PROCEEDINGS OF ECOS 2023 - THE 36TH INTERNATIONAL CONFERENCE ON
EFFICIENCY, **C**OST, **O**PTIMIZATION, **S**IMULATION AND ENVIRONMENTAL IMPACT OF ENERGY SYSTEMS
25-30 JUNE, 2023, LAS PALMAS DE GRAN CANARIA, SPAIN

Use of residual energy from underground infrastructures: Madrid – Sevilla metro station.

Susana Sánchez-Orgaz[a], Javier Muñoz Antón[b], Marina Tadeo Cañete[c] and Javier Rodríguez Martín[d]

a UPM, Madrid, Spain, susana.sanchez.orgaz@upm.es, ***CA***
b UPM, Madrid, Spain, javier.munoz.anton@upm.es,
c UPM, Madrid, Spain, m.tadeo@alumnos.upm.es,
d UPM, Madrid, Spain, javier.rodriguez.martin@upm.es

Abstract:

Cities have great potential to implement innovative solutions and improve energy efficiency and the use of non-conventional energy sources, laying the foundations for a new, more sustainable urban model. With this vision and the aim of finding solutions to this challenge, Madrid Subterra emerges. This work is within the project of this association created with the aim of promoting the exploration and exploitation of the potential of clean and renewable energy of the urban subsoil of Madrid. This study analyses the energy use of the Seville Metro station for the thermal supply of the Building of the Ministry of Environment of the Community of Madrid. The main goal is to extract the waste heat generated in the metro tunnels due to the traction and braking of machines, auxiliary facilities, or transit of people on the platforms and use it to produce domestic hot water (DHW) using a heat pump to supply the referred building. This concept could reduce the temperature in Metro facilities and save energy. From the tunnel the ventilation flow and temperatures data, the performance of the system has been estimated using the CoolPack program. The results obtained show that this system could supply domestic hot water for 7640 people per month with a COP of 2.663 and SCOP of 1.975.

Keywords:

Circular Economy; Energy Saving; Sustainability; Thermal supply; Waste Heat;

1. Introduction

Cities today and focusing on the specific case of the city of Madrid, has a great potential for waste heat that is generated in tunnels and platforms. This waste heat could be studied, extracted and implemented innovative solutions for improving energy efficiency and using non-conventional energy sources. This potential can be boosted to further reduce CO_2 emissions and make cities more sustainable. For all the above, the Madrid Subterra Association arises, created with the aim of promoting the exploration and exploitation of the clean and renewable energy potential of Madrid's urban land, allowing to transform the current vicious circle of waste, and overheating into a virtuous circle of energy efficiency.

Heat pumps are considered one of the most efficient heating and cooling systems and, according to Directive 28/2009/EC, the aerothermal, geothermal or hydrothermal energy captured by these appliances is considered energy from renewable sources. For these reasons, they will play a key role in reducing greenhouse gas emissions [1].

Aerothermal energy encompasses all the systems that allow energy to be extracted from the air. The most used technology for the use of aerothermal energy is the air-water heat pump that is developed in this work and that allows heating or cooling the water of a building, Figure 1.

Heat pumps consume up to 70% less energy than a traditional heating system. The consumption will depend on the type of heat pump we have, in the case of the aerothermal heat pump, the consumption would be 1 kW electric and would allow to deliver up to 4 kW of heating.
The International Energy Agency, in its 2016 report on Energy Efficiency, considered the heat pump as the Best Available Technology (BAT) for space heating. In addition, Greenpeace in 2011 chose the heat pump as the best heating system when it comes to energy efficiency.
The objective of this work is to take advantage of clean and renewable energy from the subsoil in Metro de Madrid's infrastructures. It is intended to extract the waste heat generated in the metro tunnels due to the traction and braking of machines, auxiliary facilities, or transit of people on the platforms and use it to produce domestic heat water (DHW) of a building in the Community of Madrid, thus assuming savings in kWh.

https://doi.org/10.52202/069564-0259

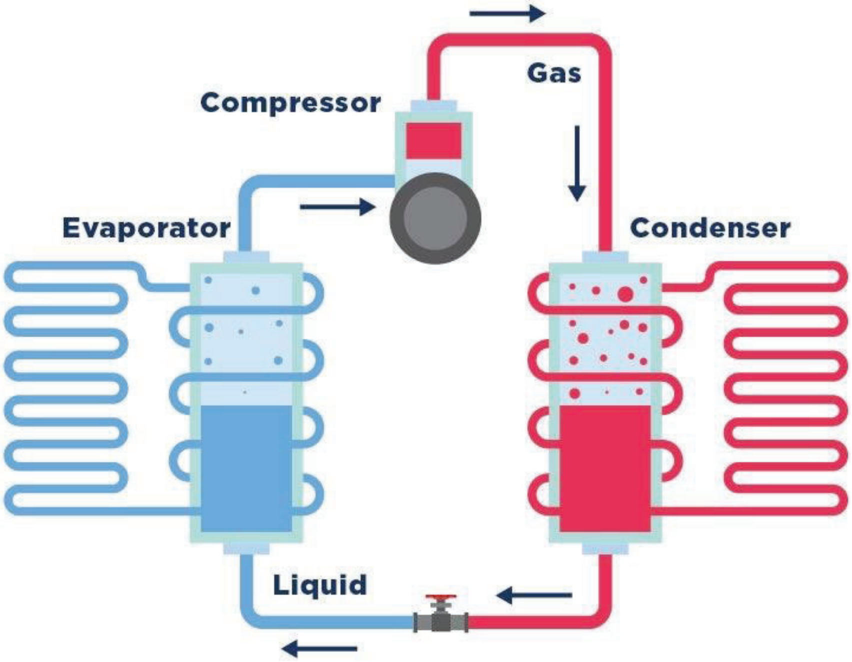


Figure 1. Heat Pump. Source https://www.bordgaisenergy.ie/home/heat-pump-guide

The study of the thermal use of air will be carried, out in the Seville metro station and in the Building of the Ministry of Environment of the Community of Madrid, Figure 2.

Figure 2. Location of Metro Sevilla and the building of the Ministry of Environment

https://doi.org/10.52202/069564-0259

Sevilla Metro station is a station on line 2 of the Madrid Metro located under Calle Alcala, at the junction with Calle de Sevilla, which gives its name to the station. It is a very small station of approximately 1000 m^2, with "light" trains of 4 cars, with platforms of 60 m and with low influx according to the data provided with Metro Madrid.

The objective of the work is, therefore, to have an estimate of the liters of domestic heat water that could be obtained from the flow of air extracted by means of the fans of the extraction well. In addition, the number of people inside the Building of the Ministry of Environment that could be supplied with that volume of water will be estimated.

2. Methodology

The methodology followed in this study consists of three steps:

- **Step 1:** The simulations provided by Metro de Madrid corresponding to the Seville station allow us to know and analyze the air temperatures and the ventilation flows of the tunnel and determine the design conditions.
- **Step 2:** The aerothermal heat pump is designed and sized. The result of this step is the fundamental parameters of the heat pump.
- **Step 3:** The installation performance is estimated using the CoolPack program [2] from the results obtained in the previous steps.

The centralized DHW installation object of this study has been designed according to the Technical Guide of Central Sanitary Hot Water of the IDEA [3]. This document establishes certain design criteria, especially highlighting the importance of control over Legionella. The hygienic-sanitary measures that must be adopted in those facilities in which Legionella is capable of proliferating and spreading are described on the Royal Decree 487/2022, of June 21 [4].

2.1. Analysis of the data provided by Metro Madrid.

For the calculation of the use of heat from the Seville Metro station, the simulations provided by Metro Madrid have been analyzed. In them, there is on the one hand the thermal evolution in a winter, half-time and summer day for four fundamental domains: station (platform level), vestibules and the two adjacent tunnel sections. On the other hand, there is the flow of air, which is massively directed to the shafts of the tunnels, and especially to the tunnel of Banco de España, (Tunnel 1) since it has forced ventilation. The place chosen to put the pump and carry out the study is Tunnel 1 due to its greater air extraction flow, Figure 3.

The following points have been considered for the study and diagnosis:

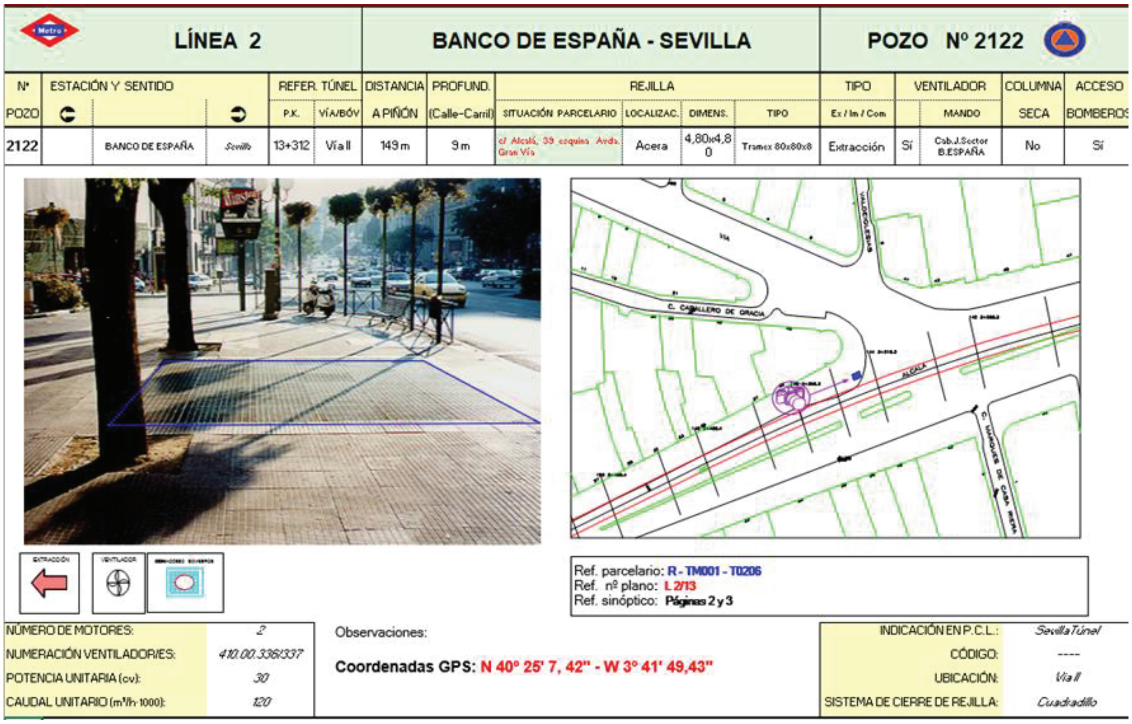
- It has been decided to work according to the key variables of the problem: the ventilation flow and temperatures. An average temperature will be taken for winter, halftime and summer, Table 1.
- Operating hours of the fans, in winter they would work 10 hours a day, in halftime 13 and in summer they would be on for 15 hours extracting the air flow, according to the data provided by Metro Madrid.

Table 1. Average data of the selected temperature Metro Madrid.

Parameters	Values
Average winter temperature	12.9 ^{o}C
Average halftime temperature	17.6 ^{o}C
Average summer temperature	20.9 ^{o}C

With these data, we proceed to define the design of the components of the heat pump that would be installed in Tunnel 1 and the selection of the parameters of the simple thermodynamic cycle with which it would work.

https://doi.org/10.52202/069564-0259



LÍNEA 2 **BANCO DE ESPAÑA - SEVILLA** **POZO Nº 2122**

Nº	ESTACIÓN Y SENTIDO			REFER. TÚNEL		DISTANCIA	PROFUND.	REJILLA				TIPO	VENTILADOR		COLUMNA	ACCESO
POZO	↶		↷	P.K.	VÍA/BÓV	A PIÑÓN	(Calle-Carril)	SITUACIÓN PARCELARIO	LOCALIZAC.	DIMENS.	TIPO	Ex / In / Com		MANDO	SECA	BOMBEROS
2122		BANCO DE ESPAÑA	Sevilla	13+312	Vía II	149 m	9 m	c/ Alcalá, 39 esquina Avda. Gran Vía	Acera	4,80x4,80	Tramex 80x80x8	Extracción	Sí	Cab.J.Sector B.ESPAÑA	No	Sí

Ref. parcelario: R - TM001 - T0206
Ref. nº plano: L 2/13
Ref. sinóptico: Páginas 2 y 3

NÚMERO DE MOTORES:	2
NUMERACIÓN VENTILADOR/ES:	410.00.336/337
POTENCIA UNITARIA (cv):	30
CAUDAL UNITARIO (m³/h·1000):	120

Observaciones:

Coordenadas GPS: N 40° 25' 7, 42" - W 3° 41' 49,43"

INDICACIÓN EN P.C.L.:	Sevilla Túnel
CÓDIGO:	----
UBICACIÓN:	Vía II
SISTEMA DE CIERRE DE REJILLA:	Cuadradillo

Figure 3. Photo of the place where the Heat Pump will be located.

2.2. Design and size of the aerothermal heat pump.

The proposed installation will comprise a simple heat pump system that consists of four primary components: a compressor, two heat exchangers (condenser and evaporator), and an expansion valve, as depicted in Figure 1. The initial calculation parameters selected for the analysis are specified in Table 2.

Table 2. Parameters for calculating.

Parameters	Data
Domestic hot water temperature	55ºC [3]
Condensation temperature	60ºC
Subcooling temperature	10K
Useful superheat temperature	7K
Exhaust air flow	30 m^3/s (Tunnel 1)
Seasonal temperature data	Table 1

2.2.1. Evaporator.

The evaporator serves as a heat exchanger, facilitating the phase change of the refrigerant from liquid to vapor while absorbing heat. In order to determine the power associated with the evaporator, Eq. (1) is employed, considering the parameters outlined in Table 3 in order to evaluate the energy balance between evaporator refrigerant and air to subtract heat.

$$Q(kW) = \dot{v}_{air}\left(\frac{m^3}{s}\right)\cdot\ \rho\left(\frac{kg}{m^3}\right)\cdot C_p\left(\frac{kJ}{kgK}\right)\cdot\Delta T = \dot{m}_{\text{refrigerant}}\cdot\Delta\text{h} \tag{1}$$

Table 3. Parameters for calculating the power of the evaporator.

Parameters	Data
Air flow ($\dot{v}_{air}$)	30 m^3/s
Air density (ρ)	1.2 kg/m^3
Specific heat of air (C_p)	1.007 kJ/kg K
Temperature difference at evaporator inlet and outlet (ΔT)	Variable in the study

Upon analysis of the evaporator power results presented in Table 4, it was observed that the extracted power remained independent of variations in the air inlet temperature. It should be noted that the calculation of efficiency is contingent upon the time of year for which it is being computed.

Table 4. Evaporator powers (Q_{evap}), according to thermal jump.

ΔT (K)	Q_{evap}(kW)
2	72.5
4	145.0
6	217.5
8	**290.0**

In order to determine the efficiency of the evaporator, the NTU (Number of Transfer Units) Effectiveness Method [5] is employed, Eq. (2). Number of Transfer Units (NTU) is a dimensionless parameter used in heat exchanger analysis to determine the efficiency of heat transfer. It is defined as the product of the overall heat transfer coefficient and the effective heat exchanger length, divided by the heat capacity rate.

$$\epsilon = 1 - e^{NTU} \tag{2}$$

The analysis entails experimentation with varied evaporator temperatures (T_{evap}) during three seasonal periods, namely winter, mid-season, and summer. Ultimately, the case with the highest evaporator power Table 4 is chosen. The results, including T_{ms} (the temperature at the outlet of the evaporator), LMTD (logarithmic mean temperature difference), and UA (heat transfer coefficient per unit area) are documented in Table 5 and referred in [5] and [6].

Table 5. Data selected for the highest evaporator power in each seasonal period.

Period seasonal	T_{evap} (ºC)	ΔT (K)	T_{ms} (ºC)	Q_{evap} (kW)	LMTD	UA (kW/K)	NTU	**Efficiency**
Winter	4	8	4.9	290.0	3.4	83.1	2.3	89.9
Mid-season	4	8	9.6	290.0	10.2	28.4	0.8	54.3
Summer	4	8	12.9	290.0	12.4	23.3	0.6	47.4

2.2.2. Condenser.

The next point is to set the condensation temperature. As previously defined, the purpose of the work is to be able to supply DHW, therefore the temperature is 60 ºC which will be the same as the condenser temperature ($T_{cond.}$), the domestic heat water temperatures T_s = 55 ºC [3], the cold-water temperature (T_{water}), whose average values are detailed in Table 6 and the Eq. (3) and Eq. (4), we are defined the analysis of the condenser at each time of the year Table 7.

Table 6. Temperature of the cold-water network of Madrid.

Seasonal period	Cold water temperature of the Madrid network (ºC)
Winter	7
Halftime	13
Summer	10.8

https://doi.org/10.52202/069564-0259

$$Qcond = UA \cdot LMTD \tag{3}$$

$$LMTD = \frac{T_S - T_{water}}{ln\left(\frac{T_{cond} - T_{water}}{T_{cond} - T_S}\right)} \tag{4}$$

Table 7. Analysis of the condenser according to the seasonal period.

Seasonal period	T_s (° C)	T_{water} (° C)	T_{cond} (° C)	LMTD	UA (kW/K)	Q_{cond} (kW)
Winter	55	7	60	20.1	83.1	1667.2
Halftime	55	13	60	18.7	23.2	435.7
Summer	55	10.8	60	19.3	28.4	548.0

2.2.3. Compressor.

Its function is to increase the pressure (and temperature) of the refrigerant. The refrigerant must be entirely in a gaseous state, if there was refrigerant in a liquid state, it would lead to serious damage to the compressor. Gas compression losses and charge losses in refrigerant circulation shall be considered. Will be used for the calculation of the thermodynamic cycle performance the CoolPack Software [2]. This program uses by default an isentropic yield of 0.7 and a loss factor of 10%.

2.2.4. Expansion Valve.

The coolant passes through expansion valve, decreasing its pressure and increasing its volume abruptly at the outlet. Some of the liquid coolant evaporates with the pressure reduction. The amount of refrigerant gas produced shall be kept to a minimum to increase the performance of the evaporator.

2.3. Study of the system efficiency.

Determination of fundamental parameters of this type of systems to realistically study how efficient the aerothermal system is:

- ✓ **COP** (electricity consumption required to meet heat demand)
- ✓ **sCOP** (Seasonal Coefficient of Performance).

2.3.1. Determination of the COPs

This parameter is essential when analyzing the operation of a heat pump. To improve the COP of the pumps, it is important that the condensation temperature is as low as possible, and that the evaporation temperature is as high as possible.

The **operating or performance coefficient (COP)** is an expression of the efficiency of a heat pump. In this case, it is a relationship between the heat transferred and the electrical energy consumed mainly by the compressor, Eq. (5).

$$COP = \frac{Q}{W_{comp}} \tag{5}$$

The study of the electricity consumption required to meet the heat demand is carried out with the CoolPack program that allows you to enter the parameters with which you are going to work:

- evaporation temperature, condensation temperature, subcooling, useful reheating, compressor performance, air flow and type of refrigerant. The values of these parameters are reflected in the Table 8.

https://doi.org/10.52202/069564-0259

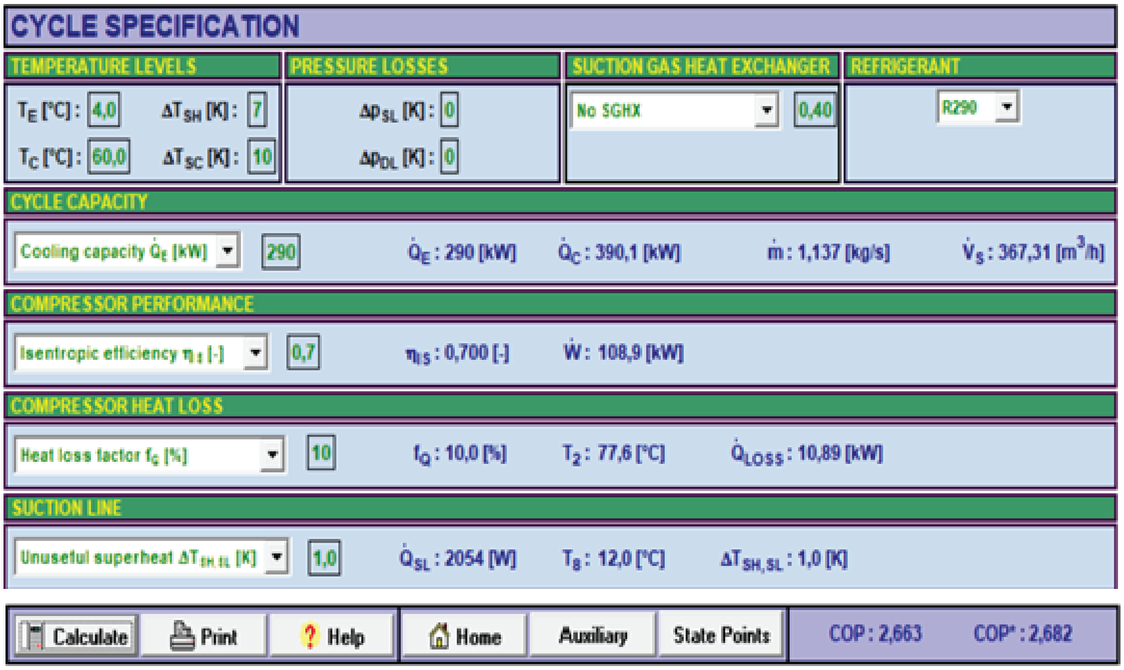
Table 8. Heat pump cycle parameters.

Parameters	Values
Evaporation temperature	4 ºC
Condensation temperature	60 ºC
Compressor efficiency	0.7
Evaporation capacity	290.016 kW
Subcooling	10 K
Useful reheating	7 K
Useless overheating	1 K
Refrigerant (the program defaults)	R290

This analysis is fundamental since this value supposes a first filter for the viability of the supposed heat pump. For proper efficiency and functionality, a heat pump must reach a COP of between 2 and 6, depending on the difference between the temperatures of both sources (indoor or outdoor). In this work, a value of **COP= 2.663** is obtained, Figure 4.

In addition, with the COP you can make a comparison between the price of fuel (natural heat source) and the electricity price (cost of operation with heat pump) that we see below:

- COP < 2.5 the variable thermal cost with boilers is lower than with heat pump.
- COP=2.5 the thermal variable cost is identical in both cases.
- COP > 2.5 the variable thermal cost with boilers is higher than with heat pumps.



T_E [°C] : 4,0
ΔT_{SH} [K] : 7
T_C [°C] : 60,0
ΔT_{SC} [K] : 10
Δp_{SL} [K] : 0
Δp_{DL} [K] : 0
No SGHX 0,40
R290
Cooling capacity $\dot{Q}_E$ [kW] 290
$\dot{Q}_E$: 290 [kW]
$\dot{Q}_C$: 390,1 [kW]
$\dot{m}$: 1,137 [kg/s]
$\dot{V}_S$: 367,31 [m^3/h]
Isentropic efficiency η_{is} [-] 0,7
η_{IS} : 0,700 [-]
$\dot{W}$: 108,9 [kW]
Heat loss factor f_Q [%] 10
f_Q : 10,0 [%]
T_2 : 77,6 [°C]
$\dot{Q}_{LOSS}$: 10,89 [kW]
Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0
$\dot{Q}_{SL}$: 2054 [W]
T_8 : 12,0 [°C]
$\Delta T_{SH,SL}$: 1,0 [K]
COP : 2,663
COP* : 2,682

Figure 4. Obtaining COP with the CoolPack program.

There is a very important point in heat pumps, and it is the issue of the refrigerant used. In the previous analysis, the R290 is used, which defaults to the CoolPack program. To get an idea of how important the refrigerant is, is represented in the following, how the COP change depending on the refrigerant used, Figure 5.

https://doi.org/10.52202/069564-0259

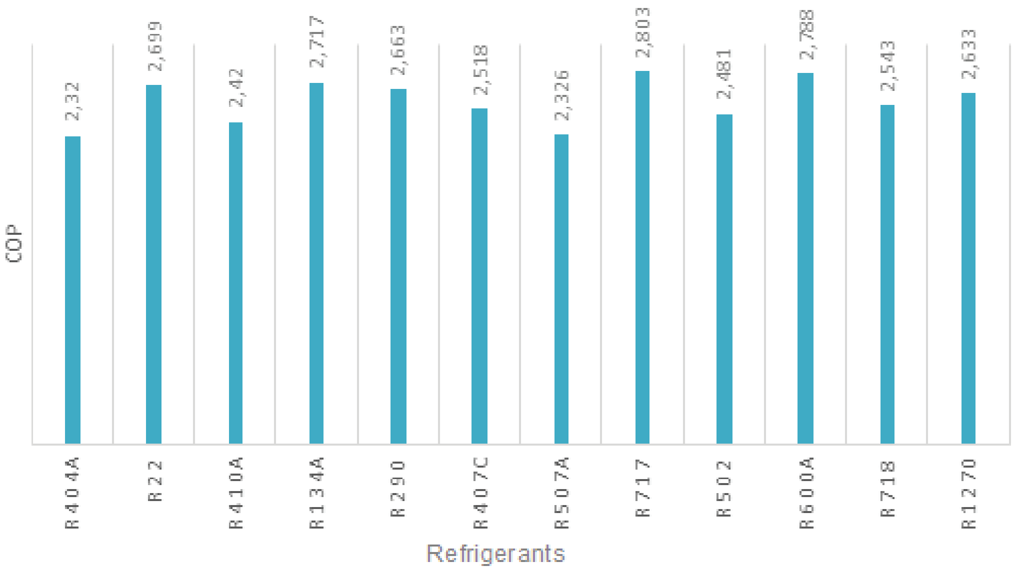


Figure 5. COP results with different refrigerants.

Based on this analysis we could conclude that the best refrigerant would be R717, but differences are quite small with R600A, R134a, R22, R1270, model uncertainties can group these refrigerants with the same possibilities to be the better one. However, other factors must be considered when choosing the refrigerant, such as safety, environment, corrosion resistance and at the same time good thermodynamic properties [7].

2.3.2. Determination of sCOP

After obtaining the potential COP values for a heat pump, an additional parameter that holds greater significance is introduced when evaluating the installation's performance. This parameter is known as the Seasonal Coefficient of Performance (sCOP). The main difference between the COP and the sCOP is that the latter refers to the seasonal term, i.e. a certain period of time, while the former refers to a specific operation conditions.

For the estimation of sCOP values, the procedure explained in the IDEA [8] document "Average seasonal performance of heat pumps" is followed.

The average seasonal performance of an equipment or system (SPF) shall be calculated by multiplying its rated performance (COP) by a factor called the representative weighting factor (FP), Table 9, and by a correction factor (FC), Table 10, for the different technologies and applications of electrically driven heat pumps. The weighting factor considers the different climatic zones of Spain marked by the CTE [9] and has been calculated using an exclusively technical methodology, using objective values and existing recognized documents. The correction factor considers the difference between the distribution or use temperature and the temperature for which the COP have been obtained in the test.

Table 9. Weighting factor (FP) [8]

Heat pump energy source	A	B	C	D	E
Aerothermal energy. Centralized teams	0.87	0.80	0.80	0.75	0.75
Aerothermal energy. Individual teams type Split.	0.66	0.68	0.68	0.64	0.64
Hydrothermal Energy	0.99	0.96	0.92	0.86	0.80
Closed-loop geothermal energy. Horizontal heat exchangers	1.05	1.01	0.97	0.90	0.85
Closed-loop geothermal energy. Vertical heat exchangers	1.24	1.23	1.18	1.11	1.03
Open Circuit Geothermal Energy	1.31	1.30	1.23	1.17	1.09

https://doi.org/10.52202/069564-0259

Madrid is a zone of climatic severity D as can be seen in Figure 6. The type of heat pump used throughout the document is aerothermal, therefore, the weighting factor (FP) that corresponds to it is FP **= 0.75.**

Figure 6. Climate zones in Spain

For the choice of the correction factor (FC) the one that agrees with the COP at 60ºC is chosen, this being the ideal condensation temperature to supply domestic hot water to the building of the Ministry of the Environment. According to Table 10 we are left with the correction factor being **FC=1.**

Table 10. FC correction factor

Condensation temperature (ºC)	FC (COP a 35 ºC)	FC (COP a 40 ºC)	FC (COP a 45 ºC)	FC (COP a 50 ºC)	FC (COP a 55 ºC)	FC (COP a 60 ºC)
35	1.00	------	------	-------	------	------
40	0.87	1.00	------	------	------	------
45	0.77	0.89	1.00	------	------	------
50	0.68	0.78	0.88	1.00	------	------
55	0.61	0.70	0.79	0.90	1.00	------
60	0.55	0.63	0.71	0.81	0.90	1.00

With the data explained above and the Eq. (6) a **sCOP = 1.975** is obtained**.**

$$\boldsymbol{sCOP} = COP\ nominal \cdot FP \cdot FC \tag{6}$$

The use of standard methodology to evaluate sCOP is clear, and it is quite useful. In this case, common methodology is not correct at all, because temperature in tunnels suffer different variations that in the environment air, so these coefficients are underestimating the real value of sCOP considering the comparison between air tunnels temperature variation and environmental air temperature variation. Further analysis with implemented temporal series will include this calculation of sCOP.

3. Amount of domestic heat water produced

With the data provided by Metro Madrid and the results calculated to compose the heat pump, the hot water flow is determined with the following Eq. (7),

$$Qcond = \dot{m}_{water}\left(\frac{kg}{s}\right) \cdot C_p\left(\frac{kJ}{kg*K}\right) \cdot (T_s - T_e) \tag{7}$$

https://doi.org/10.52202/069564-0259

The power of the condenser (Q_{cond}) estimated to calculate the production of domestic hot water is considered since the heating of the water occurs in this component. The formula uses a C_{pwater} = 4.196 kJ / kg K, the density of water ρ_{water} = 999.91 kg / m^3, the outlet temperature of domestic heat water T_s = 60ºC and the inlet temperature that coincides with that of the cold water of the Madrid network that will depend on the time of year collected in Table 6. Thus, the flow of domestic heat water produced during the months of the year would be represented in the following graph, Figure 7.

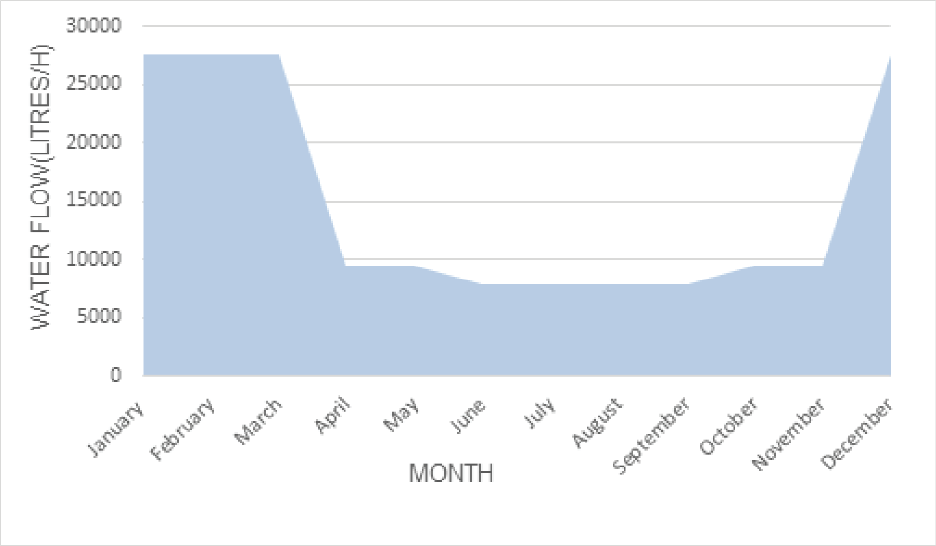


Figure 7. DHW flow during the year per operating hour

Upon analysis of the current air extraction rate from Tunnel 1 (108000 m^3/h), it was determined that an average flow rate of 15007 liters/hour of fan operation can be produced. It should be noted that higher flows will be obtained during the winter season due to the greater power in the condenser resulting from much lower average air intake temperatures. Additionally, the hours of fan operation coincide with those of the heat pump to be installed in Tunnel 1. As such, the monthly amount of domestic hot water that can be obtained is also presented in Figure 8.

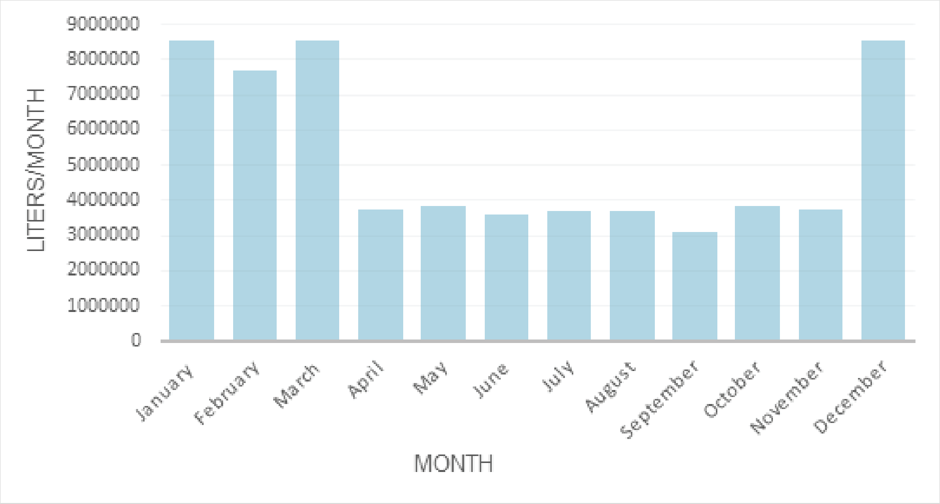


Figure 8. Flow rate in liters of DHW per month

Finally, if it is considered that in a public building such as the Building of the Ministry of Environment a person can consume 22.5 liters per day [9], an estimate can be made of the number of people that could cover the demand for DHW per month, Table 11.

Table 11. People who can be supplied with DHW from the extracted air flow.

Month	Q_{cond} (KW)	T_e (°C)	$\dot{v}_{water}$(m^3/s)	$\dot{v}_{water}$(l/h)	Hours of operation	l/day	People
January	1667	8	0.00764	27510	310	275096	12227
February	1667	8	0.00764	27510	280	275096	12227
March	1667	8	0.00764	27510	310	275096	12227
April	548	11	0.00265	9557	390	124243	5522
May	548	11	0.00265	9557	403	124243	5522
June	436	13	0.00221	7955	450	119325	5303
July	436	13	0.00221	7955	465	119325	5303
August	436	13	0.00221	7955	465	119325	5303
September	436	13	0.00221	7955	390	103415	4596
October	548	11	0.00265	9557	403	128384	5706
November	548	11	0.00265	9557	390	124243	5522
December	1667	8	0.00764	27510	310	275096	12227

4. Conclusions and future works

Once the methodology for the study of the use of waste heat from the Bank of Spain Tunnel of the Seville Metro station has been established, it can be concluded that the

1. Heat pump would produce an average flow of 15007 liters / hour during the hours in which they are operating throughout the day. In addition, the extraction of the air flow from Tunnel 1 could supply an average of 7640 people per month.
2. With respect to the heat pump placed at the outlet of the air extraction well, it is obtained that using the refrigerant R290, an evaporation temperature of 4ºC and a condensation temperature according to that required for DHW (60ºC) a COP = 2.663 is reached, a positive value (> 2.5) when measuring the viability of the system for correct efficiency and functionality that would mean savings in euros compared to gas, but which remains below the minimum COP necessary (6.08 for DHW at 60ºC) to be considered renewable.
3. The results of the Seasonal Coefficient of Performance are less encouraging since a value of 1.975 will be obtained, which is below the minimum 2.5 to be considered renewable according to the IDAE report. Anyway, IDAE sCOP methodology calculation procedure underestimate real value, as working conditions for Metro are better than the ones expected for the common use of this systems.[8]
4. This concept helps to reduce the temperature in Metro facilities, that is a problem for the infrastructure as an all-year net heat source.

However, the main essence of this project is the dissemination and awareness of the possibility of installing aerothermal systems capable of taking advantage of the residual heat of the subsoil of Madrid to achieve an energy use since a virtuous circle of heat recovery that would otherwise be wasted in the environment will be implemented regardless of whether it is renewable.

The methodology developed will allow, using more precise data and details on the installation of the Building of the Ministry of the Environment, the dimensioning of the installation necessary to take advantage of the heat of the metro or numerous other equally unknown infrastructures with enormous potential to exploit.
In conclusion, the results show that heat pumps are presented as indispensable allies of the energy efficiency of buildings in cities as it is the only proven and available system capable of producing great savings and reducing CO_2 emissions in indoor air conditioning.

Future works include the more precision evaluation of the annual production with temporal series of temperature and comparison with the alternative of a common aerothermal facility working with environmental air, as in summer the overground higher temperature gives better performance to the production of hot water.

https://doi.org/10.52202/069564-0259

Acknowledgments

Authors thanks to A*ula-Empresa Madrid Subterra* – Escuela Técnica Superior de Ingenieros Industriales de Madrid for promote and support this work.

References

[1] Madonna F., Bazzocchi F., Annual performances of reversible air-to-water heat pumps in small residential buildings, Energy and Buildings 2013; 65:299-309, https://doi.org/10.1016/j.enbuild.2013.06.016. (https://www.sciencedirect.com/science/article/pii/S037877881300354X)

[2] CoolPack, https://www.ipu.dk/products/coolpack/, IPU, Technology Driven Business Innovation, [accessed 15.3.2023]

[3] Documents_08_Guide_technica_hot_water_sanitary_central_906c75b2.pdf. Available in: https://www.idae.es/uploads/documentos/documentos_08_Guia_tecnica_agua_caliente_sanitaria_central_906c75b2.pdf. [accessed 15.3.2023]

[4] Royal Decree 487/2022, of June 21, which establishes the health requirements for the prevention and control of legionellosis. BOE-A-2022-10297 Real Decreto 487/2022, de 21 de junio, por el que se establecen los requisitos sanitarios para la prevención y el control de la legionelosis. [accessed 15.3.2023]

[5] Tadeo-Cañete M., Energy use of the Seville metro station for thermal supply of the building of the Ministry of Environment of the Community of Madrid, September 2022, ETSII-UPM, final degree project.

[6] Bergman, T. L., and Frank P. Incropera. Fundamentals of Heat and Mass Transfer. Seventh edition. Hoboken, NJ: Wiley, 2011.

[7] Royal Decree 552/2019, of 27 September, approving the Safety Regulation for refrigeration installations and their complementary technical instructions.

[8] Average seasonal performance of heat pumps for heat production in buildings (energia.gob.es)

[9] Technical Building Code, "Royal Decree 314/2006, of March 17, which approves the Technical Building Code.

https://doi.org/10.52202/069564-0259