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Annual energy performance of R744 and R410A heat pumping systems

Zhequan Jin[a], Trygve M. Eikevik [a], Petter Nekså [b], Armin Hafner [a], Ruzhu Wang [c]

[a] Norwegian University of Science and Technology, Trondheim, 7491, Norway, e-mail: zhequan.jin@ntnu.no
[b] SINTEF Energy Research, Trondheim, 7465, Norway
[c] Shanghai Jiao Tong University, Shanghai, 200240, China

ABSTRACT

This work compares the annual energy performance of heat pumping systems using R744 and R410A as refrigerant. Focus is the annual energy efficiency of R744 hybrid ground-coupled heat pumping system. The hybrid system uses both ambient air and ground borehole as heat sinks in the cooling mode. This is important to eliminate the underground thermal accumulation phenomenon in warm climates. Several dynamic models of heat pumping systems, using R744 and R410A as refrigerant, have been developed. Simulation results show that the annual COP_c and COP_h of an R744 hybrid system reaches 3.55 and 3.32, and its cooling performance is 42% better than for a R744 ASHP and 23% better than for a R744 GCHP system. The annual energy performance factor of a R410A ASHP system is better than for a R744 hybrid system, but the COPc for the R410A system seriously decreases when the ambient temperature is higher than 30 ºC.

Keywords: R744/CO_2; R410A; Hybrid ground-coupled heat pump; Energy efficiency

1. INTRODUCTION

Environmental sustainability and energy conservation have become the key issues facing the development of modern society (Omer., 2008). This is the reason that 195 countries' delegation gathered at Paris in December of 2015 and committed that the world to limit a rise in global temperature this century below 2°C. One of the important strategies to achieve this goal is energy efficiency improvement in the different society sectors. For example in the building sectors, it is generally considered that HVAC system occupies around 40% of a commercial building's total energy usage. The chiller or boiler consumes most of the energy in the HVAC system. So the energy efficiency improvement of a heat pumping system, which can generate not only cooling or heating capacity but also combined cooling and heating capacity, is critical to reduce the energy consumption in the building sectors.

In the past 80 years, the synthetic halocarbon refrigerants had wide applications. For example in the industrial refrigeration, commercial refrigeration, mobile air condition, and space air conditioning area. However, halocarbons' high ozone depletion potential or global warming potential limited their further development, as agreed in Montreal and Kyoto protocol. According to IPCC's data, synthetic halocarbon refrigerants' contribution to global warming accounting for 11% of all anthropogenic radiative forcing. This is due to their chemical structure result in much higher global warming potential comparing with natural refrigerant like R744

(CO2). For example, the GWP value for R410A is 1725 (Forster and Ramaswamy., 2007) which means the global warming potential is 1725 times the effect of CO2. So, in European region, the F-gases regulation will phase down the total amount of the frequently-used HFC refrigerant sales in steps to one-fifth of 2014 in 2030. This is due to the high GWP value of the HFC refrigerant

In reality, CO_2 as a natural refrigerant shows great potential as an important refrigerant in the future because of its environmental characteristics and superior thermodynamic properties. Modern utilization of CO_2 in a transcritical cycle was first proposed by Lorentzen in 1990, which is a turning point for revival of CO_2 as a refrigerant. Lorentzen also proposed a complete solution to replace HFC with natural refrigerant [1991?] So far many examples show that CO_2 refrigerant has successfully applied and commercialized in the supermarket refrigeration and heat pump water heater. On the other hand, the combination of sustainable energy technology with environment friendly refrigerant will be an important trend for the development of refrigeration, air conditioning and heat pump industry. Jin et al. (2016) has proposed a concept of CO_2 hybrid ground coupled heat pumping system, and showed the combined COP of the system varies from 3.0 to 5.5 when taking both space conditioning and service hot water supply into account. However, there is little research on the comparison of R410A and CO_2 heat pumping system performance for the commercial building application.

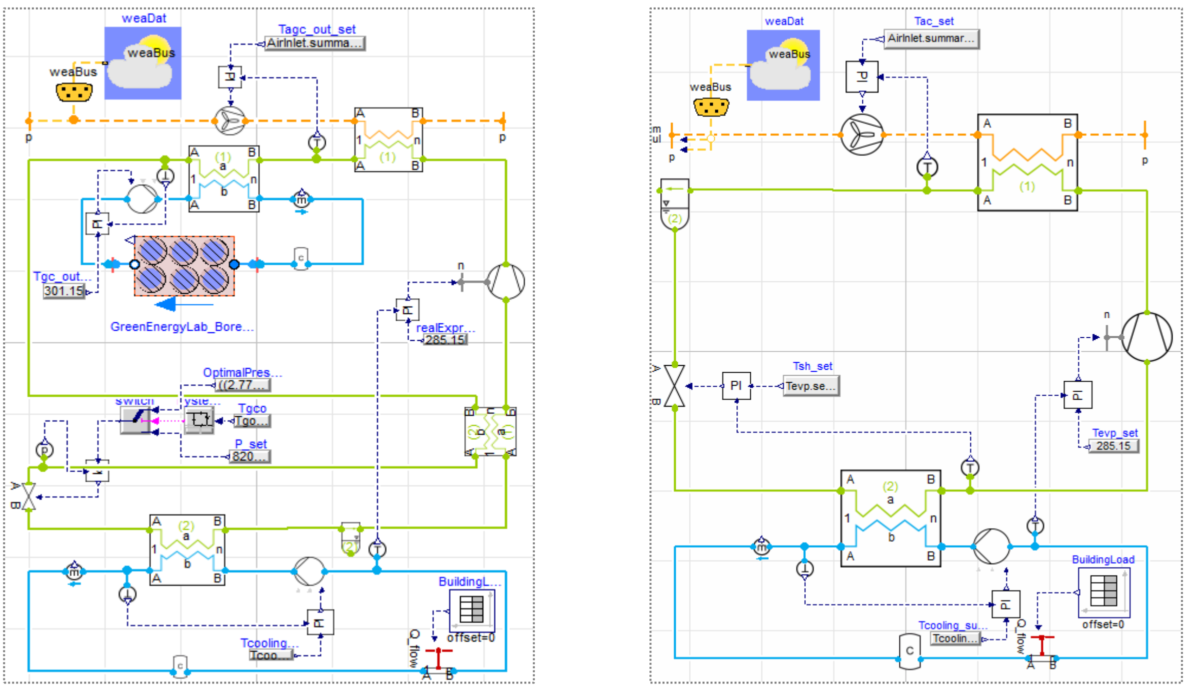
This study involves 4 different heat pumping systems, all of which are able to satisfy both space cooling and heating capacity. Moreover the dynamic models of CO_2 hybrid GCHP, CO_2 GCHP and CO_2 ASHP are developed for R744 system, while the dynamic model of R410A ASHP system is constructed based on a commercialized product. Then the system annual performance is simulated and compared for one reference cooling and heating season.

2. METHODOLOGY

The energy efficiency comparison conducted in this work is based on transient simulation results. The transient dynamic models take as much as practical factors into account, for example the efficiency curve of compressor, geometry and heat transfer coefficient of heat exchangers, ambient environmental conditions, etc. This aims to insure the system operating characteristic of the developed dynamic models is comparable with that of field heat pumping facility.

2.1 System configuration

In this section, the configuration of R410A and R744 heat pumping system will be showed and illustrated. Fig.1 (a) shows the schematic diagram of the cooling mode of R744 hybrid GCHP system, which includes R744 refrigerant loop, heat transfer fluid loops and system control strategy loops. For the R744 refrigerant loop, CO_2 circulates in the necessary typical refrigeration components, which submits to the classic transcritical refrigeration cycle (Lorentzen., 1990). So it allows the gas cooling heat rejected under a certain temperature glide to ambient heat sinks. This characteristic of CO_2 transcritical cycle shows the advantage in improving energy efficiency by utilizing two gas coolers to reject the heat into two temperature levels of heat sinks. This idea is fully presented in high pressure side of CO_2 system as showed in Fig.1 (a), which firstly rejects the gas cooling heat to ambient air under relatively high refrigerant temperature, and then the ground borehole is used to remove the part of gas cooling heat with relatively low refrigerant temperature. So the R744 hybrid GCHP system mentioned in this work refers to that which integrates both air-cooled gas cooler and ground-cooled gas cooler in the system. Similarly, the R744 GCHP or R744 ASHP refers to the system that rejects the gas cooling heat only into the ground borehole or ambient air. However, the CO_2 system control strategies are almost the same for the different systems. For example, the optimal gas cooler pressure control based on the CO_2 outlet temperature from gas cooler, variable compressor speed control based on the cooling water return temperature and constant cooling water mass flow control based on return and supply temperature difference are applied to these 3 systems. In addition, the air volume flow control and water mass flow control of air-cooled and ground-cooled gas cooler are based on the CO_2 outlet temperature from respective gas coolers. At last, all of the models of heat pimping system showed in Fig.1 also integrated building cooling load and weather data for the simulation, both of them either from field test or building simulation results.



(a). R744 hybrid GCHP/GCHP/ASHP system (b). R410A ASHP system

Figure 1 Configuration of R744 and R410A heat pumping system for cooling mode

On the other hand, Fig.1 (b) shows the schematic diagram of R410A ASHP dynamic model. As well known, the normal refrigeration cycle of R410A is operated under subcritical phase. So the control strategy of R410A system's expansion valve will be different from that of R744 system. As observed, the R410A ASHP system control the openness of expansion valve by the overheated degree of refrigerant at outlet of evaporator, as thermal expansion valve or electronic expansion valve does. In fact, the R410A ASHP model is constructed based on the practical commercialized system with same boundary conditions from the manufactures, and the necessary model validation process is also conducted in next section.

2.2 System simulation boundary conditions

Since the dynamic models of different systems are constructed to compare the energy efficiency, it is of importance to clarify the reasonable restricted conditions for the components and environmental conditions of the investigated systems. Actually, the nominal cooling capacity (43kW) of 4 systems are equally sized based on the laboratory existed reference R410A ASHP system, which is installed in a warm climate city of China. So it is possible to select some component specifications from the existed system, like the type of compressor, heat exchangers, hydraulic pump, and air side pump as well as system control strategy. Moreover, the theoretical design procedure according to the nominal operating conditions can be used to calculate the size of R744 compressor and geometry of air-cooled or water-cooled heat exchangers. So Table 1 lists the component specifications of the dynamic model of R744 and R410A heat pumping systems both from the theoretical calculation and existed system information.

Table 1. Simulation boundary condition

Components	Type and specification for R744 system	Type and specification for R410A system
Compressor	B***_hgx46_345_4s_50Hz with variable frequency control	C***_ZP83KCE_TFD_50Hz with variable frequency control
Air-cooled gas cooler/ Condenser	Fin and tube heat exchanger	Fin and tube heat exchanger
Water-cooled gas cooler/ Condenser	Plate heat exchanger	Plate heat exchanger
Water-cooled evaporator	Plate heat exchanger	Plate heat exchanger
Expansion device	Back pressure control valve with optimal control strategy	Thermal expansion valve with fixed superheat
Borehole parameters	9 boreholes in green energy laboratory in SJTU, China (Yu.,2011)	
Environmental condition	Weather and underground conditions for Shanghai, China	
Cooling water pump	Variable-speed pump with 0.9~3.0 l/s and 20~120 kPa	
Air side fan	Axial Flying Bird IV with rotating shroud, 3800 l/s	

In addition, Table 2 lists the detail information of different heat exchangers used for heat pumping systems, which are also from the theoretical calculation and manufacturer information. To be mentioned, the heat transfer coefficients of CO_2, R410A and H_2O are chosen as averaged values from the literatures. (Park and Hrnjak., 2007, Thome and Ribatski., 2005)

Table 2. Simulation boundary condition for different heat exchangers

Parameters	CO_2-hybrid GCHP	CO_2-GCHP	CO_2-ASHP	R410A-ASHP
α_{CO2}, α_{R410A},α_{H2O} in plate HX	α_{CO2}=2500W/m^{2}K α_{H2O}=2500W/m^{2}K	α_{CO2}=2500W/m^{2}K α_{H2O}=2500W/m^{2}K	α_{CO2}=2500W/m^{2}K α_{H2O}=2500W/m^{2}K	α_{R410A}=2500W/m^{2}K α_{H2O}=2500W/m^{2}K
Evaporator (43kW)	K***_C095*72 (43.9kW)	K***_C095*72 (43.9kW)	K***_C095*72 (43.9kW)	K***_K200H*30 (43.95kW)

Gas cooler - GHX	K***_C097*48 (26.37kW)	K***_C097*88 (52.74kW)	N/A	N/A
Borehole length (q=35W/m)	585m (65m*9)	1775m (195m*9)	N/A	N/A
α_{CO2},α_{R410A}, α_{air} in AHX	α_{CO2}=2500W/m^{2}K α_{air} by Haaf model	N/A	α_{CO2}=2500W/m^{2}K α_{air} by Haaf model	α_{R410A}=2500W/m^{2}K α_{air} by Haaf model
Gas cooler/ Condenser-AHX (Tube D_i=7mm)	length=1.2*5*20 m	N/A	length=2.2*6*20 m	length=2*6*25 m

As a whole, the defined simulation boundary conditions aim to insure the operating characteristic of the dynamic models is comparable with that of field heat pumping facilities. On the other hand, these restricted conditions of the components and environment are used to make system comparison is in a reasonable level.

3. ANNUAL ENERGY PERFORMANCE COMPARISON

The system performance is simulated and compared for one reference cooling and heating season. So the cooling performance of 4 different systems is involved in this study. Firstly, the model validation process for R410A system is discussed, and then the detail energy efficiency and operating characteristic with corresponding reason are presented.

3.1 Model validation for R410A system

Fig.2 shows the seasonal cooling and heating load of the reference building under selected climatic conditions. These data have been retrieved from Energy Plus for a reference hotel building based on PNNL's study (Goel. et al, 2014). The simulation period is selected from late May until late September for cooling season, while from early November to late march for heating season. The climate data corresponds to Shanghai, which locates in the warm climate zone in China. Since frequent variation of the cooling or heating load is critical for numerical calculation of the Dymola software, the load value showed in Fig.2 is acquired through 1 day's moving average of the practical calculated building load. To be mentioned the dehumidification load is not considered in this study, which means that the cooling load need to be removed by heat pumping system to control the building space temperature.

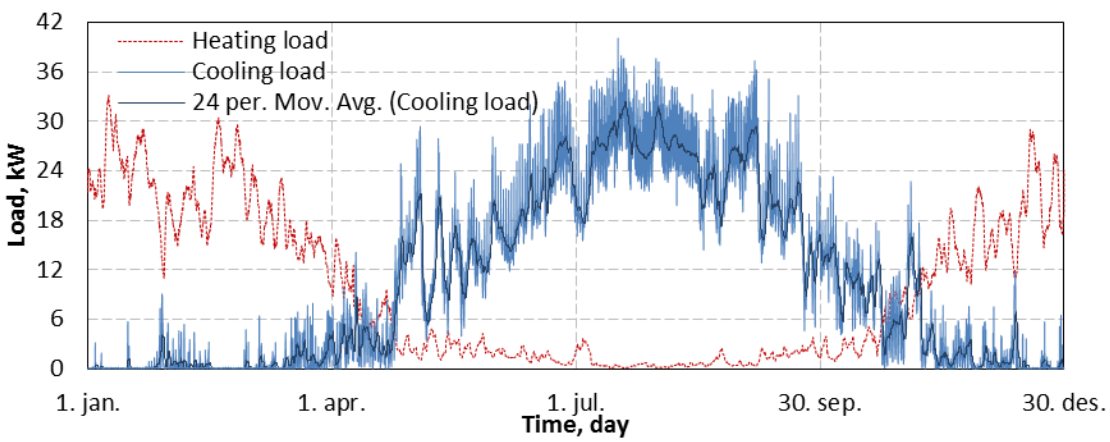


Figure 2 Seasonal cooling and heating load of the building

The R410A ASHP model validation with field test experimental data is showed in Fig.3 and Fig.4. The cooling seasonal simulation results of COP_c for R410A system are hourly pointed with different ambient temperature. Meanwhile, the randomly selected field test R410A unit COPc values are also plotted in Fig.3. Then the 2nd polynomial trend lines were made, and the Eq.2 and Eq.3 shows the corresponding COP_c calculation equations with the degree Celsius value of ambient air temperature.

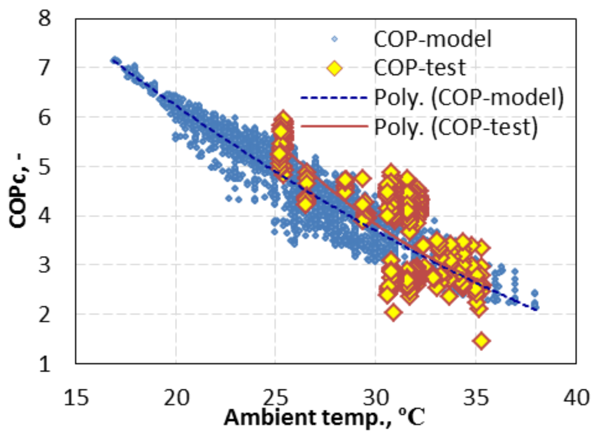


Figure 3 Model and test COP_c for R410A ASHP

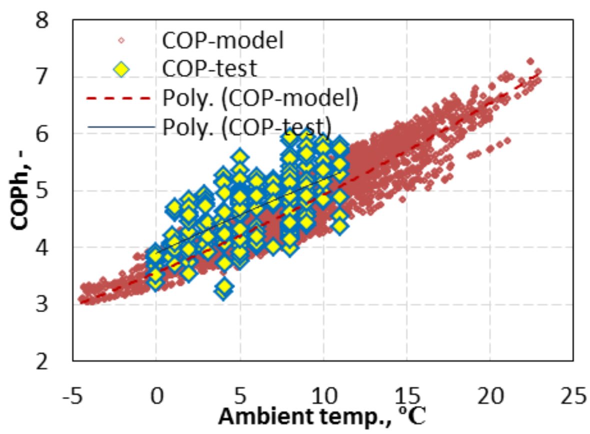


Figure 4 Model and test COP_h for R410A ASHP

$$COP_{c-model} = 0.003T_{air}^2 - 0.4019T_{air} + 13.098,\ T_{air} \in [17 \sim 38°C] \tag{2}$$

$$COP_{c-test} = 0.0123T_{air}^2 - 1.0188T_{air} + 23.371,\ T_{air} \in [26 \sim 35°C] \tag{3}$$

Based on the developed COPc correlations, the values of COP under different ambient air temperature have been calculated and compared. It is shown that the difference is less than 10% in the compared temperature range, and the difference is decreasing to around 3% when the ambient temperature is high. So the difference between the model and measurements for the R410A ASHP is lower than 10%.

Table 3. Quantitive comparison of model and test COPc for R410A ASHP

Ambient air temp.,°C	**COP-model, -**	**COP-test, -**	**Error**
26	4.6766	5.197	10.0 %
27	4.4337	4.8301	8.2 %
28	4.1968	4.4878	6.5 %
29	3.9659	4.1701	4.9 %
30	3.741	3.877	3.5 %
31	3.5221	3.6085	2.4 %
32	3.3092	3.3646	1.6 %
33	3.1023	3.1453	1.4 %
34	2.9014	2.9506	1.7 %
35	2.7065	2.7805	2.7 %

Similar with cooling seasonal simulation results, the COP_h for R410A system are hourly pointed with different ambient temperature. Meanwhile, the randomly selected field test R410A unit COP_h values are also plotted in Fig.4. In the same way, the Eq.4 and Eq.5 show the corresponding COP_h calculation equations with the degree Celsius value of ambient air temperature.

$$COP_{h-model} = 0.0013T_{air}^2 + 0.1232T_{air} + 3.5601,\ T_{air} \in [-5 \sim 23°C] \quad (4)$$

$$COP_{h-test} = -0.0008T_{air}^2 + 0.1372T_{air} + 3.8982,\ T_{air} \in [0 \sim 11°C] \quad (5)$$

According to the developed COPh correlations, the values of COP under different ambient air temperature have been calculated and compared. It is shown that the difference is less than 10.2% in the compared temperature range, but the difference is worse than the validation results of cooling simulation results. However, the difference between the model and measurements for the R410A ASHP is lower than 10% as a whole.

Table 4. Quantitive comparison of model and test COPh for R410A ASHP

Ambient air temp.,°C	**COP-model, -**	**COP-test, -**	**Error**
0	3.5601	3.8982	8.7 %
1	3.682	4.0346	8.7 %
2	3.8013	4.1694	8.8 %
3	3.918	4.3026	8.9 %
4	4.0321	4.4342	9.1 %
5	4.1436	4.5642	9.2 %
6	4.2525	4.6926	9.4 %
7	4.3588	4.8194	9.6 %
8	4.4625	4.9446	9.8 %
9	4.5636	5.0682	10.0 %

3.2 Annual energy performance comparison

The annual energy efficiency of the different systems is compared using both instant and seasonal averaged COP values. This is because the instant COP value can indicate the transient variation of system performance, which is important to analyze how the system boundary conditions influence the system operation, and then support the information about the system operation. On the other hand, the seasonal averaged COP value indicates the annual energy efficiency of the heat pumping system.

Eq 6 shows the definition of the instant COP_c and COP_h. The power of the compressor and cooling/ heating capacity are used to calculate the COP value.

$$COP_c = \frac{\dot{Q}_c}{\dot{W}_{compr}} \text{ and } COP_h = \frac{\dot{Q}_h}{\dot{W}_{compr}} \quad (6)$$

Fig.5 and Fig.6 show the 4 different instant COP_c variations for the cooling and heating season. For the cooling season, it is obvious that the system performance is better at low ambient temperature, when the cooling load of reference building is also low. Note that the R744 hybrid GCHP and the R410A ASHP systems show a better energy performance than the R744 GCHP or the R744 ASHP systems. The R744 hybrid GCHP system is of course less influenced by the ambient air temperature comparing to the R410A ASHP. As the R744 hybrid system yields a higher COPc value during the period when the ambient temperature is highest. This is of course due to that the ground heat sink ensures stable and low temperature conditions, as showed in Fig.7.

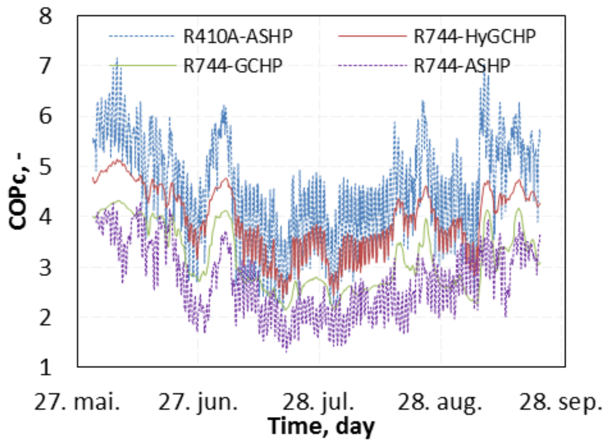


Figure 5 Seasonal cooling COP variation of different systems

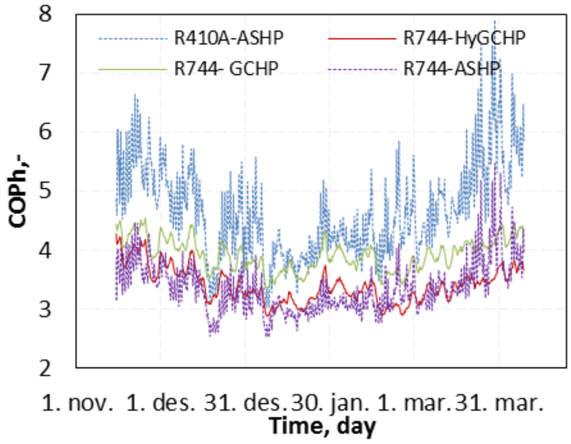


Figure 6 Seasonal heating COP variation of different systems

For the heating season, the system performance is better at higher ambient temperature, as showed in Fig.6. The R410A ASHP system shows the best energy performance. In addition, there are two reasons that R744 GCHP shows better performance than R744 hybrid GCHP system. First, only the ground borehole heat exchanger is used for heating operation of R744 hybrid GCHP system. Second, the length of the borehole heat exchanger is 3 times shorter that of R744 GCHP, as showed in Table 2. So the ground borehole wall temperature decreases more for the R744 hybrid GCHP system, as shown in Fig. 8. This is also the reason that the COPh of R744 ASHP is better than that of R744 hybrid GCHP system in the end of heating season, when the ambient temperature is high.

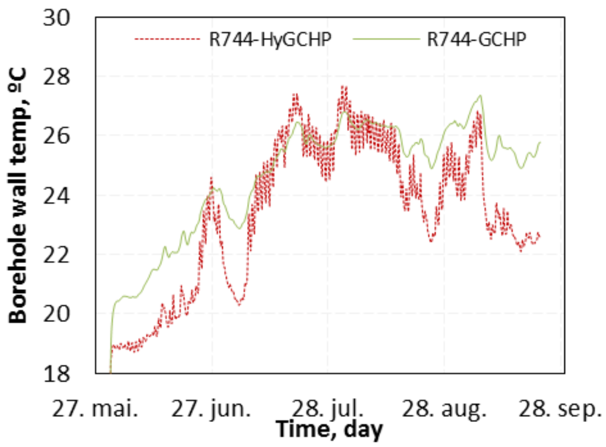


Figure 7 Seasonal borehole wall temperature variation in cooling season

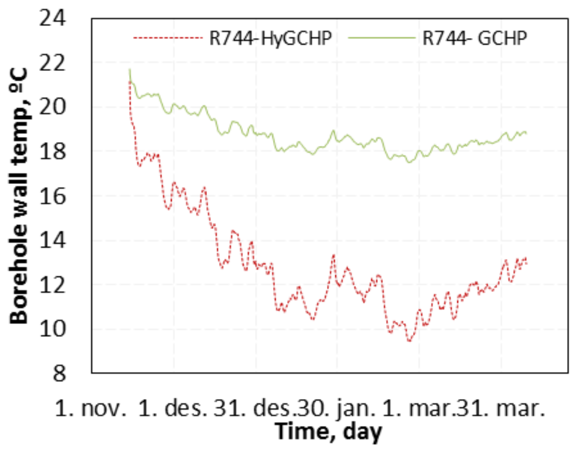


Figure 8 Seasonal borehole wall temperature variation in heating season

Fig.9 shows the seasonal rejected heat (condensing or gas cooling heat) amount of the different systems to the environment. The total heat rejection amount can indicate the different system performance, because the cooling loads are the same for the different heat pumping systems. It also shows 73% of the gas cooling heat of hybrid CO_2 GCHP system will be rejected to ambient air, which can effectively reduce the borehole heat accumulation and avoid the borehole wall temperature increment.

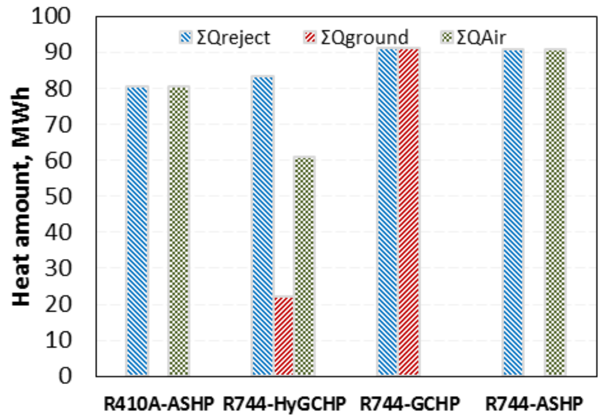


Figure 9 Heat rejection amount in cooling season

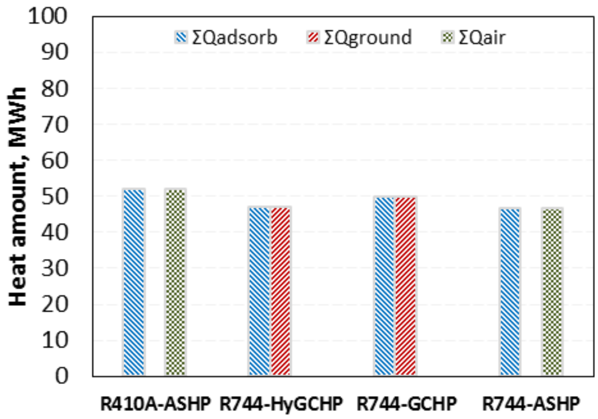


Figure 10 Heat absorption amount in heating season

Similarly, Fig.10 shows the seasonal absorbed heat (evaporating heat) amount of the different systems to the environment. The total heat absorption amount can indicate the different system performance, because the heating loads are the same for the different heat pumping systems. It can be observed that R410A and R744 ASHP system only absorb the heat from the ambient air, while both R744 HyGCHP and GCHP system absorb heat from the underground borehole. The main reason that R744 HyGCHP system absorbs less heat than R744 GCHP system is the smaller size of the borehole heat exchanger, which results in lowering evaporating temperature of the R744 HyGCHP system. Table 5 lists the detail heat rejection, heat absorption, and work consumption amount of the different heat pumping system.

Table 5. Heat rejection, heat absorption, and work consumption amount of heat pumping systems

Items / Systems	Cooling mode			Heating mode		
	Q_{reject},MWh	Q_{adsorb},MWh	W_{compr},MWh	Q_{reject},MWh	Q_{adsorb},MWh	W_{compr},MWh
R410A-ASHP	80.60	65.00	15.59	67.30	52.00	15.32
R744-HyGCHP	83.15	64.87	18.27	67.29	47.00	20.29
R744-GCHP	91.08	64.88	26.19	67.29	49.75	17.54
R744-ASHP	90.63	65.00	25.80	67.32	46.49	20.82

Based on the total annual heat rejection or absorption amount with work consumption amount, the seasonal averagedCOPc and COPh can be defined, as Eq 7 shows.

$$_{\text{seasonal averaged}}COP_c = \frac{\sum_{t=start}^{end} Q_c}{\sum_{t=start}^{end} W_{compr}} \text{ and } _{\text{seasonal averaged}}COP_h = \frac{\sum_{t=start}^{end} Q_h}{\sum_{t=start}^{end} W_{compr}} \quad (7)$$

Fig.11 shows the seasonal averaged COPc for the reference building. The seasonal COPc of the R744 hybrid system is 3.55, which is around 42% and 23% better than for the R744 ASHP and for the R744 GCHP system. The main reason is of course again, that high CO2 outlet temperature from the gas cooler will seriously decrease the system performance. The culprits are the high ambient air temperature and borehole heat accumulation. It also can be observed that the annual cooling performance of the standard R410A ASHP system is better than R744 hybrid system.

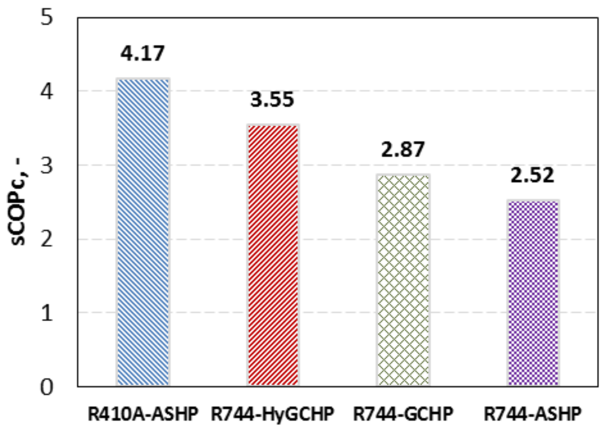


Figure 11 Seasonal cooling COP variation of different systems

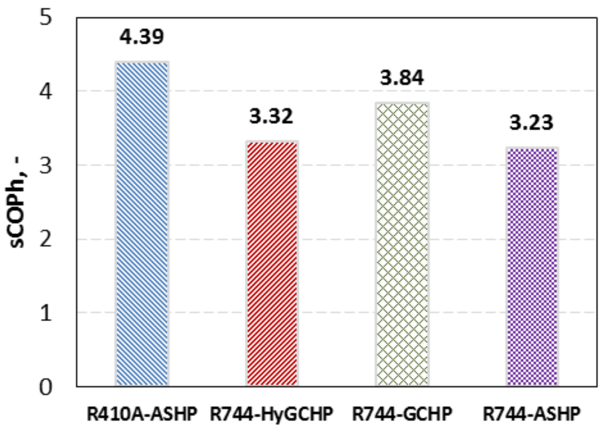


Figure 12 Seasonal averaged heating COP of different systems

Fig.12 shows the seasonal averaged COPh for the reference building. The seasonal COPh of the R744 hybrid system is 3.32. The main reason for the lower heating COP value is of course again, that the length of the borehole heat exchanger is 3 times shorter than that of R744 GCHP, as explained before. The culprit is the borehole heat dissipation. It also can be observed that the annual cooling performance of the standard R410A ASHP system is better than others under the used reference heating load.

4. DISCUSSION

The seasonal heat accumulation or dissipation underground will seriously influence the system performance of GCHP system. This is the main reason that the annual cooling performance of the R410A ASHP system is better than R744 HyGCHP system and others. Fig.13 and Fig.14 compared the annual borehole heat transfer rate variation for the R744 HyGCHP and pure GCHP systems. The heat transfer rate in cooling season is designed as same value (35W/m) for both systems, while heat transfer rate in heating season shows a huge difference under the same heating load profile. R744 HyGCHP has a much higher borehole heat transfer rate than R744 GCHP. This is because the length of the borehole heat exchanger is 3 times shorter, as explained previously.

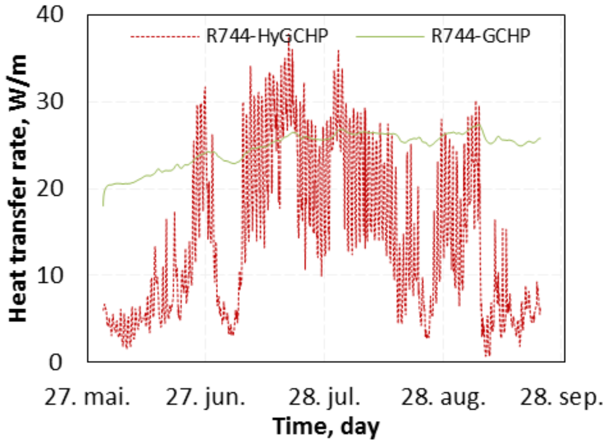


Figure 13 Seasonal borehole heat transfer rate in cooling season

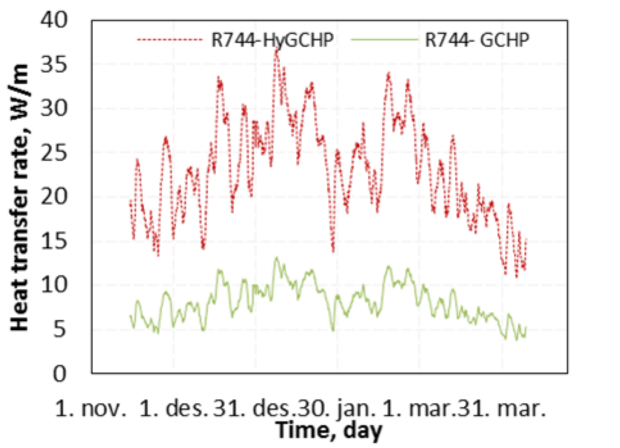


Figure 14 Seasonal borehole heat transfer rate in heating season

In order to investigate how the length of borehole influences the performance of R744 hybrid GCHP system, 3 kinds of borehole length are selected for the seasonal simulation. Fig.15 and Fig.16 show the seasonal averaged COP_c and COP_h values for the respective boundary conditions. It is obvious the system performance will be well increased when the borehole length is longer. For example, the $sCOP_c$ of R744 hybrid GCHP system is increased by 18% when single borehole length is increased from 65m to 195m. In addition, it is possible to further improve the $sCOP_h$ value to the same level of R744 GCHP with the longer length of borehole, as Fig.16 shows. Though the simulation results show that the heating performance is not as good as cooling, it is possible to improve the performance by integrating the service hot water supply (Jin et al., 2016).

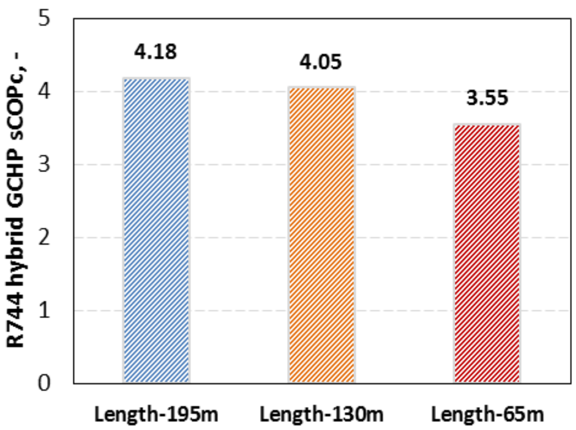


Figure 15 Seasonal averaged cooling COP for different length of each borehole

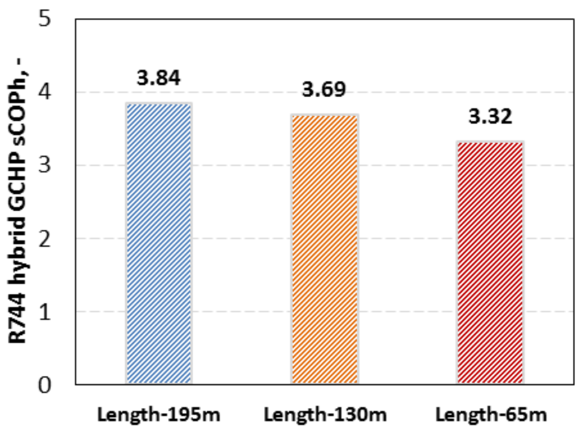


Figure 16 Seasonal averaged heating COP for different length of each borehole

On the other aspect, the main reason for the lower sCOPc of the R744 hybrid system comparing to the R410A system is that the simulated model is operated continuously, meeting the instant cooling demand. This is due to the fact that a hotel was chosen as reference building. A hotel must be cooled even during the night. The system performance of R410A system is much better when the ambient temperature is low –

during the night. However, for most other buildings, like offices, shopping malls, schools etc, the system will mostly be operated during daytime and high ambient temperatures. Thus it is possible to make a R744 hybrid GCHP system that can compete efficiency wise, with the R410A ASHP system when applied to other forms of buildings.

At last, the R744 systems investigated in this study utilize a classical CO_2 transcritical cycle, and there is still potential to improve system performance by decreasing the compressor loss and expansion loss adopting novel CO_2 refrigeration cycles. For example, the CO_2 ejector cycle can effectively decrease the expansion loss and thus improve the system performance, especially for the systems operated in a warm climate. Thus the potential to use CO_2 for air conditioning applications in warm climates is believed to be feasible when combined with a good heat sink for the gas cooling.

5. CONCLUTION

This work compared the annual energy performance of various heat pumping systems using R410A and R744 as refrigerant by means of annual simulation. The following results can be concluded:

1. Simulation results show that the seasonal COP_c and COP_h of R744 hybrid system are 3.55 and 3.32, and the cooling performance is 42% and 23% better than both R744 ASHP and R744 GCHP system.
2. Annual system performance of R410A ASHP system is better than R744 hybrid system, but the cooling performance of R410A system seriously decreases when the ambient temperature is high.
3. R744 system shows superiority on the integration of two heat sinks, and the system performance will be further improved by increasing the length of boreholes.

In the scope of this study, it is still less efficiency to use R744 for the space air conditioning heat pumping system comparing with R410A system. However, the hybrid utilization of heat sinks based on R744 transcritical cycle well improved the cooling energy efficiency, especially under warm climate. In addition, some smart technologies, like heat recovery from gas cooling process for service hot water heating and work recovery from expansion process by a ejector, can further improve the energy efficiency of the R744 heat pumping system.

NOMENCLATURE

Abbreviation

AC	Air conditioning	CO_2	Carbon dioxide
AHX	Air heat exchanger	GCHP	Ground coupled heat pump
ASHP	Air source heat pump	GWP	Global warming potential
COP	Coefficient of performance	HyGCHP	Hybrid GCHP

Subscripts and superscripts