie}) - h_{aoe}(T_{aoe},\phi_{aoe})} \quad (8)$$

where $\dot{V}_{ea}$ is the air flow rate, ν_{ea} is the air specific volume, h_{suc}, P_{suc} and T_{suc} are the refrigerant enthalpy, pressure and temperature in the suction line, h_{ll}, P_{ll} and T_{ll} are the refrigerant enthalpy, pressure and temperature in the liquid line, h_{aie}, T_{aie}, ϕ_{aie} are the enthalpy, temperature and relative humidity of the inlet air, respectively, and h_{aoe}, T_{aoe}, ϕ_{aoe} are the enthalpy, temperature and relative humidity of the outlet air.

It has been demonstrated that these virtual sensors can detect significant reductions in the air flow rate caused by evaporator fouling [80].

7. Conclusions and insights into future research

This paper reviews faulty heat pump behavior in field and laboratory studies, along with methodological aspects including instrumentation with virtual sensors and simulation techniques for FDD methods. The methodologies have been evaluated in terms of the sensor requirements, fault types and the level of the faults with new contributions:

- Novel comparison of reported performance ratings of different FDD methods is done in section 3 and summarized in Table 1.
- The effects on thermal capacity and COP of different types of heat pump faults are explained in section 4 and put together in Table 2 differentiating between heat pump types and compressors.
- An acute description of different fault testing methods is made in section 5 and summarized in Tables 3 and 4 for laboratory and simulations, respectively.
- Description of the useful features for FDD is made in section 6.1 and showed in Table 5.
- Virtual sensors for FDD are described in section 6.2 and condensed in Table 6.

A detailed analysis revealed several areas of improvement of different aspects and the following insights into future research have been identified:

- Standard performance assessment methods need to be used to compare the efficiency of FDD methods so that the appropriate methods for specific systems can be selected.
- Updated field data on heat pump performance is scarce and further research is needed to characterize the faulty behavior of heat pumps in real installations in order to determine the frequency, evolution and incidence of faults.
- The methods applied to reproduce fault behavior in simulations and laboratory tests are not representative of the actual occurrence and level of faults and studies are not homogeneous in terms of the fault levels and methodologies they use. New methods that are more representative of realistic conditions need to be developed to account for the gradual generation and probabilistic nature of fault behavior in heat pumps.
- Virtual sensors are cost-efficient but, to date, they have been applied only in a limited number of FDD studies. Further research is needed to analyze the performance of fault diagnosis systems that use these alternative sensors and to identify potential limitations and areas of improvement.

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9. CRediT authorship contribution statement

I.Bellanco: Conceptualization, Writing – Original draft preparation. **E. Fuentes:** Methodology, Conceptualization and writing. **M. Vallès:** Supervision, review. **J. Salom**: Conceptualization, review, lead researcher of TRI-HP project.

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