Fluid selection and parametric analysis on condensation temperature and plant height for a thermogravimetric heat pump

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Nomenclature

AE	auxiliary heat exchanger
C	condenser
COP	coefficient of performance
C_F	carrier fluid
E	evaporator
G_W	mass flow rate of the working fluid (kg/s)
G_C	mass flow rate of the carrier fluid (kg/s)
GWP	global warming potential
h	enthalpy (kJ/kg)
h_{VapC}	vapor enthalpy of the carrier fluid at condensation temperature (kJ/kg)
k	height increasing factor
H_P	pump total head
L_P	pump power (kW)
LD	liquid duct for carrier fluid
M	mixer
MM	molar mass
ODP	ozone depletion potential
O&M	operation and maintenance
P	pressure (MPa)
P	pump
ΔP	difference between condensation and evaporation saturation pressure
ΔP_{fr}	frictional pressure losses
P_{Red}	ratio between ΔP of any fluid and ΔP_{R134a}
$P_{Sat@Tcond}$	saturation pressure at condensation temperature
$P_{Sat@Tevap}$	saturation pressure at evaporation temperature
P_{SatC}	saturation pressure of the carrier fluid
P_{SatW}	saturation pressure of the working fluid
q_a	specific heat exchanged on the horizontal duct after the mixer (kJ/kg)
q_{ae}	specific heat exchanged in the auxiliary heat exchanger (kJ/kg)
q_c	specific heat generated during isothermal compression (kJ/kg)
q_{circ}	specific heat available to the end user (kJ/kg)
q_{Cond}	specific latent heat of condensation (kJ/kg)
q_{Evap}	specific latent heat of evaporation (kJ/kg)
Q_{CEv}	partial evaporation heat of the carrier fluid on T-PD (kW)
Q_{CFfr}	heat generation rate in the carrier fluid owing to friction in single phase ducts (kW)
$Q_{\Delta P_{T-PD}}$	heat generation rate in the carrier fluid owing to friction in two phase duct (kW)
Q_P	heat generation rate in the pumping process (kW)
Q_Y	total heat generation in the carrier fluid due to the dissipation phenomena (kW)
Q_{AE}	auxiliary Exchanger thermal power (kW)
Q_U	user's required thermal power (kW)
S	separator
s	entropy (kJ/kg/K)
T	temperature (Kelvin)
T_{Cond}	condensation temperature (K)
T_{Evap}	evaporation temperature (K)
ΔT_{Reg}	temperature difference between the two fluids on the regenerator(suction-line heat exchanger) (K)
TV	throttling valve
$T-PD$	two phase duct
VD	vapor duct for the working fluid
W_F	working fluid
w_W	working fluid velocity (m/s)
w_C	carrier fluid velocity (m/s)
X_V	gas fraction of volume flow
Z_{Min}	plant's minimum height (m)
Z_{Real}	plant's real height (m)

Greek symbols

α	cross section average void fraction
ρ	density (kg/m^3)
$\overline{\rho}$	mean photographic density of the mixture along T-PD
ρ_{CF}	density of the working fluid (kg/m^3)
η_P	pump's efficiency
η_{hyP}	pump's hydraulic efficiency

1. Introduction

The global energy consumption is predicted to increase about 7% by 2030, i.e. of approximately 6 quadrillion BTU [1]. In industrialized countries, service and residential buildings consume approximately 40% of the total primary energy supply, and out of this amount more than 60% of the energy consumption is related to heating and cooling applications [2–4]. Efficiency of thermal devices, plants and systems has therefore become a major subject of research and analysis [5,6].

A thermogravimetric heat pump (TGHP) is a system based on a non-conventional regenerative thermodynamic cycle whose theoretical coefficient of performance (COP), thanks to a very favorable opportunity of regeneration, is pretty close to the ones of a reversible cycle.

The thermogravimetric system was first proposed during the second half of the last century as a power generating system with a non-conventional direct cycle [7–10]. The reverse cycle was then proposed and several investigations [11–14] were carried out in order to study different aspects which may affect the system's performance. Since techno-economic feasibility, safety issues and environmental aspects – chemical compatibility, thermal stability, toxicity, ozone depletion and global warming potentials (ODP and GWP) – can limit the set of fluids which can be employed, these considerations were also taken into account in the conducted studies. As explained in the following section, the thermogravimetric systems require both a working fluid, as in conventional Rankine cycles, and a carrier fluid, which always remains in the liquid state. Power generation plants, heat pumps and refrigerating systems all require a careful choice of both fluids, depending on the range of the operating temperature. For the power generation plants, high enthalpy regions require that the carrier fluid would have elevated density and vaporization temperature, the necessity which results in the selection of less manageable and practically or economically inconvenient fluids (e.g. molten lead or sulfur was investigated). Due to the mentioned issues, the studies were focused on the low enthalpy region, typically represented by the geothermal fields [7,11], where water can be the best option as the carrier fluid. The latter choice is also valid for the TGHP system which is analyzed in the present study.

The TGHP thermodynamic cycle resembles the reverse Rankine cycle, where the adiabatic compression is substituted by a quasi-isothermal compression of the working fluid in the vapor phase, taking place in a downward two-component two-phase duct, due to the action of a carrier fluid. The compressor used for the Rankine cycle is thus replaced by a feed pump, which is an advantage in terms of vibration and noise reduction and can also lead to a decrement in the investment and O&M costs. With respect to Rankine cycles, two disadvantages are also found. The first is that the fluid dynamics aspects related to the two-phase flow (mainly the increased friction losses) reduce the efficiency of the system [14]. Despite the mentioned issue, the obtainable COP should still be larger than those obtained by the conventional heat pumps [11–13]. Moreover, the system can, in theory, work with several combinations of working-carrier fluids, which makes it quite versatile and able to satisfy a wide range of end user requirements. The second disadvantage is that the height needed for the vertical duct, where the compression takes place, is quite large: depending on the operating temperatures and the thermophysical properties of the chosen fluids, such height may range between 20 m to more than 200 m, particularly when using the most common refrigerants (e.g., HFC134a) as the working fluid. Such aspect practically confines the potential use of the TGHP systems to cases in which the existing structure can be used to support the vertical ducts, e.g. to be employed as the centralized heat pump for buildings with a minimum of 10–12 storeys to skyscrapers.

Previous studies, carried out on TGHP systems, demonstrated the considerable effect of regeneration [9,11] and also analyzed the negative consequences of partial evaporation and dissipative effects [14]. In these studies water was selected as the carrier fluid, and the evaporation and condensation temperatures were fixed. For each specific working fluid, the corresponding plant height and COP (together with other minor parameters) were computed. The obtained results demonstrated the significant dependence of the required plant height on the fluid thermophysical properties. The required height could differ up to factor of 7 by employing different substances. The PF5050 perfluorocarbon and the HFC134a (1,1,1,2-tetrafluoroethane) hydrofluorocarbon were also found to be the most promising working fluids (apart from some aspects related to GWP), with the main disadvantage being the obtained plant heights, based on which PF5050 was recommended for 10-12 storey buildings while HFC134a for skyscrapers [12].

The previous studies on the TGHP systems were conducted employing limited number of working fluids and investigating the performance of the system using a wider range of fluids can evidently provide useful results which can be employed for improving the performance and facilitating the implementation of this type of technology. The first aim of the present study is to analyze the performance of the TGHP plant for a wide range of different working fluids and to investigate the ones using which, under the same input conditions, the required plant height may be significantly reduced with respect to the previously proposed fluids. Particularly the comparison is made with respect to HFC134a, which is chosen, due to its broad application as a refrigerant, as the reference fluid. After choosing the promising working fluids, as the next step of the investigation, a parametric analysis is carried out in order to study the effect of variations in the dimensionless plant height k and the condensation temperature T_{Cond} on the performance of the system.

2. Plant description

A sketch of the thermogravimetric heat pump (TGHP) is shown in Fig. 1. As previously stated, the TGHP unit is based on a modified reverse Rankine cycle where the compression of the working fluid (W_F) is quasi-isothermal as it is performed by a high thermal capacity carrier liquid along a downward two-component, two-phase flow duct. The two fluids are then separated and each fluid proceeds along a different path. The theoretical thermodynamic cycle of the working fluid starts when it is mixed with the carrier fluid (C_F) at the mixer M(state 3″), then the working and the carrier fluids reach the thermal equilibrium through an isobaric process in a horizontal duct (3″–3), through this process, the temperature of the working fluid rises up to T_{Cond}, which is the highest temperature in the cycle. It is worth mentioning that, in the same point, the two fluids are also at the minimum pressure P_{min}. The mixture then descends along the downward adiabatic vertical two-phase duct ($T-PD$) where, due to the large difference between the thermal capacities of the fluids, the process can be modeled as an isothermal compression of the working fluid (3–4). The maximum pressure P_{max} is reached at the base of the column. Afterward, the two fluids are separated by gravity in the separator S and the

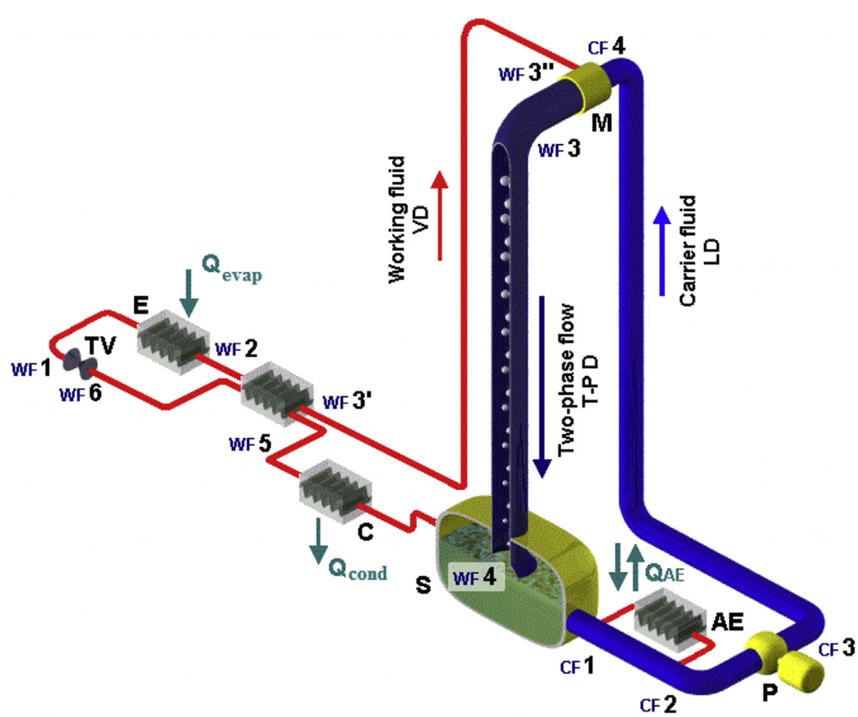

Fig. 1. Sketch of the TGHP system.

working fluid subsequently undergoes an isothermobaric condensation (4–5) in the condenser *C*; where Q_{cond}, which is the major component of the heat given to the end user, is released.

In the next step, the working fluid enters the regenerative heat exchanger (commonly called suction-line heat exchanger) where, through an isobaric process, the heat released by the high temperature working fluid (5–6) is transferred to the low temperature working fluid (2–3′). The main thermodynamic function of this heat exchanger, besides the described regenerative effect, is that the working fluid is subcooled before entering the expansion valve; thus the working fluid, entering the evaporator, is as "wet" as possible resulting in an increment in its latent cooling capacity.

Afterward, the cooled working fluid, through the throttling valve *TV* ("isenthalpic" process 6–1), reaches the evaporation temperature T_{evap} which is the lowest temperature in the cycle. The working fluid completes its cycle by undergoing the isothermobaric evaporation in the evaporator *E* (1–2), the isobaric heating (2–3′) and the rise to the mixer *M* (3′–3″). It should be pointed out that the process 3′–3″includes no work or heat exchange, so the total energy through this process is kept constant (i.e. the sum of static enthalpy, kinetic energy and potential energy is constant). Owing to the small difference between P3″ and P3′, due to the kinetic and gravitational terms, a slight change in the static enthalpy is present. On the contrary, by assuming the friction to be negligible, the process can be approximated to be isentropic. The working fluid transformations are represented in the T-s diagram sketched in Fig. 2.

During the compression along the downward duct (3–4), heat is transferred from the working fluid to the carrier one. The amount of the transferred heat varies according to the thermophysical properties of the two fluids and it might not be equal to the heat given by the carrier to the working fluid during the 3″–3 path. Hence, it is then necessary to include an auxiliary heat exchanger (*AE*) in the system, to subtract this energy from the carrier fluid. As can be thermodynamically verified, the heat transferred in 3–4 process is larger than the one transferred in 3″–3, this extra heat (Q_{AE}) is added to Q_{cond} in the total heat Q_U supplied to the end user. Q_{AE} is clearly the product of the mass flow rate of the carrier fluid and the difference in the specific enthalpy of the carrier fluid before and after the auxiliary heat exchanger.

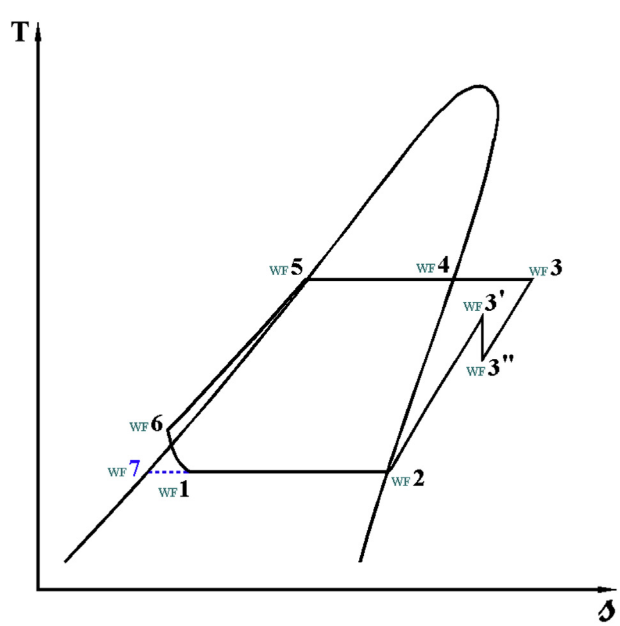

Fig. 2. Theoretical thermodynamic cycle of the TGHP working fluid in the T–s chart.

The carrier fluid, which is always in the liquid phase, after passing through the mentioned auxiliary heat exchanger (*AE*) is fed to the pump *P* and then sent through the liquid duct *LD* to the mixer *M*.

3. Mathematical model

The performances and the required height of the TGHP system have been determined employing a numerical model using the procedure which will be described in the following.

The thermophysical properties of the fluids were determined using the NIST software REFPROP.

The first input values to the model are the condensation and evaporation temperatures T_{Cond} and T_{evap}. The condensation and evaporation pressures are therefore known and their difference ΔP can be calculated as:

$$\Delta P = P_{SatV@T_{Cond}} - P_{SatV@T_{Evap}} \tag{1}$$

Such pressure rise must be imposed on the working fluid by gravity along the downward duct and this value is the factor which determines the minimum required height of the plant. Considering the fact that in the two phase duct, the carrier fluid is always in the liquid phase and the working fluid is always in the vapor phase, the maximum density – and consequently minimum required height – of the two-phase column would be obtained as the limit of the two-phase density for mass flow rate of the working fluid going to zero. It can be therefore computed as:

$$Z_{Min} = \frac{\Delta P}{g\rho_{CF}} \tag{2}$$

In other words, the above determined value corresponds to a two-phase mixture at limit of zero void fraction. The actual height must obviously be always greater: the increasing factor k is accordingly defined as:

$$k = \frac{Z_{\mathrm{Real}}}{Z_{Min}} = \frac{\rho_{CF}}{\overline{\rho}} \tag{3}$$

which, by definition, must always be greater than unity and can also be seen as a dimensionless plant height. It is chosen as one of the inputs for the calculation.

In order to define the different parameters involved in the whole process, the heats which are directly related to the thermodynamic cycle of the working fluid will be firstly introduced. Sign convention is such that the heat and work given to the working fluid are positive. Referring to Fig. 2, the regeneration device is designed to maintain a $\Delta T_{Reg} = T_{Cond} - T_{3'} = 6$ K. The rise in the temperature of the working fluid up to T_{Cond} requires an amount of heat, supplied by the carrier fluid, as described by Eq. (4):

$$q_a = h_3 - h_{3''} > 0 \tag{4}$$

Through the isothermal compression, the heat exchanged between the couple of working fluid and carrier fluid follows the expression:

$$q_c = T_{Max}(s_4 - s_3) < 0 \tag{5}$$

afterward, the heat exchanged in the condenser can be defined as:

$$q_{Cond} = h_5 - h_4 < 0 \tag{6}$$

As was previously pointed out, the algebraic sum of q_a and q_c is the heat which is transferred via the auxiliary heat exchanger (*AE*). Since the heat should always be subtracted from the system in the auxiliary heat exchanger, these three values are summed up as q_{Circ} representing the total heat the working fluid cycle makes available to the end user.

$$q_{Circ} = q_{Cond} + q_c + q_a < 0 \tag{7}$$

It should be pointed out that, as indicated previously, q_a and q_c have different signs. Finally, it is necessary to define the heat input which is received through the evaporator:

$$q_{Evap} = h_2 - h_1 > 0 \tag{8}$$

One peculiar aspect of the TGHP is that the internal energy production due to dissipative effects which prevailingly takes place in processes occurring at T_{Cond}. Consequently the mentioned energy can be transferred as heat to the end user, thus reducing the corresponding negative effect on the COP.

The processes associated with the carrier fluid include its compression through the pump, the flow of carrier fluid along the *LD* duct (causing ΔP_{fr} pressure losses) and the flow of the two-phase mixture along the two phase duct (causing the $\Delta P_{frT-P\,D}$ pressure losses). Such effects have all a positive sign and are quantified by Eqs. (9)–(11) respectively:

$$Q_P = \frac{H_P G_C}{\rho_{CF}} \left(\frac{1}{\eta_{hyP}} - 1 \right) \tag{9}$$

$$Q_{C_F fr} = \frac{\Delta P_{fr} G_C}{\rho_{CF}} \tag{10}$$

$$Q_{\Delta P_{T-PD}} = \sum \frac{\Delta P_{fr_{T-PD}}}{\overline{\rho}} (G_W + G_C) \tag{11}$$

Consequently:

$$Q_Y = Q_P + Q_{C_F fr} + Q_{\Delta P_{T-PD}} \tag{12}$$

Next, Q_U (<0) being fixed as an input, the mass flow rate of the working fluid (G_W) can be determined by the following relation:

$$Q_U + Q_Y = G_W q_{Circ} \tag{13}$$

Since partial evaporation of the carrier fluid inside the two-phase mixture may have significant effects [15,16], it is also considered in the model, following Eq. (14):

$$Q_{CEv} = h_{VapC} G_W \frac{MM_C}{MM_W} \frac{P_{SatC}}{P_{SatW} - P_{SatC}} \tag{14}$$

Q_{CEv} is released by the carrier fluid along the two phase duct and returned back to it at the condenser so it has no influence in terms of net thermal power balance, but it plays a role in the sizing of heat exchangers surfaces. It is also important to underline that the partial evaporation modifies the two-phase structure and pressure drop, thus negatively affecting the pumping power and consequently the COP as will be discussed in the following.

Once mass flow rate of the working fluid (G_W) is determined, the pressure losses for the whole system can be computed. The desired mass flux in the two-phase duct is given as an input, and the mass flow rate of the carrier fluid (G_C) can be computed in order to satisfy the required value for the photographic two-phase flow density $\overline{\rho}$. This requires an iterative procedure that takes into account Eq. (3) as well as the pressure losses in the two phase duct.

For the prediction of the pressure losses in the two phase duct (*T–PD*), the gravitational, accelerative and frictional contributions have been calculated using an iterative procedure after subdivision of the downward column in one hundred slices, so that on each slice the fluid properties and the void fraction could be considered as constant. Different correlations were tested for the prediction of the void fraction [15–18] and finally the Kashinsky correlation [17] was selected. The effect of the partial evaporation of the carrier fluid was modeled considering the variation of the cross section average void fraction α and its consequences on $\overline{\rho}$. For the estimation of the frictional losses, the Davis correlation [19] was used. The fluid volumetric properties may imply significant differences on these aspects.

The single-phase working and carrier fluids velocities (w_W and w_C) are also fixed as inputs, thus duct diameters can be calculated and finally the pump power can be computed as:

$$L_P = \frac{H_P G_C}{\rho_{CF} \eta_P} \tag{15}$$

In order to evaluate the performance of the plant using each fluid, two expressions for the COP have been considered. In order to define the thermodynamic COP, the specific work received by the working fluid cycle, which is applied by the carrier fluid through the isothermal compression in the two phase duct, should be taken into account. Considering the energy balance of the working fluid cycle, this specific work will be equal to the opposite of the algebraic sum of the specific heats which are absorbed or distracted from the working fluid in its cycle ($w = -\sum q_{W_F} = -(q_a + q_c + q_{cond} + q_{evap})$). Accordingly considering the q_{Circ} which is the overall heat made available to the user, the thermodynamic COP can be defined as follows:

$$COP_{TD} = \frac{q_{Circ}}{\sum q_{W_F}} = \frac{q_{Circ}}{q_{Circ} + q_{Evap}} = \frac{1}{1 + \frac{q_{Evap}}{q_{Circ}}} \tag{16}$$

The second performance index which can be employed for analyzing the system, is the real COP in which the effects of partial evaporation as well as pressure drops are considered through the L_P term. The real COP is determined as follows:

$$COP_{Real} = \frac{-Q_U}{L_P} \tag{17}$$

It is worth mentioning that the mixing process is an important issue for TGHP systems and its resulting pressure drop will negatively affect the COP_{Real}. Various experiments have been performed by the co-authors on liquid–gas two-phase flow in vertical pipes,

Table 1
Fixed values for the input parameters.

Parameter	Q_U	η_P	w_W	w_C	T_{Evap}	T_{Cond}	k
Value	100 kW	0.85	20 m/s	2 m/s	283 K	313 K	1.1

Table 2
Pressure reduction respect to R134a for every analyzed fluid keeping $T_{Cond} = 313$ K and k = 1.1, and the percentage of change in the pressure reduction by increasing the temperature to $T_{Cond} = 348$ K.

	P_{Red} T = 313 K	ΔP_{Red}	Z_{Real} k = 1.1 (m)
R245ca	80.26%	3.06%	13.49
R245fa	71.94%	3.34%	19.21
R114	65.14%	1.18%	23.98
R236ea	63.51%	3.00%	25.06
RE134	61.99%	3.43%	26.06
R236fa	53.81%	2.59%	31.82
RC318	49.51%	1.44%	35.00
R227ea	29.86%	0.83%	48.89
R152a	10.88%	0.26%	61.44
R134a	0.00%	0.00%	69.54

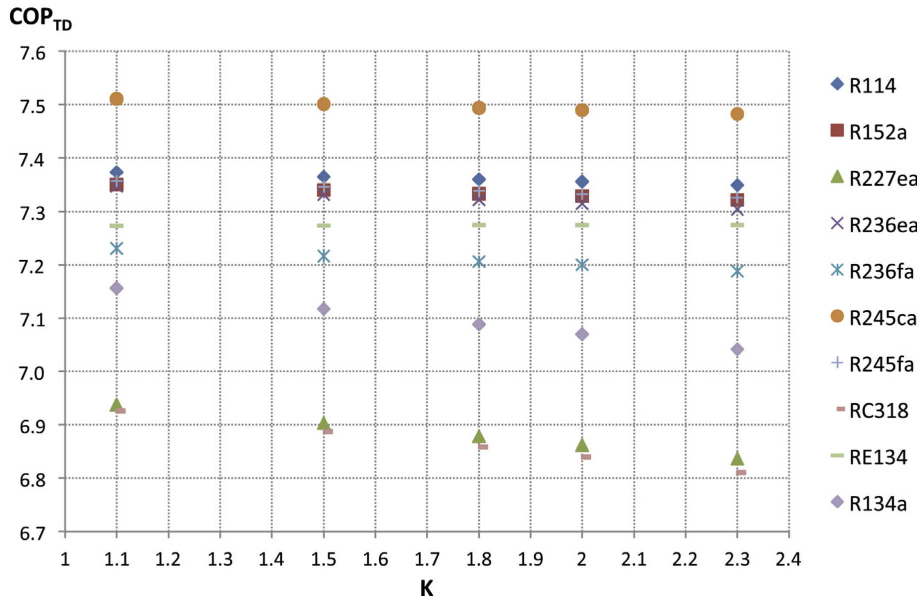

Fig. 3. Variation of the COP_{Real} as a function of the plant height.

with both upward and downward flow, and in horizontal ducts. In these experiments mixing systems embedded within the flow of the liquid (i.e. injectors made of porous bronze or coaxial nozzles provided with various holes) and also external mixing systems were implemented, consisting of a circular chamber externally applied to the liquid duct and fed by various gas ducts. In particular, the latter approach was implemented in the thermogravimetric experimental power plant (direct cycle) constructed in Larderello (Tuscany, Italy) [8]. Through this experimental study, the mixing system has shown an excellent behavior with minimal pressure drops. Nevertheless, the pressure drops of the mixing system are accounted in the calculation of COP_{Real}.

The whole model, apart from the thermophysical properties calculation, has been implemented using a spreadsheet software.

The values of the fixed input parameters utilized in the model are given in Table 1. The T_{Cond} and k values are kept constant at the given values when the other parameter is changed.

4. Investigation procedure

4.1. Fluid selection

The first step of the analysis was finding a group of fluids that, compared to HFC134a (defined as the reference fluid), lead to a

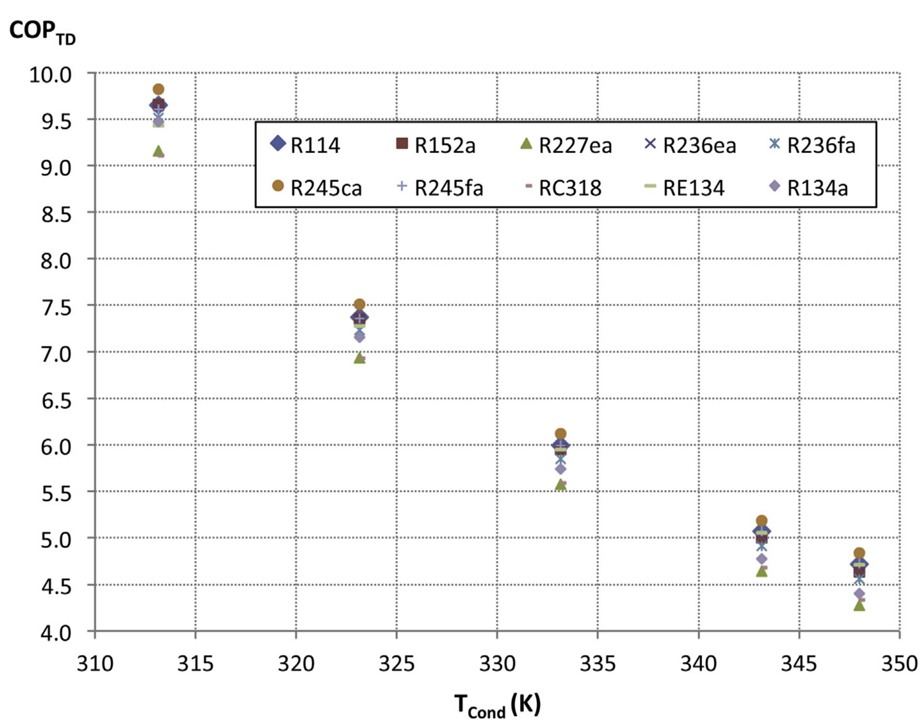

Fig. 4. Variation of the COP_{TD} as a function of k.

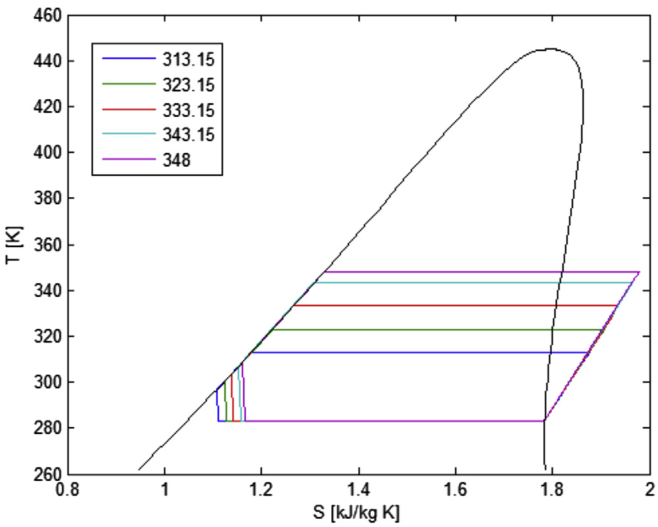

Fig. 5. Variation of the COP_{TD} as a function of T_{Cond}.

lower required height. Considering Eq. (2), it is evident that to achieve this aim, it is necessary that, under the same input temperature conditions, these fluids have a lower ΔP. On the basis of the values returned by the REFPROP software, 17 refrigerant fluids were evaluated, where the ΔP of each fluid was compared with the one of the reference fluid. The fluids with ΔP larger than the reference were immediately discarded, since this would mean an increase in height. Nine apparently promising fluids were selected for evaluation of their TGHP performance.

4.2. Parametric analysis

For the nine selected fluids, a parametric analysis was carried out as the next step of the investigation, in order to study the effect of variations in the dimensionless plant height k and condensation temperature T_{Cond} on the performance of the system. The variation of these values was performed independently, and all the other inputs were kept constant. First of all, T_{Cond} was varied between 313 K and 348 K while keeping $k = 1.1$. This variation resulted in a change in all the terms composing COP_{TD}, while for COP_{Real} the increase in T_{Cond} led to an increase in the height, so both phenomena and their consequences were analyzed. Then, k was varied from 1.1 to 2.3 while maintaining $T_{Cond} = 313$ K, and the corresponding effect on the required mass flow rates of working fluid and the carrier fluid (G_W and G_C) and consequently on COP_{Real} were analyzed. Since T_{Evap} and T_{Cond} were kept constant in this second analysis, the effect of an increase of height on COP_{TD} should be dependent only on Q_C and Q_a, and such a hypothesis was also tested.

5. Results and discussion

As already stated, the first step of the analysis was to identify the fluids that may result in plant heights lower than the one obtained using HFC134, which was chosen as the reference fluid. Table 2 reports the results for the 9 fluids which, among the 17 investigated, granted height reductions. It can be seen that the heights much lower than the one required for HFC134a can be obtained, and that, for some fluids, the plant height is low enough for being employed in small buildings. The results are obtained for the height increasing factor of $k = 1.1$ which corresponds to the minimum required heights for the analyzed fluids. In Table 2, the first column (P_{Red}) indicates each fluid's pressure difference (difference between the saturation pressures at T_{Cond} and T_{Evap}) compared to the reference fluid's (HFC134a) pressure difference under the same T_{Cond} and T_{Evap} conditions; where the percentages are evaluated for a fixed T_{Cond} of 313 K. In the next step, in order to demonstrate the effect of changing T_{Cond} on the evaluated pressure difference percentages, the same analysis has been done for $T_{Cond} = 348$ K and the resulting pressure difference reduction has been compared to the corresponding value in the first case ($T_{Cond} = 313$ K) and the difference percentage has been given in the second column (ΔP_{Red}) (keeping $k = 1.1$ constant). As can be observed in this column, changing the T_{Cond} from 313 K to 348 K does not make a considerable change in the pressure reduction ratio of any of the analyzed fluids. Accordingly, it can be deduced that, although each particular fluid has a very different thermophysical properties with respect to HFC134a, the pressure reduction ratio P_{Red} does not significantly change with T_{Cond}. Thus the pressure and height reduction can be considered temperature independent.

The effect of the variation of k and T_{Cond} on COP_{TD} is shown in Figs. 3 and 4 respectively. As k is increased, Z_{Real} is directly increased while q_{Cond} and q_{Evap} remain constant, so the terms which vary and consequently cause the change in COP_{TD} are q_C and q_a (Eq. (7) and Eq. (16)). It can be observed that the slope of the regression line is different for every fluid but it is always negative, which is consistent to the fact that the increase in Z_{Real} produces an entropy variation (due to frictional losses) such that q_{circ} increases, and since q_{Evap} is kept constant, it consequently leads to a decrement in COP_{TD}. However, since the magnitudes of the change of q_C and q_a in comparison to the other two heat contributions are small, the highest COP_{TD} reductions are just 1.67% for RC318 and 1.60% for R134a, which represents a very slight decrease. The increase on k also causes a reduction of G_C and consequently of Q_Y, opposing to the increase in q_{circ}. The final result of the above described variations is that, depending on the thermophysical properties of each fluid, the increment in G_W varies from 1.57% up to 7.60%, which means that, according to Eq. (13), the effect of Q_Y on G_W is greater than that of q_{circ}.

A quite different effect on COP_{TD} is observed when T_{Cond} is increased: it can be seen from Fig. 4 that the decreasing trend is

Table 3
COP_{TD} and specific heats exchanged for the case of $T_{cond} = 313$ K and the change in the corresponding values while increasing T_{cond} to 348 K.

	q_{Cond}	q_a	Δq_{Cond}	Δq_a	q_{Evap}	Δq_{Evap}	q_c	Δq_c	Δq_{Circ}	G_W	ΔG_W	COP_{TD}	ΔCOP_{TD}
RE134	192.93	5.60	−14%	17%	193.30	−9%	28.90	121%	3%	0.680	−4.7%	9.43	−50%
R245ca	192.68	5.73	−11%	14%	190.62	−8%	25.27	122%	2%	0.313	−5.3%	9.82	−51%
R245fa	181.73	5.63	−13%	15%	180.34	−10%	25.20	121%	−1%	0.802	−2.6%	9.60	−51%
R114	121.95	4.55	−14%	18%	120.64	−9%	17.18	128%	2%	0.561	−4.5%	9.65	−51%
R236ea	147.28	5.62	−16%	17%	146.52	−10%	21.83	127%	1%	0.606	−3.2%	9.64	−51%
R152a	259.92	7.71	−23%	24%	269.95	−11%	48.92	134%	4%	0.413	−5.1%	9.66	−52%
R236fa	136.69	5.57	−19%	20%	136.73	−12%	21.68	131%	3%	0.449	−4.2%	9.51	−52%
RC318	97.10	5.32	−22%	23%	96.28	−14%	16.38	136%	0%	0.859	−3.0%	9.11	−52%
R227ea	102.32	5.74	−28%	27%	104.19	−15%	20.38	141%	3%	0.426	−4.4%	9.16	−53%
R134a	163.00	6.50	−29%	32%	170.19	−14%	33.77	142%	0%	0.496	−3.6%	9.47	−54%

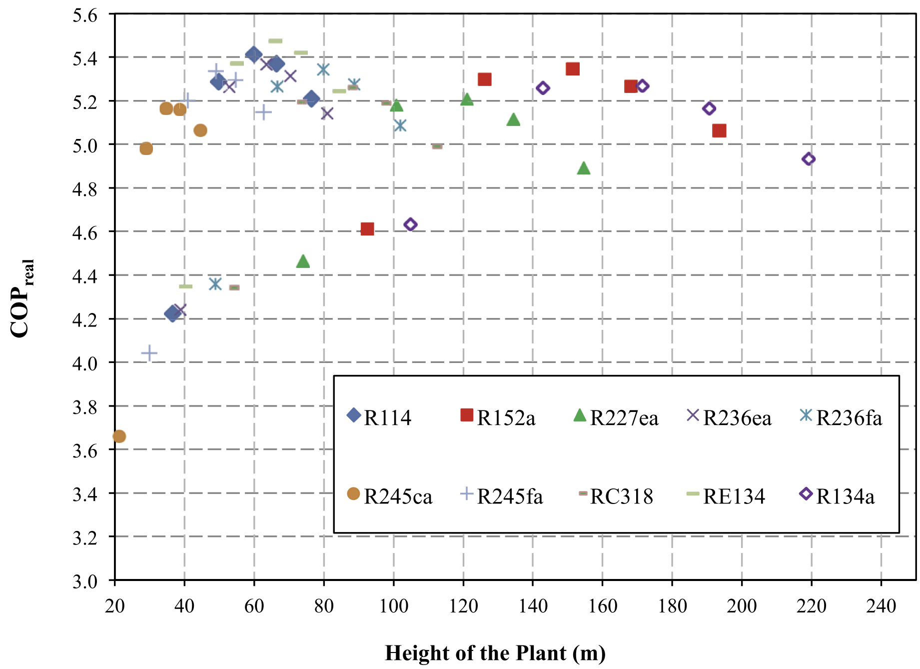

Fig. 6. Variation of the theoretical thermodynamic cycle of the TGHP working fluid as T_{Cond} is increased, fluid R245ca.

almost identical for all fluids, and for the temperature range evaluated is as high as 51.7%. Following Eq. (16), this behavior can be explained due to the simultaneous increase in Q_{Circ} and the reduction in Q_{Evap}. Fig. 5 depicts the changes in the shape and area of the TGHP cycle drawn on the T-s chart as T_{Cond} is increased, when R245ca is used as the working fluid. It can be deduced that q_{Cond} decreases as T_{Cond} rises due to the narrowing of the fluid liquid−vapor coexistence region at higher temperature, then T_{Cond} displacement causes a reduction of q_{Evap} via the regeneration process. The increase on T_{Cond} causes both q_C and q_a to be augmented, finally implying a slight decrease of G_W. Table 3 shows the variation of COP_{TD} among the investigated fluids in two different conditions of $T_{Cond} = 313$ K and $T_{Cond} = 348$ K. The first columns demonstrate the corresponding values of q_{Cond}, q_a, $q_{Evap}q_C$, G_W and COP_{TD} in a reference condition ($T_{Cond} = 313$ K), while the next columns demonstrate the corresponding percentage of change in each of the mentioned parameters while operating at the second condition ($T_{Cond} = 348$ K) compared to the first condition ($T_{Cond} = 313$ K).

When analyzing Table 3 it can be seen that the difference upon the thermophysical properties of each fluid is mainly observed on

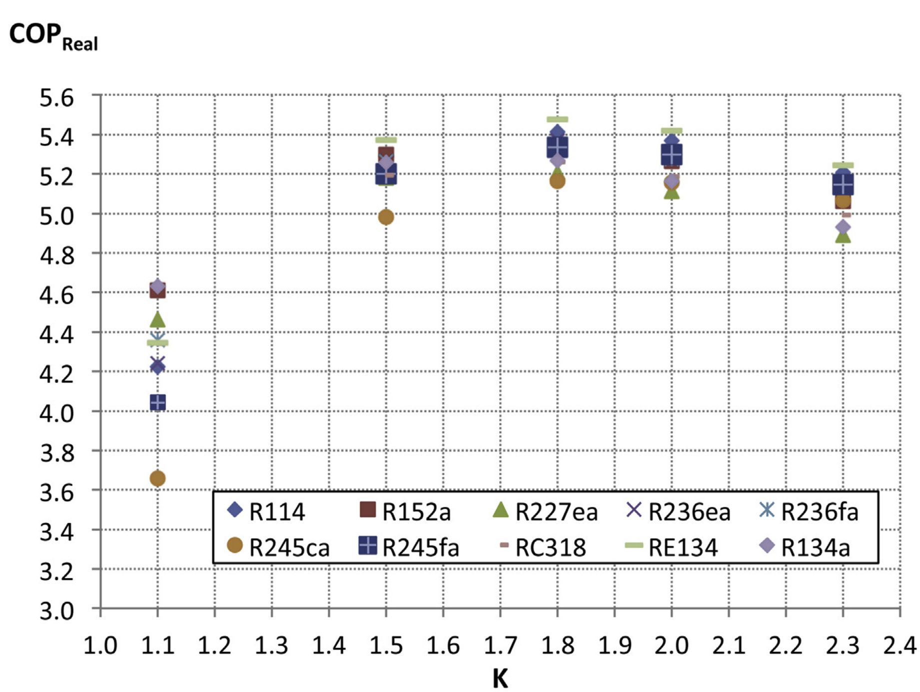

Fig. 7. Variation of GC as a function of k.

Table 4
The height increasing factors leading to maximum achievable COP for each fluid and the corresponding required height and the COP increment with respect to k = 1.1 condition.

	COP_{Real}^{max}	Z_{Real}^{max} (m)	k_{max}	ΔCOP_{Real}
R245ca	5.167	35.00	1.811	29%
R245fa	5.337	48.23	1.769	24%
R114	5.414	58.63	1.765	22%
R236ea	5.370	61.50	1.747	21%
RE134	5.478	64.06	1.746	21%
R236fa	5.349	76.57	1.727	18%
RC318	5.270	83.09	1.717	18%
R227ea	5.228	112.88	1.680	15%
R152a	5.358	143.14	1.702	14%
R134a	5.318	158.16	1.660	13%

the different values obtained for G_W whose value can be almost doubled when comparing the fluids with the two extreme values, meanwhile the COP_{TD} shows small variation. Actually when analyzing Fig. 4 it can be seen that basically all the fluids follow the sameCOP_{TD} trend, moreover, under the same conditions all the fluids have similar COP_{TD} values, and the increment from the lowest to the highest COP_{TD} varied from 8% to 13%.

Therefore, it can be deduced that for the analyzed substances, under fixed input conditions, the COP_{TD} of the TGHP could be estimated to be into a range, and the greatest improvement that could be achieved by changing the working fluid was of about 10% with respect to the lowest value.

After analyzing the effect of height increasing factor and the condensation temperature on the thermodynamic coefficient of performance, the same investigation has been next performed for the real COP. Fig. 6 shows the variation of COP_{Real} as a function of the plant height (which comes from the variation of k for each fluid). It can be clearly noticed that the selection of the working fluid should be made according to the specific building size. For the lowest heights, R245ca seems to be the best choice. Furthermore, from the interrelation of the different profiles of the results it can be concluded that within the range that goes from 30 m up to around 195 m, the fluid spectrum will always present an option with a COP_{Real} higher than 5. This implies that for buildings within the mentioned height range the TGHP might be a promising option. Even though all the analyzed fluids except R227ea have a specific volume higher than HFC134a, which in the end causes an increase of the duct sizes (given the fixed input velocity) and of G_C (in order to obtain the required ΔP), it can be seen that this phenomena has no noticeable impact on COP_{Real}.

Apart from minor differences, COP_{Real} curves, as a function of the plant height for the different fluids, follow nearly the same trend, and they reach closely similar maximum values at what could be called the optimum conditions for each one of the analyzed fluids. It can be therefore deduced that COP_{Real} is strongly related to the parameter k as it can be clearly seen on Fig. 7 which presents the results for all the analyzed fluids. Moreover, when considering the theoretical minimum height for each fluid, it can also be found out that this optimum value at which COP_{Real} is maximized lies within a height increasing factor range of k = 1.6–1.8 for all the analyzed fluids. Table 4 demonstrates the maximum obtainable COP (COP_{Real}^{max}) for each fluid together with the corresponding height increasing factor (k_{max}) and the required height (Z_{Real}^{max}) while T_{Cond} = 313 K is kept constant. The COP enhancement percentage ΔCOP_{Real} with respect to the first condition (k = 1.1) has also been given in order to demonstrate the corresponding increase in the efficiency of the system. The explanation of this behavior can be seen in Eq. (15) and Eq. (17), asCOP_{Real} is a function of the carrier fluid's flow rate G_C. Under fixed temperature limits, as k increases so does Z_{Real}, and from Eq. (3) it is evident that this implies a lower$\overline{\rho}$ in the T-PD in order to get the same ΔP and consequently G_C diminishes. It was found that the pumping power's L_P dependency on k behaves in such a way that for low values of k (close to 1.8), as it

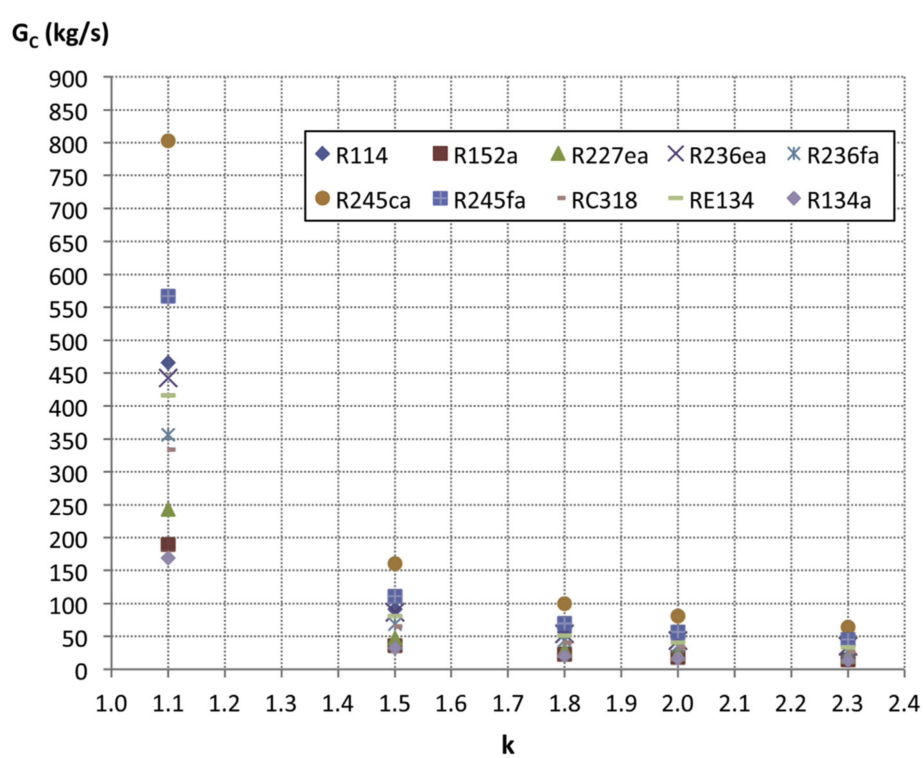

Fig. 8. Variation of the COP_{Real} as a function of k.

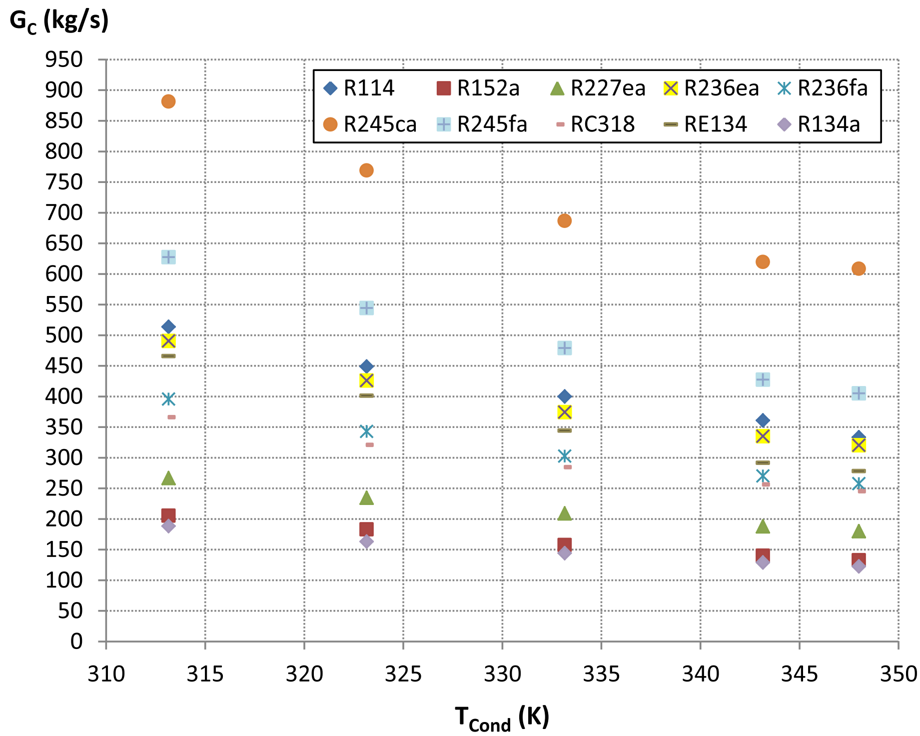

Fig. 9. Variation of GC as a function of T_{Cond}.

can be observed in Fig. 8, L_P also decreases since G_C decrease is abrupt. Approximately, over $k = 1.8$ the steepness of the slope of the G_C curve considerably reduces and the influence of H_P becomes dominant, with a consequent increment of L_P. Since Q_U is a fixed design parameter, COP_{Real} varies in inverse proportion to L_P, which means that as k increases COP_{Real} reaches a maximum and then starts decreasing.

In the next step, the effect of variation in the condensation temperature on the real coefficient of performance is studied. By increasing T_{Cond}, Z_{Real} is also augmented because of the increase in ΔP, however, as was previously stated, the combination of the effects of q_C and q_{Cond} cause q_{Circ} to experience little variation and consequently the variations on G_W oscillate from −5.26% to −2.60%. The overall effect of the increasing T_{Cond} causes ρ_W to decrease, and as a generalized result G_C decreases following a quasi-linear trend (Fig. 9), which as a matter of fact represents a much slower drop than the one observed for k variations.

This much smaller rate of reduction of G_C, compared to the previous case, is the reason for such a different behavior between Figs. 7 and 10. In the latter, the effect of the larger Z_{Real} is dominant

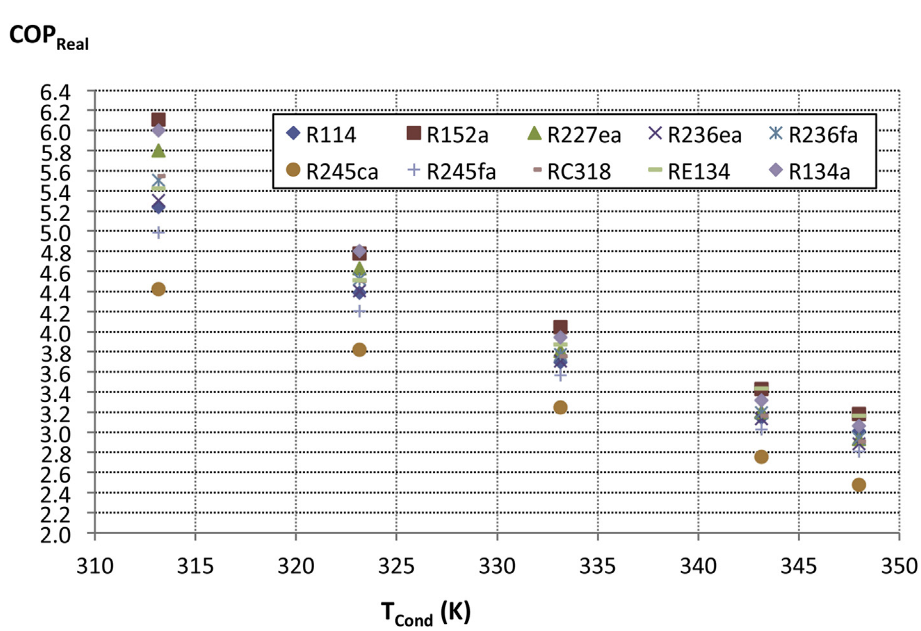

Fig. 10. Variation of the COP_{Real} as a function of TCond.

through H_P such that L_P curve is always increasing, and COP_{Real} always decreases. From where it can be quantified how an increase on T_{Cond} causes a severe penalization on COP_{Real}: for the analyzed cases the increase of 35 K resulted in a mean decrease of 45.8%, with a maximum of 49.4% for R227ea, values which are similar to the corresponding reduction in COP_{TD}.

6. Conclusions

The performance and the required plant height of a thermogravimetric heat pump were investigated using several refrigerants as working fluids. Nine fluids were selected and a diagram indicating the most suitable fluid, in terms of COP_{Real}, and according to the height of the building, was provided. It was also evidenced that R245ca should give the best performance in the height range of 20–35 m. Furthermore, it was also shown that within the range 30–195 m, a specific fluid can always be chosen which offer a real COP_{Real} higher than 5.

The height reduction offered by the investigated fluids with respect to the broadly used HFC134a refrigerant proved to be fairly independent from the condensation temperature, thus granting flexibility to the system.

The results of a few parametric analyses also demonstrated that the height increasing factor k does not significantly affect the resulting thermodynamic (COP_{TD}), as for the analyzed fluids the greatest decrease was of 1.67% for RC318 as k was increased from 1.1 to 2.3.

On the contrary and as it could be expected, COP_{TD} was confirmed to be quite sensitive to the temperature range, in such a way that an increase of 35 K results in reducing COP_{TD} down to approximately half of its initial value. Moreover, it was observed that for fixed input conditions, COP_{TD} changed within a narrow range between the analyzed fluids: the greatest improvement that could be achieved by changing the working fluid was only 10%.

Concerning the global COP (COP_{Real}), it was proved to be strongly related to the plant height. The pressure behavior of the fluid was observed to have the major influence – also due to fluid dynamics issues in the downward two-phase flow – with effects larger than the ones related to the entropy and enthalpy differences between the fluids. As k increases, the behavior of COP_{Real} is such that a maximum is reached and then it starts decreasing. The value of k that maximizes COP_{Real} can be found for each fluid and for all the investigated fluids it was within the range of 1.6–1.8. The effect of an increase of T_{Cond} is on the contrary was demonstrated to cause a severe penalization on COP_{Real}; as for the analyzed cases an increase of 35 K resulted in a mean decrease of about 46% (consistent with the COP_{TD} reduction).

References

[1] Administration USEI, Annual energy outlook, in: Energy USDo, 2012. Washington, DC 205852012.

[2] M. Orme, Estimates of the energy impact of ventilation and associated financial expenditures, Energy Build. 33 (3) (2001) 199–205.

[3] N. Pardo, Á. Montero, A. Sala, J. Martos, J.F. Urchueguía, Efficiency improvement of a ground coupled heat pump system from energy management, Appl. Therm. Eng. 31 (2–3) (2011) 391–398.

[4] Agency IE, International Energy Agency, 2012. www.worldenergyoutlook.org/ (accessed Sept 2013).

[5] V.N. Šarevski, M.N. Šarevski, Energy efficiency of the thermocompression refrigerating and heat pump systems, Int. J. Refrig. 35 (4) (2012) 1067–1079.

[6] M. Zago, A. Casalegno, R. Marchesi, F. Rinaldi, Efficiency analysis of independent and centralized heating systems for residential buildings in Northern Italy, Energies 4 (11) (2012) 2115–2131.

[7] S. Arosio, A. Muzzio, M. Silvestri, G. Sotgia, A thermogravimetric pilot plant for the production of mechanical energy from low enthalpy sources, in: 2nd International Symposium on Development and Utilization of Geothermal Energy. San Francisco, USA, 1975.

[8] S. Arosio, M. Balestri, F. Bonfanti, G. Sotgia, Experimental results of a non conventional facility thermogravimetric system, in: 4th Miami International Conference of Alternative Energy Sources. Miami, USA, 1981.

[9] S. Arosio, R. Carlevaro, Non-conventional thermodynamic converters for low temperature geothermal applications, Geothermics 32 (3) (2003) 325–348.

[10] S. Arosio, R. Carlevaro, Characteristics of thermogravimetric plants, J. Energy Resour. Technol. Trans. ASME 124 (2002).

[11] S. Arosio, R. Carlevaro, Thermogravimetric heat pumps, in: 55° Congresso Nazionale ATI. Bari-Matera, Italy, 2000.

[12] S. Arosio, R. Carlevaro, M. Guilizzoni, Downward two-phase flow applica-tions for a non-conventional heat pump, Int. J. Refrig. 27 (6) (2004) 629–638.

[13] S. Arosio, M. Guilizzoni, R. Carlevaro, Main characteristics of a non conventional refrigerating plant, in: V Minsk International Seminar "Heat Pipes, Heat Pumps, Refrigerators". Minsk, Belarus, 2003, pp. 8–11.

[14] S. Arosio, R. Carlevaro, M. Guilizzoni, Influence of two-phase flow on the performances of thermogravimetric heat pumps, in: 10th International Conference on Multiphase Flow in Industrial Plant, Centro Editoriale e Librario UNICAL, Tropea (VV), Italy, 2006, pp. 233–244.

[15] J.R.K. Yijun, A study on void fraction in vertical co-current upward and downward two-phase gas–liquid flow – I: experimental results, Chem. Eng. Commun. (1993) 221–243.

[16] H. Goda, T. Hibiki, S. Kim, M. Ishii, J. Uhle, Drift-flux model for downward two-phase flow, Int. J. Heat Mass Transf. 46 (25) (2003) 4835–4844.

[17] O.N. Kashinsky, V.V. Randin, Downward bubbly gas–liquid flow in a vertical pipe, Int. J. Multiphase Flow 25 (1) (1999) 109–138.

[18] M.A. Woldesemayat, A.J. Ghajar, Comparison of void fraction correlations for different flow patterns in horizontal and upward inclined pipes, Int. J. Multiphase Flow 33 (4) (2007) 347–370.

[19] V.T. Davis, The effect of the Froude number in estimating vertical two-phase gas–liquid friction losses, Br. Chem. Eng. 8 (7) (1963) 467.