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Exergetic Performance Assessment of a Ground Source Heat Pump Drying System

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ABSTRACT

In evaluating the efficiency of heat pump (HP) systems, the most commonly used measure is the energy (or first law) efficiency, which is modified to a coefficient of performance (COP) for HP systems. However, for indicating the possibilities for thermodynamic improvement, energy analysis is inadequate and exergy analysis is needed. This study presents an exergetic assessment of a ground-source (or geothermal) HP (GSHP) drying system. This system was designed, constructed and tested in the Solar Energy Institute of Ege University, Izmir, Turkey. The exergy destructions in each of the components of the overall system are determined for average values of experimentally measured parameters. Exergy efficiencies of the system components are determined to assess their performances and to elucidate potentials for improvement. COP values for the GSHP unit, and overall GSHP drying system are found to range between 1.63-2.88 and 1.45-2.65, respectively, while corresponding exergy efficiency values on a product/fuel basis are found to be 21.1 and 15.5% at a dead state temperature of 27^{o}C, respectively. Specific moisture extraction rate (SMER) on the system basis is obtained to be 0.122 kg/kWh.

1. INTRODUCTION

To reduce carbon dioxide emissions and thereby protect against global warming, the effective and efficient use of energy, including the recovery of waste heat and the application of renewable energy, should be promoted. A heat pump (HP) system can contribute to this objective, normally delivering more thermal energy than the electrical energy required operating it (Okamoto, 2006).

Drying is an energy intensive operation that easily accounts for up to 15% of all industrial energy usage, often with relatively low thermal efficiency in the range of 25–50%. Thus, to reduce energy consumption per unit of product moisture, it is necessary to examine different methodologies to improve the energy efficiency of the drying equipment (Chua et al., 2001). A significant portion of global energy consumption is attributable to domestic and industrial heating and cooling. Heat pumps are advantageous and widely used in many applications due to their high utilization efficiencies compared to conventional heating and cooling systems. There are two common types of HPs: air-source heat pumps (ASHPs) and ground-source (or geothermal) heat pumps (GSHPs).

Exergy analysis is a very useful tool, which can be successfully used in the design of an energy system and provides the useful information to choose the appropriate component design and operation procedure. This information is much more effective in determining the plant and operation cost, energy conservation, fuel versatility and pollution. Bejan (1982) pointed out that the minimization of lost work in the system would provide the most efficient system. Moreover, Bejan (1988) and Szargut et al. (1988) emphasized that the effect of operating conditions on the system efficiency was much stronger for lost-work analysis than it is for the heat balance analysis. This explanation is required to determine the inefficient process, equipment, or operating procedure during drying.

In the recent years, exergy analysis has been widely used for the performance evaluation of thermal systems. By using exergy analysis method, magnitudes and locations of exergy destructions (irreversibilities) in the whole system are identified, while potential for energy efficiency improvements is introduced. One of the thermal systems is a HP dryer, which is a combination of a HP unit and a dryer, while it has been used in many drying applications.

As for as some recent studies conducted on exergy analysis of HPs are concerned, Bilgen and Takahashi (2002) presented exergy analyses of heat pump-air conditioner systems and derived an energy-based coefficient of performance, their optimum values, and an exergy-based efficiency and coefficient of performance. Sarkar et al. (2005) studied on exergy analysis and optimization of a transcritical carbon dioxide-based HP cycle for simultaneous heating and cooling applications. Ma and Li (2005) evaluated the performance of a HP system with an economizer coupled with a scroll compressor and derived expressions for exergy loss and efficiency. Hepbasli (2005) carried out a thermodynamic analysis of GSHP systems for district heating and derived mass, energy, entropy and exergy balance relations for them. Ceylan et al. (2007) investigated exergetically drying of poplar and pine timbers using air-source HP dryer. The timbers were dried from 1.28 kg water/(kg dry matter) and 0.60 kg water/kg to 0.15 kg water/(kg dry matter) at 40°C dry bulb temperature, 0.8 m/s air velocity. The exergy efficiency values on a ratio of output/input basis were found to vary from 40 to 90%. The values for coefficient of performance (COP) and specific moisture extraction rate (SMER) were 1.87 and 1.86, and 0.188 and 0.243 for pine and poplar, respectively.

The present work differs from the previously conducted studies on exergetic analysis of HP drying systems as follows: (i) it consists of a vertical GSHP drying system, which is exergetically analyzed as a whole for the first time to the best of the authors' knowledge, while HP drying systems assessed using exergy analysis method in the open literature are mostly air-source types. (ii) it includes an investigation of some thermodynamic parameters, such as relative irreversibility, fuel depletion rate, productivity lack and exergetic factor as well as exergetic improvement rate. (iii) it calculates the exergetic COP values for the GSHP unit and whole system. These were the motivation behind the present study. In this regard, energy and exergy analyses of this system, in which laurel leaves are used as a product being dried, are performed for performance evaluation purposes. Exergy losses for each component of the system are identified, while the potential for efficiency improvements is presented. This work also aims at revealing insights that will aid investigators, designers and operators of such systems.

2. MATERIALS AND PROCEDURE

2.1 Experimental Setup

Drying experiments were performed in a vertical GSHP drying system (or a GSHP dryer) designed, constructed and tested in the Solar Energy Institute, Ege University, Izmir, Turkey. Figure 1 shows a schematic diagram of the system investigated, while its outside view is illustrated in Figure 2. This system consists of mainly three separate circuits, namely: (I) the ground coupling circuit (brine circuit or water–antifreeze solution circuit), (II) the refrigerant circuit (or a reversible vapor compression cycle) and (III) the drying cabinet (chamber) circuit (air circuit). The main components of the HP system are an evaporator, a condenser, a compressor and an expansion valve. To avoid freezing the water under the working condition and during the winter, a 10% ethyl glycol mixture by weight was prepared. The refrigerant circuit was built on the closed loop copper tubing. The working fluid was R-22.

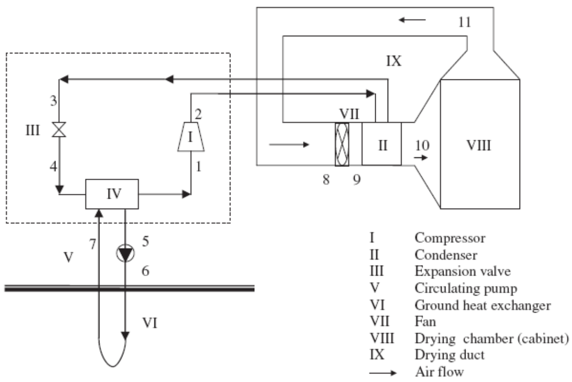

Figure 1: Schematic of the air/water heat pump system.

Figure 2: Outside view of the GSHP drying system investigated

The R-22 at low pressure is vaporized in the evaporator by heat drawn from the soil using a ground heat exchanger. The compressor raises the enthalpy of the R-22 of the heat pump and discharges it as superheated vapor at high-pressure. Heat is removed from the R-22 and returned to the process air at the condenser. The R-22 is then throttled to the low-pressure line (using an expansion valve) and enters the evaporator to complete the cycle. In the cabinet drying system, the hot air at the exit of the condenser is allowed to pass through the drying chamber where it gains latent heat from the product to be dried. Some of the fresh air from the ambient air is mixed with the moist air expelled from the drying chamber before entering to the condenser.

2.2 Experimental Procedure

Measurements were performed to evaluate the performance of the system by calculating energy and exergy efficiencies. Before starting the experiments, the system was run for at least one hour to obtain steady-state conditions. The experiments were performed at a drying temperature of 45°C with a relative humidity of 16%.

Fresh laurel leaves picked for drying tests were separated randomly in three groups as the weight of each group was 0.012 kg (± 0.0005 kg). The leaves were 0.090-0.100 m long and 0.030-0.040 m wide, and ones with no blemish were selected and used for the drying tests. The leaves were sprinkled on the tray, so that they could not touch each other. The initial moisture content of laurel leaves was determined using a standard method (AOAC, 1990), by vacuum drying at 70°C for 24 hours. This was repeated three times to obtain a reasonable average. The initial moisture content of the meant leaves samples was determined to be 48.5% w.b. (wet basis). During the experiments, temperature and relative humidity of ambient and inlet-outlet drying air, temperature of product, water-antifreeze solution and refrigerating fluid were recorded. Temperature, pressure, and humidity values were measured with sensors and recorded in the data logger. The surface temperature of the drying chamber and airflow rates were measured using a Fluke 61 infrared thermometer and a digital anemometer, respectively.

3. MODELING AND ANALYSIS

Mass, energy and exergy balances are employed to find the heat inputs, the rates of exergy destructions, and energy and energy efficiencies. Steady-state, steady-flow processes are assumed. A general mass balance can be expressed in rate form as

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

where $\dot{m}$ is the mass flow rate, and the subscript in stands for inlet and out for outlet.

Energy and exergy balances, equating total energy (exergy) inputs to total energy (exergy) outputs, can be written as

$$\dot{E}_{in} = \dot{E}_{out} \tag{2}$$

$$\dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_{dest} \tag{3}$$

The specific flow exergy of refrigerant, air or water is evaluated as

$$\psi_{r,w} = (h - h_0) - T_0(s - s_0) \tag{4}$$

The enthalpy and entropy of air are calculated from the following equations, respectively (Schmidt et al., 1998).

$$h = C_p T + \omega h_{fg} \tag{5}$$

$$s = C_{p,air}\ \ln\frac{T}{T_0} - R_{air}\ \ln\frac{P_0 - RH.P^{''}}{P_0} + \omega\left(s^{''} - R_w \ln\ RH\right) \tag{6}$$

The exergy rate is determined as

$$\dot{E}x = \dot{m}\psi \tag{7}$$

where h is enthalpy, s is entropy, and the subscript zero indicates properties at the dead (reference) state (i.e., at P_0 and T_0).

The exergy destructions in the heat exchanger (condenser or evaporator) and circulating pump are evaluated as follows, respectively:

$$\dot{E}x_{dest,\,HE} = Ex_{in} - \dot{E}x_{out} \tag{8}$$

$$\dot{E}x_{dest,\,pump} = \dot{W}_{pump} - \left(\dot{E}x_{out} - \dot{E}x_{in}\right) \tag{9}$$

The energy-based efficiency measure of the GSHP unit (COP_{HP}) and the overall GSHP system (COP_{sys}) is calculated as follows, respectively (Hepbasli et al., 2006):

$$COP_{GSHP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{10a}$$

or, in terms of electrical input,

$$COP_{GSHP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp,elec}} \tag{10b}$$

and

$$COP_{sys} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp} + \dot{W}_{pump} + \dot{W}_{fans}} \tag{11a}$$

or, in terms of electrical input,

$$COP_{sys} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp,elec} + \dot{W}_{pump,elec} + \dot{W}_{fans,elec}} \tag{11b}$$

Here,

$$\dot{W}_{comp,elec} = \dot{W}_{comp}\ /(\eta_{comp,elec}\eta_{comp,mech}) \tag{12a}$$

$$\dot{W}_{pump,elec} = \dot{W}_{pump}\ /(\eta_{pump,elec}\eta_{pump,mech}) \tag{12b}$$

$$\dot{W}_{fans,elec} = \dot{W}_{fans}\ /(\eta_{fan,elec}\eta_{fan,mech}) \tag{12c}$$

$$\dot{W}_{com,elec} = \sqrt{3}\,V_{comp} I_{comp} Cos\varphi \tag{13a}$$

$$\dot{W}_{pump,elec} = V_{pump} I_{pump} Cos\varphi \tag{13b}$$

$$\dot{W}_{fan,elec} = V_{fan} I_{fan} Cos\varphi \tag{13c}$$

The specific moisture extraction rate is defined as the ratio of the moisture removed in kg to the energy input in kWh (Hawlader and Jahangeer, 2006):

$$SMER = \frac{\text{Moisture removed in kg}}{\text{Energy input in kWh}} \tag{14a}$$

$$SMER_{HPD} = \frac{\dot{\mathrm{m}}_{\mathrm{w}}}{\dot{W}_{comp,elec} + \dot{W}_{pump,elec} + \dot{W}_{fans,elec}} \tag{14b}$$

where $\dot{m}_w$ is the moisture in kg water per hour.

The exergy efficiency is expressed as the ratio of total exergy output to total exergy input:

$$\varepsilon = \frac{\dot{E}x_{output}}{\dot{E}x_{input}} \tag{15}$$

where “output” refers to “net output” or “product” or “benefit” or “desired value”, and “input” refers to “driving input” or “fuel”.

The exergetic coefficients of performance of the GSHP unit and whole system are as follows:

$$COP_{ex,GSHP} = \frac{\dot{Q}_{cond}\,(1 - \frac{\mathrm{T_0}}{\mathrm{T_{cond}}})}{\dot{W}_{comp,elec}} \tag{16a}$$

$$COP_{ex,sys} = \frac{\dot{Q}_{cond}\,(1 - \frac{\mathrm{T_0}}{\mathrm{T_{cond}}})}{\dot{W}_{comp,elec} + \dot{W}_{pump,elec} + \dot{W}_{fan,elec}} \tag{16b}$$

The exergy efficiency of the heat exchanger (condenser or evaporator) is determined as the increase in the exergy of the cold stream divided by the decrease in the exergy of the hot stream, on a rate basis, as follows:

$$\varepsilon_{HE} = \frac{E\dot{x}_{cold,out} - E\dot{x}_{cold,in}}{E\dot{x}_{hot,in} - E\dot{x}_{hot,out}} = \frac{\dot{m}_{cold}(\psi_{cold,out} - \psi_{cold,in})}{\dot{m}_{hot}(\psi_{hot,in} - \psi_{hot,out})} \tag{17}$$

Van Gool (1997)’s improvement potential on a rate basis, denoted $I\dot{P}$, is expressible as

$$I\dot{P} = (1 - \varepsilon)\,(E\dot{x}_{in} - \mathrm{E}\dot{\mathrm{x}}_{\mathrm{out}}) \tag{18}$$

4. APPLICATION OF ENERGY AND EXERGY ANALYSES TO SYSTEM

The following assumptions are used during the energy and exergy analyses:

a) All processes are steady-state and steady-flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.

b) Heat transfer to the system and work transfer from the system are positive.

c) Heat transfer and refrigerant pressure drops in the tubing connecting the components are neglected since their lengths are short.

d) The compressor mechanical ($\eta_{comp,mech}$) and the compressor motor electrical ($\eta_{comp,elec}$) efficiencies are 81% and 70%, respectively. These values are based on actual data in which the power input to a compressor is 2.025 kW.

e) Air is an ideal gas with a constant specific heat.

f) The circulating pump mechanical ($\eta_{pump,mech}$) and the circulating pump motor electrical ($\eta_{pump,elec}$) efficiencies are 90% and 86%, respectively. These values are based on an electric power of 0.057 kW obtained from the pump characteristic curve.

g) The fan mechanical ($\eta_{fan,mech}$) and the fan motor electrical ($\eta_{fan,elec}$) efficiencies are 40% and 70%, respectively. These values are based on the fan characteristic data and the proposed efficiency values for a small propeller fan (Nagano et al., 2003).

Mass and energy balances as well as exergy destructions obtained from exergy balances for each of the GSHP drying system components illustrated in Figure 1 are derived as follows:

Compressor (I):

$$\dot{m}_1 = \dot{m}_{2,s} = \dot{m}_{act,s} = \dot{m}_r \tag{19a}$$

$$\dot{W}_{comp} = \dot{m}_r (h_{2,act} - \mathrm{h}_1) \tag{19b}$$

$$E\dot{x}_{dest,comp} = \dot{m}_r (\psi_1 - \psi_{2,\mathrm{act}}) + \dot{W}_{comp} \tag{19c}$$

where heat interactions with the environment are neglected.

The mechanical-electrical losses and internal reversibility due to fluid friction are obtained as follows (Kotas, 1995):

$$E\dot{x}_{dest,comp,mech,elec} = \dot{W}_{comp,elec}(1 - \eta_{comp,elec}\eta_{comp,mech}) \tag{19d}$$

$$E\dot{x}_{dest,comp,\mathrm{int}} = E\dot{x}_{dest,comp} - E\dot{x}_{dest,comp,mech,elec} \tag{19e}$$

Condenser (II):

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_r \,;\, \dot{m}_{10} = \dot{m}_8 = \dot{m}_{air} \tag{20a}$$

$$\dot{Q}_{cond} = \dot{m}_r (h_{2,act} - \mathrm{h}_3) \,;\, \dot{Q}_{cond} = \dot{m}_{air} C_{p,air} (T_{10} - \mathrm{T}_8) \tag{20b}$$

$$E\dot{x}_{dest,cond} = \dot{m}_r (\psi_{2,act} - \psi_3) + \dot{m}_{air} (\psi_{10} - \psi_8) \tag{20c}$$

Expansion (throttling) valve (III):

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_r \tag{21a}$$

$$(h_3 = \mathrm{h}_4) \tag{21b}$$

$$E\dot{x}_{dest,\mathrm{exp}} = \dot{m}_r (\psi_3 - \psi_4) \tag{21c}$$

Evaporator (IV):

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_r \tag{22a}$$

$$\dot{Q}_{evap} = \dot{m}_r (h_1 - h_4) \,;\, \dot{Q}_{evap} = \dot{m}_w C_w (T_7 - \mathrm{T}_5) \tag{22b}$$

$$E\dot{x}_{dest,evap} = \dot{m}_r (\psi_4 - \psi_1) + \dot{m}_{air} (\psi_7 - \psi_5) \tag{22c}$$

Circulating pump (V):

$$\dot{m}_5 = \dot{m}_{6s} = \dot{m}_{6,act} = \dot{m}_w \tag{23a}$$

$$\dot{W}_{pump} = \dot{m}_w (h_{6,act} - \mathrm{h}_5) \tag{23b}$$

$$E\dot{x}_{dest,pump} = \dot{m}_r (\psi_6 - \psi_5) + \dot{W}_{pump} \tag{23c}$$

Ground heat exchanger (VI):

$$\dot{m}_5 = \dot{m}_7 = \dot{m}_w \tag{24a}$$

$$\dot{Q}_{ghe} = \dot{m}_w (h_7 - h_5) \tag{24b}$$

$$E\dot{x}_{dest,ghe} = \dot{m}_w (\psi_7 - \psi_5) + \dot{Q}_{ghe} (1 - \frac{\mathrm{T}_0}{\mathrm{T}_{ghe}}) \tag{24c}$$

Fan (VII):

$$\dot{m}_5 = \dot{m}_{5'} = \dot{m}_{air} \tag{25a}$$

$$\dot{W}_{fan} = \dot{m}_{air} [(h_9 - h_8) + \frac{w^2{}_{exit}}{2}] \tag{25b}$$

$$E\dot{x}_{dest,fan} = \dot{W}_{fan,elec} + \dot{m}_{air} (\psi_{9'} - \psi_8) \tag{25c}$$

Drying chamber (cabinet) (VIII):

$$\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{air} \,;\, \dot{m}_{evap} = \dot{m}_{air} (\omega_{11} - \omega_{10}) \tag{26a}$$

$$\dot{E}x_{m2} - \dot{E}x_{m1} = \dot{E}x_{da1} - \dot{E}x_{da2} + \dot{E}x_{evap} - \dot{E}x_{loss} - \dot{E}x_{dest} \tag{26b}$$

$$\dot{Q}_{evap} = \dot{m}_{evap}\, h_{fg} \tag{26c}$$

$$\dot{E}x_{loss} = \dot{Q}_{loss} (1 - \frac{T_0}{T_b}) \tag{26d}$$

$$\dot{E}x_{evap} = \dot{Q}_{evap} (1 - \frac{T_0}{T_m}) \tag{26e}$$

$$E\dot{x}_{dest,dry} = \dot{m}_{air} (\psi_{11} - \psi_{10}) + \dot{m}_{dp} (\psi_{p2} - \psi_{p1}) - \dot{E}x_{loss} + \dot{E}x_{evap} \tag{26f}$$

where interactions with the environment are neglected.

Drying duct (IX):

$$\dot{m}_8 = \dot{m}_{11} = \dot{m}_{air} \tag{27a}$$

$$\dot{Q}_{dc} = \dot{m}_{da} (h_{11} - \mathrm{h}_8) \tag{27b}$$

$$E\dot{x}_{dest,evap} = \dot{Q}_{dc} (1 - \frac{T_0}{T_{dd}}) + \dot{m}_{air} (\psi_{11} - \psi_8) \tag{27c}$$

Exergy efficiencies of the GSHP drying system and its components are evaluated as follows:

GSHP unit (I-IV):

$$\varepsilon_{GSHP} = \frac{\dot{E}x_{heat}}{\dot{W}_{comp,elec}} = \frac{\dot{E}x_{in,cond} - \dot{E}x_{out,cond}}{\dot{W}_{comp,elec}} \tag{28}$$

Overall GSHP system (I-VII):

$$\varepsilon_{GSHP,sys} = \frac{\dot{E}x_{in,cond} - \dot{E}x_{out,cond}}{\dot{W}_{comp,elec} + \dot{W}_{pump,elec} + \dot{W}_{fans,elec}} \tag{29}$$

Compressor (I):

$$\varepsilon_{comp} = \frac{\dot{E}x_{2,act} - \dot{E}x_1}{\dot{W}_{comp,elec}} \tag{30}$$

Condenser (II):

$$\varepsilon_{cond} = \frac{\dot{E}x_{10} - \dot{E}x_8}{\dot{E}x_{2,act} - \dot{E}x_3} = \frac{\dot{m}_w(\psi_{10} - \psi_8)}{\dot{m}_r(\psi_{2,act} - \psi_3)} \tag{31}$$

Expansion (throttling) valve (III):

$$\varepsilon_{\exp} = \frac{\dot{E}x_4}{\dot{E}x_3} = \frac{\psi_4}{\psi_3} \tag{32}$$

Evaporator (IV):

$$\varepsilon_{evap} = \frac{\dot{E}x_5 - \dot{E}x_7}{\dot{E}x_4 - \dot{E}x_1} = \frac{\dot{m}_w(\psi_5 - \psi_7)}{\dot{m}_r(\psi_4 - \psi_1)} \tag{33}$$

Circulating pump (V):

$$\varepsilon_{pump} = \frac{\dot{E}x_6 - \dot{E}x_5}{\dot{W}_{pump,elec}} = \frac{\dot{m}_w(\psi_6 - \psi_5)}{\dot{W}_{pump,elec}} \tag{34}$$

Ground heat exchanger (VI):

$$\varepsilon_{ghe} = \frac{\dot{E}x_7}{\dot{E}x_5 + \dot{Q}_{ghe}(1 - \frac{T_0}{T})} = \frac{\dot{m}_w\psi_7}{\dot{m}_w\psi_5 + \dot{Q}_{ghe}(1 - \frac{T_0}{T})} \tag{35}$$

Fan (VII):

$$\varepsilon_{fan} = \frac{\dot{E}x_9 - \dot{E}x_8}{\dot{W}_{fan,elec}} = \frac{\dot{m}_{air}(\psi_9 - \psi_8)}{\dot{W}_{fan,elec}} \tag{36}$$

Drying (VIII):

There are mainly two ways of formulating exergetic efficiency for drying systems. A comparison of these efficiencies in a tabulated form is made by Kuzgunkaya and Hepbasli (2007) elsewhere.

The first one is proposed by Syahrul et al. (2002, 2003) as follows:

$$\varepsilon_{dry} = \frac{\dot{E}x_{evap}}{\dot{E}x_{10}} = \frac{\dot{m}_w\left(1 - \frac{T_0}{T_m}\right)h_{fg}}{\dot{m}_{air}\psi_{10}} \tag{37a}$$

The second one is used by some investigators as (Midilli and Kucuk, 2003; Akpinar, 2006):

$$\varepsilon_{dry} = \frac{\dot{E}x_{in} - \dot{E}x_{loss}}{\dot{E}x_{in}} = 1 - \frac{\dot{E}x_{loss}}{\dot{E}x_{in}} = 1 \quad \frac{\dot{m}_{air}(\psi_{11} - \psi_{10})}{\dot{m}_{air}\psi_{10}} \tag{38b}$$

Drying duct (IX):

$$\varepsilon_{dd} = \frac{\dot{E}x_8}{\dot{E}x_{11} + \dot{Q}_{dd}(1 - \frac{T_0}{T_{dd}})} \tag{39}$$

5. RESULTS AND DISCUSSION

Temperature, pressure and mass flow rate data for the working fluid (R-22), water and air are given in Table I following the state numbers specified in Figure 1. Exergy rates are also calculated for each state and presented in this table. In this study, the reference state is taken to be the state of environment at which the temperature and the atmospheric pressure on 10 July 2006 were 27°C and 101.325 kPa, respectively. The thermodynamic properties of water, air and R-22a are found using the Engineering Equation Solver (EES) software package (F-chart, 2009).

Table I presents some energetic and exergetic data for one representative unit of the GSHP drying system. As seen from this table, the exergy efficiency values for the GSHP unit, GSHP system and overall GSHP drying system on a product/fuel basis are estimated to be 21.1, 20.5 and 15.5%, respectively. The COP values are found to be 2.88 and 1.63 for the HP unit using Equations (10a) and (10b), while they are obtained to be 2.65 and 1.45 for the GSHP drying system using Equations (11a) and (11b).

It is also clear from Table II that the greatest irreversibility occurs in the motor-compressor assembly, followed by the condenser, expansion valve and evaporator on the GSHP system basis, accounting for 51.83, 20.95, 12.91 and 7.99 of this system, respectively, as given in Table II.

Figure 3 illustrates the exergy loss and flow (Grassmann) diagrams of the whole drying system. This diagram gives the quantitative information related to the share of the exergy input to the GSHP system. In this regard, the mechanical–electrical losses accounts for 46% of the system input. The mechanical–electrical losses are due to imperfect electrical, mechanical and isentropic efficiencies and emphasize the need for paying close attention to the selection of this equipment, since components of inferior performance can considerably reduce overall system performance. Since compressor power depends strongly on the inlet and outlet pressures, any heat exchanger improvements that reduce the temperature difference will reduce compressor power by bringing the condensing and evaporating temperatures closer together. From a design standpoint, compressor irreversibility can be reduced independently. Recent advances in the HP market have led to the use of scroll compressors. Replacing the reciprocating compressor used in this study by a scroll unit could increase heating COP.

The second irreversibility is partly due to the large degree of superheat achieved at the end of the compression process, leading to large temperature differences associated with the initial phase of heat transfer. Irreversibilities in the evaporator and the condenser occur due to the temperature differences between the two heat exchanger fluids, pressure losses, flow imbalances and heat transfer with the environment.

The third largest irreversibility is associated with the evaporator, and the fourth largest with the capillary tube due to the pressure drop of the refrigerant passing through it. The only way to eliminate the throttling loss is to replace the capillary tube (the expansion device) with an isentropic turbine (an isentropic expander) and to recover some shaft work from the pressure drop.

As it can be seen in Table II, where some thermodynamic parameters and exergetic improvement potential rates for one unit of the GSHP drying system under the state conditions are included, on the GSHP drying system basis, the fan has the highest exergetic improvement potential rate with about 671 W. This is followed by the compressor and condenser with some 552 and 413 W, respectively. The fan may be replaced by a higher efficient one.

Exergetic COP value (COP_{ex}) on the system basis is found to be 0.174. As found from the exergy analysis results in this study, a more efficient compressor and fan may be used by resulting in an increase in the efficiency and hence reducing power input to this equipment.

6. CONCLUSIONS

A methodology using comprehensive energy and exergy analyses is presented and applied for evaluating GSHP drying systems and their components. Experimental and assumed values are utilized in the analysis. Exergy destructions in the overall HP systems are quantified.

Some concluding remarks can be drawn from the results:

a) The values for COP_{GSHP} and COP_{sys} are found to be in the range of 1.63-2.88 and 1.45-2.65, respectively.

b) The values for $SMER_{sys}$ are found to be 0.122 kg/kWh.

c) The values for $COP_{ex,.GSHP}$ and $COP_{ex,sys}$ are obtained to be 0.196 and 0.174, respectively.

d) The exergy efficiency values for the GSHP unit and the whole system on a product/fuel basis are 21.1 and 15.5% at a dead state temperature of 27^{o}C, respectively.

e) The largest irreversibility in the GSHP drying system is associated with the condenser, followed by the compressor and expansion valve.

f) The results can focus an engineer's attention on components where the greatest potential is destroyed and quantify the extent to which modification of one component affects, favorably or unfavorably, the performance of other components of the system.

NOMENCLATURE

C	= specific heat (kJ/(kg K))
COP	= heating coefficient of performance of heat pump (dimensionless)
$\dot{E}$	= energy rate (kW)
$\dot{E}x$	= exergy rate (kW)
f	= exergetic factor (dimensionless)
$\dot{F}$	= exergy rate of the fuel (kW)
h	= specific enthalpy (kJ/kg)
I	= Amper (A)
$\dot{I}$	= irreversibility rate (kW)
$I\dot{P}$	= improvement potential rate (kW)
$\dot{m}$	= mass flow rate (kg/s)
P	= pressure (kPa)
$\dot{P}$	= exergy rate of product (kW)
$\dot{Q}$	= heat transfer rate (kW)
R	= ideal gas constant (kJ/kgK)
RH	= relative humidity (%)
s	= specific entropy (kJ/kgK)
SMER	= specific moisture extraction rate (kg/kWh)
SMExI	= specific moisture exergetic indice (kg/kWh)
T	= temperature (ºC or K)
V	= voltage (V)
w	= velocity (ms-1)
$\dot{W}$	= work rate or power (kW)

Greek letters

η	= energy (first law) efficiency (dimensionless)
ψ	= specific exergy (kJ/kg)
Cos φ	= power factor (dimensionless)
ε	= exergy (second law) efficiency (dimensionless)
χ	= relative irreversibility (dimensionless)
ω	= specific humidity ratio (kg water/ kg air)

Indices

$\cdot$ (over dot) = rate

0	= dead (reference) state
act	= actual
c	= chamber
comp	= compressor
cond	= condenser
da	= drying air
dc	= drying chamber
dd	= drying duct
dest	= destroyed (destruction)
dry	= drying
elec	= electric
evap	= evaporator
exp	= expansion valve
fg	= vaporization
g	= ground
ghe	= ground heat exchanger
GSHP	= ground source heat pump
HE	= heat exchanger
HPD	= heat pump dryer
in	= inlet
m	= material
m	= material
mech	= mechanical
out	= outlet
p	= constant pressure, product
P	= product
r	= refrigerant
s	= isentropic
sys	= system
Tot	= total
w	= water

Abbreviations

ASHP	= air-source heat pump
HP	= heat pump
GSHP	= ground-source (or geothermal) heat pump
HPD	= heat pump dryer

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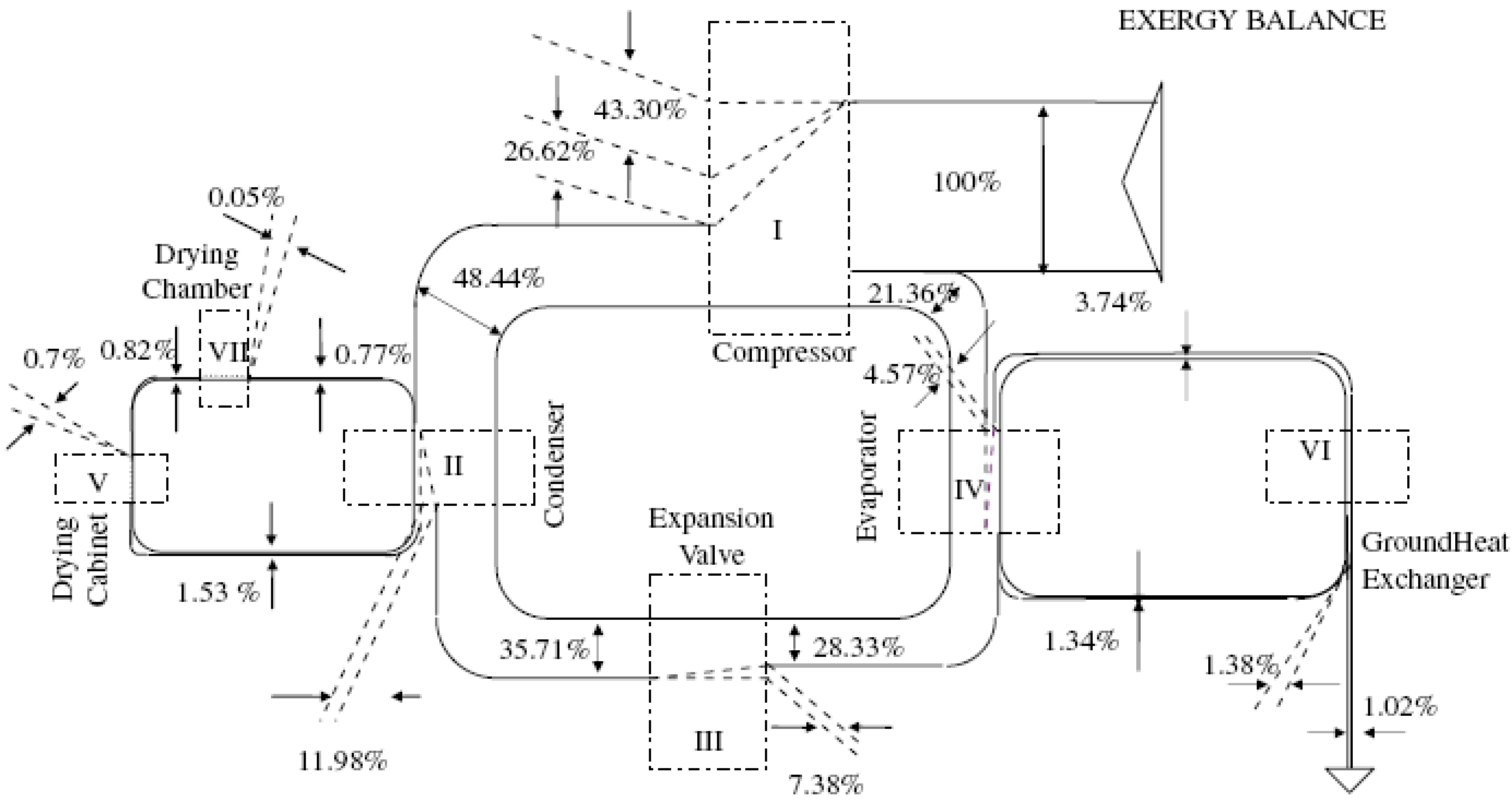

Figure 3: Exergy loss and flow (Grassmann) diagram of the GSHP drying system.

Table 1: Some thermodynamic data as well as energy and exergy rates provided for one representative unit of the GSHP drying system.

State No.	Description	Fluid	Phase	Temperature T (°C)	Pressure kPa (bar)	Specific humidity ratio ω (kgwater/kg)	Specific enthalpy h (kJ/kg)	Specific entropy s (kJ/kgK)	Mass flow rate $\dot{m}$ (kg/s)	Specific exergy ψ (kJ/kg)	Exergy rate $\dot{E}x$ (kW)	Energy rate $\dot{E}$ (kW)
0	-	Refrigerant	Dead state	27	101.325	-	430.6	1.987	-	-	-	
0	-	Water	Dead state	27	101.325	-	113.2	0.395	-	-	-	
0	-	Air	Dead state	27	101.325	0.0080	47.5	0.170	-	-	-	
1	Evaporator outlet/compressor inlet	Refrigerant	Superheated vapor	2.1	440	-	407.8	1.772	0.0183	41.70	0.761	7.45
2s	Condenser inlet/compressor outlet	Refrigerant	Superheated vapor	104.5	2900	-	458.0	1.772	0.0183	91.90	1.678	8.36
2a	Condenser inlet/compressor outlet	Refrigerant	Superheated vapor	117.5	2900	-	470.6	1.805	0.0183	94.55	1.727	8.59
3	Condenser outlet/expansion valve inlet	Refrigerant	Compressed liquid	68	2900	-	290.0	1.286	0.0183	69.70	1.273	5.30
4	Evaporator inlet	Refrigerant	Mixture	-3.8	440	-	290.0	1.334	0.0183	55.30	1.010	5.30
5	Water outlet from evaporator	Water	Liquid	18.9	101.325	-	79.3	0.280	0.2222	0.60	0.133	17.62
6	Water outlet from circulating pump	Water	Liquid	18,95	101.325		79.5	0.281	0.2222	0.50	0.111	17.67
7	Water inlet to evaporator	Water	Liquid	21.4	101.325	-	89.0	0.314	0.2222	0.22	0.048	19.77
8	Air inlet to fan/air outlet from canal	Air	Gas	36.88		0.0078	57.2	0.202	0.2716	0.10	0.027	15.52
9	Air inlet to condenser	Air	Gas	37.08		0.0078	57.4	0.202	0.2716	0.11	0.030	15.58
10	Air inlet to dryer	Air	Gas	44.51		0.0095	69.3	0.242	0.2690	0.20	0.054	18.64
11	Air outlet from dryer /air inlet to canal	Air	Gas	43.27		0.0098	68.8	0.240	0.2689	0.10	0.028	18.50
P1	In the dryer	Product	Solid	42.51	101.325	2.7563	110.3		0.7630	1.01	0.768	84.19
P2	In the dryer	Product	Solid	42.54	101.325	2.5957	110.4		0.7630	1.01	0.770	84.25

Table 2: Some energetic and exergetic analysis data provided for one representative unit of the GSHP drying system (the dead state temperature and the atmospheric pressure are 27^{o}C and 101.325 kPa, respectively).

Component		Used exergy (kW)	Available exergy (kW)	Rate of exergy destruction (kW)	Power input (kW)	Exergy efficiency (%)	COP	Rate of exergetic improvement potential (W)
No	Name	$\dot{P}$	$\dot{F}$	$\dot{Ex}_{dest}$		ε		$\dot{IP}$
I	Compressor	1.73	2.78	1.056	2.02	47.8		551.63
II	Condenser	1.33	1.75	0.427	3.30	6		413.43
III	Expansion Valve	1.01	1.27	0.263	-	79.3		54.33
IV	Evaporator	0.89	1.06	0.163	2.15	34.5		106.71
V	Circulating pump	0.11	0.19	0.080	0.06	38.7		48.82
VI	Ground heat exchanger	0.05	0.10	0.049	2.15	49.3		24.88
VII	Fan	0.03	0.71	0.680	0.68	13		671.43
VIII	Dryer	0.80	0.83	0.031	0.34	18		25.54
IX	Canal or duct	0.03	0.16	0.132	3.15	17.2		109.17
							2.88a	
I-IV	GSHP unit	4.96	6.87	1.908		21.1	1.63b	
I-VI	GSHP system	5.12	7.15	2.037		20.5	2.65c	
I-IX	Overall system	5.98	8.86	2.880		15.5	1.45d	

[a]Using Equation (10a) [b] Using Equation (10b) [c]Using Equation (11a) [d]Using Equation (11b)