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An Experimental Approach for the Dynamic Investigation on Solar Assisted Direct Expansion Heat Pumps

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In this paper, a direct expansion solar assisted heat pump (DX-SAHP) is investigated as one of the possible low environmental impact solutions to the domestic thermal energy demand, even if due to some technological and control strategy limitations, the DX-SAHP systems do not have a competitive role in the actual renewable energy market. An instrumented prototype of a small DX-SAHP system is here designed and presented, within the context of developing safe and affordable sustainable energy applications, to determine the best control strategies and to measure the energy performance indices with good reliability and accuracy. The property calculation tool and the main issues related to the performance measurements and calculations are reported, with an estimation of the measurement error. The results of test series evidence that the apparatus can fulfil significant thermal loads, with a coefficient of performance (COP) up to 6.

1. Introduction

Applications of clean energy power systems are evolving in a wide range of industrial sectors and different geographical locations while, considering the depletion of fossil fuels, a broad debate has long been opened within both the scientific and political communities concerning the strategies to be adopted for sustaining the increasing world energy demand (Nemet et al., 2012) and side-on effects of potential solutions (Milazzo et al., 2013). Additionally, the whole problem of energy supply and energy demand reduction has a wide interdisciplinary character owing to its social, environmental (Fabiano et al., 2012) and safety implications (Palazzi et al., 2014) in both industrial and domestic-scale systems. Solar energy based systems may play a positive role in integrating and phasing out fossil fuels, so reducing the risk connected to different accident scenarios (Palazzi et al., 2013) and consequently process accidents and near-misses (Fabiano and Currò, 2012). Solar energy is attracting more and more scientific interest in several processes, e.g. water splitting by photo-electrochemical reactors (Hankin et al., 2014). Within this broad context, although the solar assisted heat pump concept has been known since more than fifty years (Sporn and Ambrose, 1956), only in the 90's advanced technological characteristics like variable capacity compressor (VCC) were available for small refrigeration plants (Chaturvedi et al.,1990). Additionally, more sophisticated prediction and control techniques were recently developed, including moving phase interface modelling (Dong et al., 1998) and proper state-space representation (Schurt et al., 2009), making it possible to apply this inventive solution effectively. Due to these technological and control strategy limitations, the Direct Expansion Solar Assisted Heat Pump (DX-SAHP), also known as Integrated Solar Assisted Heat Pump (ISAHP) technology, does not have a competitive role in the actual renewable energy market, despite several solar-assisted heat pump configurations were investigated since the 80's, as from the comprehensive review by Ozgener and Hepbalsi (2007). Several DX-SAHP devices were presented in the scientific literature as swimming pool heating source (Tagliafico et al.,2012[a]), commercial small refrigeration appliances (Omojaro and Breitkopf, 2013), or as the evaporator, by using a roof-integrated solar collector (Yang et al., 2011). The results of previous studies usually suggest that proper design and optimized control criteria are recommended to exploit at best this renewable energy technology (Scarpa et al., 2011). Easy-to-handle thermodynamic models relying upon both a steady-state

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(Scarpa et al., 2013[a]) and a dynamic (Scarpa et al., 2013[b]) analysis of a DX-SAHP system can assume a basic role in multi-variable multi-control refrigeration systems, where at least the compressor capacity (variable speed compressor) and the expansion valve characteristics (electronic expansion valve) can be controlled independently from each other. This technology, only recently introduced in the market, is not yet duly exploited in current refrigeration and heat pump systems, so that a reliable experimental modelling method is required in investigating the most suitable configuration under different technical and environmental conditions. This paper aims at presenting an experimental approach as an innovative strategy to sustainable and safe solar energy applications, consisting of an instrumented patented DX-SAHP system (De Filippis et al., 2010) (DX-SHAP machine, water loop, acquisition system, properties calculation program) allowing to reliably measure the working parameters of the heat pump and to start the preliminary work needed to design an “embedded” control system using modern IC technologies. The experimental set up will be used to analyse the best control strategy able to maximize the energy savings and system efficiency, for given user thermal energy demand and environmental working conditions. The designed set up was realized in order to analyse the real capabilities of this technology and with the purpose of validating the simulation results. Preliminary results are then presented, showing how the DX-SAHP system can be well adapted to produce medium temperature (around 323 K) hot water in winter season (i.e. at very low external ambient temperatures), keeping good efficiency of the heat pump system (heat pump COP within the range 4.5 -6.5), with collector efficiency almost double than traditional solar flat plate panels and connected overall plant intensification.

2. Experimental set-up

The DX-SAHP machine consists of a traditional inverse (refrigeration) cycle (Scarpa et al., 2011) equipped with a Variable Capacity Compressor (VCC) and an Electronic Expansion valve (EEV). A consistent modelling approach for a thermosyphon heat-exchanger was proposed by Chyng et al., (2003). The project is primarily designed to realize a high-performance plant solution, using existing, commercial components, so that the cost and number of special components have been kept as low as possible. The experimental apparatus (see Figure 1) comprises the DX-SAHP machine linked to a thermostatic bath operating as a virtual user boiler/heater, regulating the temperature of the water loop which is coupled to the condenser of the refrigeration system. The thermostatic bath controls the temperature with an accuracy of 0.1 K and is equipped with a 25 L tank and a circulation pump. The solar collector, where the refrigerant boils, is realized with a bare Al flat plate of about 1 m^2 (0.97 x 1.03 m) surface. A copper coil is wounded under the panel, which is the refrigerated plate. The surface of the panel has been treated with a high absorptivity black coating. The compressor, which is a 6 cm^3 VCC (variable speed device) HBP (High Back Pressure) hermetic apparatus, is adapted for working in a higher temperature range with respect to its design working parameters. The condenser is a coaxial pipe heat exchanger, with water circulating in the outer section loop, operating with an additional flow path through a specific sub-cooler water tank. The whole structure is well insulated with expanded polypropylene. To easily match the user demand with the system (solar) working conditions, a small 20 L water tank was used as a sub-cooler section, before the refrigerant reaches the expansion valve. In this way the EEV is fed with a well-established liquid flow rate, assuring a good repetitiveness of the control operation. The valve installed in the prototype is a Pulse Modulated Electronic Expansion Valve. This type of valve is easily adaptable to different temperature and pressure working operation ranges, depending on the heat transfer rates needed and the temperature span between the evaporator and the condensing unit. The valve has a duty cycle of 6 s. The external water loop is realized with insulated rubber pipes (1/2” diameter). The inlet temperature (in the DX-SAHP) of the water is fixed by the thermostatic bath. A system of artificial lighting controlled via computer produces a time-varying radiation on the prototype’s panel, with a colour temperature of ca. 2300 K and a maximum mean irradiation on the panel surface of 800 W m^{-2}. Even if the radiation spectrum is moved towards the infrared region with respect to solar radiation, the system is suitable for the study purposes, allowing great flexibility in the experimentation procedure, independently of weather or sun shine conditions. A phase angle controller modulates the power supplied to 6 halogen 400 W lights, allowing the system to reproduce any given time-depending radiation law. The lamps have been installed in two racks at 20 cm from the panel surface. A radiation power map has been realized and the artificial solar system has been calibrated using an insulated camera, a solar meter and a steel plate equipped with thermocouples. The accurate calibration has been done for different light powers and distance from the plate. The results of the calibration were used to correlate the actual mean irradiance on the plate to the lighting power of the lamps and to a spot-measure performed by means of a small solar meter, 5x7 cm in surface, placed just over the panel. A view of the experimental set-up and a layout of the plant configuration are reproduced in Figure 1. In order to measure and collect all data required for calculating the properties and the performance parameters of the DX-SAHP, an acquisition system has been realized together with an ad-hoc designed data acquisition board.

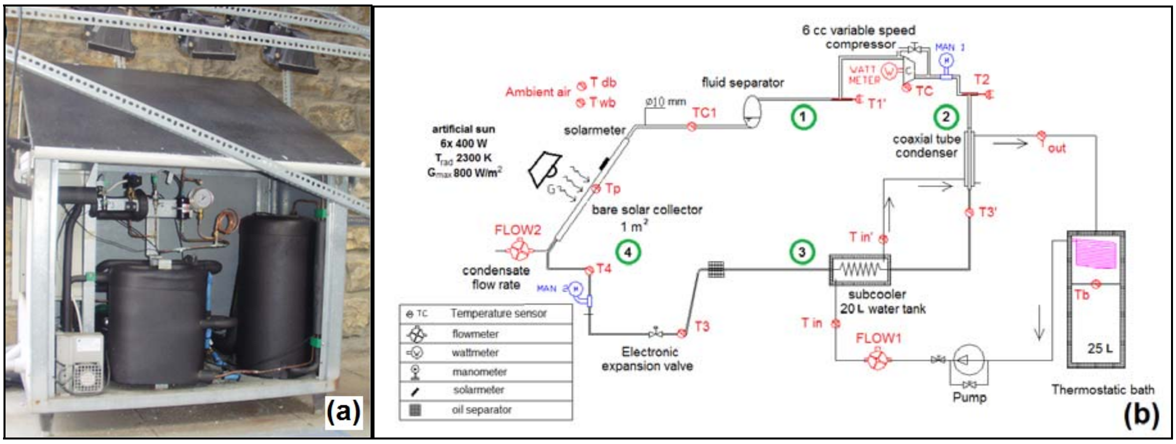

Figure 1: The DX-SAHP experimental set-up developed at DIME /University of Genoa. (a) Final configuration with the lighting system. (b) Conceptual scheme of the set-up.

As already mentioned, the second task of the project is to develop the apparatus with an embedded control system, including also the refrigerant thermodynamic states calculation and an automatic set-point optimization. For this reason, some limitations in the software development have been taken into account, trying to avoid external software references which could be inserted with difficulty in a small embedded electronic device. The system was instrumented with 14 NTC sensors for the temperature measurements (6 on the refrigerant circuit, 2 on the panel, 1 on the compressor shell, 3 for the water loop, 2 for the ambient air condition), 2 analogical and 2 piezoelectric manometers, 1 flow meter for the water loop, 1 solar meter and 1 wattmeter for the measurement of the electrical consumption of the compressor. Ambient air humidity has been measured by means of dry-bulb and wet-bulb temperature measurements. The refrigerant flow rate, for assigned compressor characteristics, has been calculated using the global energy balance described by Eqs (1), (2), (3) and (4). Figure 1 (b) depicts the structure of the experimental set-up, including the locations of the relevant sensors. A dedicated acquisition board collects all the raw signals and converts them in 0-5 V electrical signals, which can be easily translated in numerical values of the physical measurements by means of a proper calibration. All the signals collected by means of the electronic board have been sent, by means of a standard USB data acquisition board, to a linked “target” PC. The acquisition tool, the properties calculation tool and the control system have been gathered in one Matlab-Simulink program sheet. The Simulink acquisition tool converts the 0-5 V signal into the data through the calibration curves and the measurements are sent both to the control system and to the properties calculation sub-program. All sensors have been calibrated and the measurement errors were estimated by means of the standard deviation σ values.

3. System modelling

The signals are analysed using a program developed in Matlab-Simulink. The program, based on thermodynamic cycle calculations and a lumped dynamic system simulation, determines the thermodynamic working condition of the DX-SAHP and it is able to send a proper set-point control signal to the variable compressor speed and the electronic expansion valve modulation. In the meantime, all heat and power rates, including performance indexes of the system, are evaluated and stored. In order to calculate all properties for the refrigerant used as thermodynamic fluid, such as enthalpy (point 1,2,3,4, circled numbers in Figure1(b)), entropy (point 1,2) and density (point 1), an interpolation has been done on a matrix of values (300) calculated with the well-known NIST library and they have been inserted into a lookup table block. With the purpose of converting the Simulink program by a proper compiler and of subsequently transferring it into an embedded processor, the lookup tables were used instead of the common state function equations for the selected fluid to avoid assigned operations (like square roots or logarithms). The sub-program included in the aforementioned software has been designed in order to minimize the approximation error for every working condition. The maximum approximation error occurring when enthalpy is evaluated by means the lookup table (in particular near the vapour saturation curve) is smaller than 0.8%. The pressure calculation is carefully verified by comparing its value with the data acquired by a pressure transducer at point 4. The usual performance parameter COP and the heat and work transfer rates are evaluated by Eqs (1), (2), (3) and (4), (where subscripts reference is as follows: 1 = compressor inlet; 2 = compressor outlet; 2’ = isentropic compressor outlet; 3 = condenser outlet; 4 = evaporator inlet):

$$P_{comp} = \dot{m}_{comp}(h_2 - h_1) \quad (1) \qquad q_{cond} = \dot{m}_{comp}(h_3 - h_2) \quad (2)$$

$$q_{evap} = \dot{m}_{comp}(h_1 - h_4) \quad (3) \qquad COP = \frac{Q_{cond}}{P_{comp}+P_{aux}} \quad (4)$$

Isentropic and volumetric efficiencies of the variable speed drive compressor have been calculated using the performances map data available by the compressor manufacturer. The assumed functional dependences of these parameters on rotational speed and outlet to inlet pressure ratio follow as (Tagliafico et al., 2012[b]):

$$\dot{m}_{comp} = \rho_1 v_c \frac{\omega}{60} \eta_v \quad (5) \qquad \eta_c = (h_2 - h_1)/(h_2' - h_1) \quad (6)$$

$$\eta_v = 0.932 - 0.087\left(\frac{\omega}{3000}\right) - 0.0162\left(\frac{p_2}{p_1}\right) \quad (7)$$

$$\eta_c = 0.65 - 0.087\left(\frac{\omega}{3000}\right) - 0.00285 - 0.000415\left(\frac{p_2}{p_1}\right)^2 \quad (8)$$

An important issue of this study relies upon the estimation of the measurement errors in the acquisition system (Table 1) that may affect the properties calculation. We stress that the reliability of the measurements depends not only on the sensors accuracy, but also on their proper location in the circuit. It is noteworthy noting that the temperature and fluid flow rate measurements at the compressor outlet require a careful attention.

Table 1: Error prediction within the measurements range for the given data acquisition system.

Measurement	Symbol	Instrument	Max Error
Temperature	T	NTC	0.4 K
Pressure	p	Digital pressure transducer	$2.10 \cdot 10^3$ Pa
Radiation	G	Solar meter	18 W
Humidity	RH	Wet bulb hygrometer	0.3 %
Power consumption	P	Watt meter	1.2 W
Water flow rate	Q_w	Flow meter	0.3 L h^{-1}

The first problem is caused by the fact that the temperature sensors are placed upon the refrigerant pipe (insulated) and not inside the refrigerant stream. As a consequence, a correct estimation of the inner fluid temperature would require an accurate determination of the overall wall-fluid heat transfer coefficient (Reverberi et al., 2013). Considering the thermodynamic system, the conductivity of the copper tube cannot be neglected. Furthermore, the temperature of the compressor shall be significantly different from the refrigerant temperature at point 2 (normally set between the inlet and outlet temperature). The errors affecting the temperature measurements at point 2, calculated by numerical simulations, were significant and they required a further ad-hoc estimation procedure. In order to estimate the maximum error for h_2, its value has been compared with the results obtained by Eqs (5) and (6). The same analysis was performed comparing the experimental value, q_{cond}, with the calculated value based on the energy balance according to Eqs (10) and (3). In addition to the aforementioned error sources, it is worth remembering that the calculation of the heat transfer rates is equally affected by the error in the mass flow rate measurement, namely by the approximation introduced by the volumetric efficiency, according to Eq (7).

$$P_{comp} = \dot{m}_{comp}(h_2 - h_1) \quad (9)$$

$$-q_{cond} = \dot{m}_{comp}(h_2 - h_3) = \dot{m}_{H_2O}\, C_p(T_{out} - T_{in}) \quad (10)$$

We conjectured that the use of thermocouples inserted inside the refrigerant flow and a flow meter installed inside the refrigerant loop might be beneficial in order to reduce the errors listed in the above mentioned table. This technical solution has been tested, but the device was blocked by oil residuals coming from the compressor. Table 2 summarizes the results of error prediction on the estimation of the main properties of the DX-SAHP system, calculated according to the Superposition of Errors Theorem (Leaver and Thomas, 1975).

Table 2: Global error prediction on the overall heat balance (within the application measurements range).

Measurement	Max error
q_{cond}	18.7 %
q_{evap}	3.6 %
P_{comp}	14.1 %

4. Results and discussion

Preliminary pilot tests were carried out to analyse the DX-SAHP capabilities in terms of performances and working conditions limits and some reliability results are shown in Figure 2. Entering into details, Figure 2(a) shows the heat flux for winter standard environmental climate in Genoa (Italy). Under these conditions, the DX-SAHP machine can be integrated by a water heater keeping down the condenser temperature working condition of the machine. Preliminary results show that when a proper refrigeration capacity and expansion valve modulation are settled, COP values up to 6 can be achieved steadily. Available data prove that the values are reliable, despite being affected by the estimated errors, and all the derived thermodynamic and energy balance calculations are consistent with the physical constraints. Also due to the different winter environmental conditions, the attained coefficient of performance values are slightly higher than the ones reported by Yang et al, (2011) showing average COP of the heat pump system equal to 2.97 and a maximum around 4.16. Checks of energy balance equations on refrigeration side compared to water loop side give results in agreement within a ± 20% tolerance (Figure 2(b)). Owing to the reliable measurement apparatus here developed, it will be possible to verify different control strategies applied on the DX-SAHP machine during dynamic operation. Right now, it can be affirmed that the input power needs to be modulated as a function of the external temperature and, particularly, as a function of the incident solar radiation. As amply known, when considering a standard heat pump system, device performance can be quantified by the COP; the higher the COP value the lower the electricity used for the same user thermal demand. Conversely, when it comes to DX-SAHP, COP and solar collector efficiency (both depending on panel temperature) must be taken into account. Optimal behaviour can only be achieved through a balance between two conflicting requirements: on the one hand, a low panel temperature, next to that of the environment, to reduce the heat losses and then use a greater fraction of the radiation available solar; on the other hand, a high panel temperature, next to that of the condenser, to achieve high COP values, thus reducing the need for electric power. Following the pilot test series, a further step, will be devoted to investigate system performances under a wider range of operating conditions in order to develop an optimal and reliable control strategy.

5. Conclusions

An experimental apparatus of an Integrated Solar assisted heat Pump has been presented, together with the data acquisition and control system implemented. The main issues related to the refrigerant fluid flow estimation ad the measurements errors have been described and their influence on the performance indexes calculation has been analysed. Some preliminary tests have been carried out showing the reliability of the experimental set up, even if a ±20 % uncertainty is still present in the global COP calculations and energy balance between the heated water-side and the heat pump fluid-side calculations. The apparatus aims at characterizing this plant configuration, in order to identify the best control strategy. This work has a twofold purpose: the first one is focused on testing different system control strategies in order to select the optimum one. The second one refers to the identification of the energy performances parameters of the machine to accurately set up the control and acquisition system embedded in the DX-SAHP. The focus of this research is the realization of a reliable Integrated Solar assisted heat Pump (DX-SAHP) capable to operate with maximum energy performance indexes in order to allow the coverage of the user loads even under critical operating conditions, like night or winter period, preserving all the operating and technical constraints (e.g. panel frosting, overpressure for high temperature in the compressor).

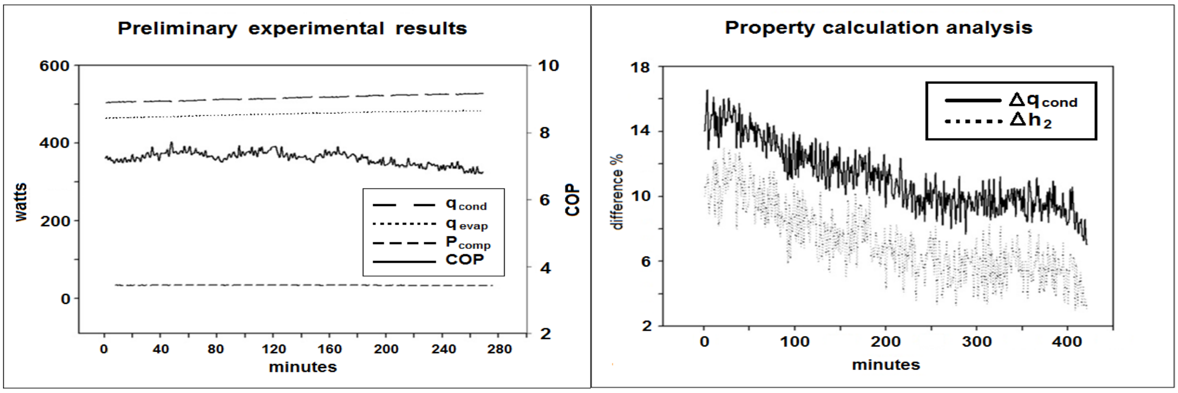

Figure 2: (a) Preliminary experimental results obtained at following conditions: G = 200 W m^{-2}, T_{amb} = 278.6 K, T_{user} = 303 K, ΔT_{H2O} = 6.8 K. (b) Percent differences among values of enthalpy and condenser heat flux calculated in differ ways.

Nomenclature

c_p	specific heat, kJ kg^{-1} K^{-1}	T_{amb}	environment temperature, K
G	lamp radiation, W m^{-2}	T_{user}	thermostatic bath temperature, K
h	enthalpy, kJ kg^{-1}	V_c	compressor volume, m^3
$\dot{m}$	refrigerant mass flow rate, kg s^{-1}	ΔT_{H2O}	inlet – outlet water temperature, K
P_{comp}	compressor power, W	η_c	isentropic efficiency, -
P_{aux}	auxiliary power, W	η_v	volumetric efficiency, -
q_{cond}	condenser heat flux, W	ω	compressor speed, rpm
q_{evap}	evaporator heat flux, W	ρ	density, kg m^{-3}

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