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ANALYZING THE EFFECTIVENESS OF NIBE F1155 GEOTHERMAL HEAT PUMP BY USING COOLPACK SOFTWARE SIMULATION IN HEATING MODE.

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Abstract

Geothermal energy is one of the most promising renewable energy sources and its importance is increasing worldwide. It has already proven to be reliable, safe and clean. Therefore, its use for power generation, heating and cooling is among many developed countries, as it is an energy source that generates electricity with minimal impact on the environment.

This project aims to describe and contextualize the use of a machine as a geothermal closed loop heat pump, and the reasons why it is preferable in certain cases, in particular as regards a more common boiler heating system. The results of the study will be treated more technically.

Keywords: geothermal energy, heat pump, NIBE, Coolpack, COP, closed circuit.

ISITISH REJIMIDA COOLPACK DASTURIDA SIMULASYA QILISH ORQALI NIBE F1155 GEOTERMAL ISSIQLIK NOSOSI SAMARADORLIGINI TAHLIL QILISH.

Madumar Musurmonov
Toshkent shahridagi Yodju Texnika Instituti

Annotatsiya

Geotermal energiya qayta tiklanadigan energiya manbalari orasida eng istiqbollilaridan biri bo'lib, uning ahamiyati butun dunyo bo'ylab ortib bormoqda va u allaqachon ishonchli, xavfsiz va toza ekanligi isbotlangan va shuning uchun uni

energiya ishlab chiqarish, isitish va sovutish uchun ishlatish ko'plab rivojlangan mamlakatlarda ommalashmoqda, chunki u atrof-muhitga minimal ta'sir ko'rsatadigan energiya manbai hisoblanadi.

Ushbu loyiha yopiq sikllі geotermal issiqlik nasosidan foydalanishni tavsiflash va kontekstuallashtirishga qaratilgan va ba'zi hollarda, ayniqsa, an`anaviy isitish tizimiga nisbatan afzalroq bo'lishini isbotlash maqsad qilingan. Tadqiqotdan olingan natijalar ko'proq texnik jihatdan ko'rib chiqiladi.

Kalit so'zlar: geotermal energiya, issiqlik nasosi, NIBE, Coolpack, COP, yopiq sikl.

АНАЛИЗ ЭФФЕКТИВНОСТИ ГЕОТЕРМАЛЬНОГО ТЕПЛОВОГО НАСОСА NIBE F1155 С ИСПОЛЬЗОВАНИЕМ ПРОГРАММНОГО МОДЕЛИРОВАНИЯ COOLPACK В РЕЖИМЕ ОТОПЛЕНИЯ.

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Аннотация

Геотермальная энергия является одним из наиболее перспективных среди возобновляемых источников энергии, и ее значение растет во всем мире, и она уже доказала свою надежность, безопасность и чистоту, и поэтому ее использование для производства электроэнергии, отопления и охлаждения является одним из многих развитых стран, поскольку она источник питания, производящий электроэнергию с минимальным воздействием на окружающую среду.

Этот проект направлен на описание и контекстуализацию использования такой машины, как геотермальный тепловой насос с замкнутым контуром, а также причин, по которым в некоторых случаях это предпочтительнее, особенно в отношении более распространенной системы отопления с бойлером. Результаты, полученные в результате исследования, будут рассмотрены более технически.

Ключевые слова: геотермальная энергия, тепловой насос, NIBE, Coolpack, COP, замкнутый контур.

Introduction

Initially, the objectives of the analysis carried out were to quantify the yields and amounts of energy involved during the operation of the Nibe F1155 6kW heat pump installed in the Energy Center. Data obtained from sensors inside the machine itself, which operated in heating or cooling mode, were used, with which the COPs were then calculated using a spreadsheet. In parallel, we will calculate the results using the Coolpack software and finally we will be able to make a decision about the operating efficiency of the system by comparing the experimental and simulated results.

Geothermal energy

Geothermal energy, one of the most promising renewable energy sources, has proven to be reliable, safe, and clean and therefore its use for power generation, cooling and heating is growing rapidly. Geothermal energy is an energy source that produces electricity with minimal impact on the environment (Fridleifsson, 2001; Barbier, 1997). Geothermal energy, all over the world, is used in various ways, such as space heating and domestic hot water production, CO2 and dry ice production, greenhouse heating, hydrotherapy, water pumps. Terrestrial production of heat and electricity.

“Geothermal energy is the energy contained within the high-temperature mass of the Earth’s crust, mantle, and core. Since the Earth’s interior is much hotter than its surface, energy flows continuously from the deep, hot interior up to the surface. This is the so-called terrestrial heat flow. The temperature of the Earth’s crust increases with depth following Fourier’s law of heat conduction. Thus the energy content of a unit of mass also increases with depth.”

Geothermal energy is produced by capturing thermal energy in the deepest layers of the earth's crust. It is obtained by channeling steam from the subsoil to the turbines used to generate electricity and by recycling the water vapor produced to heat buildings, greenhouses and various uses in heating systems.

The following figure shows the evolution of ground temperature during the winter and summer months as a function of depth.

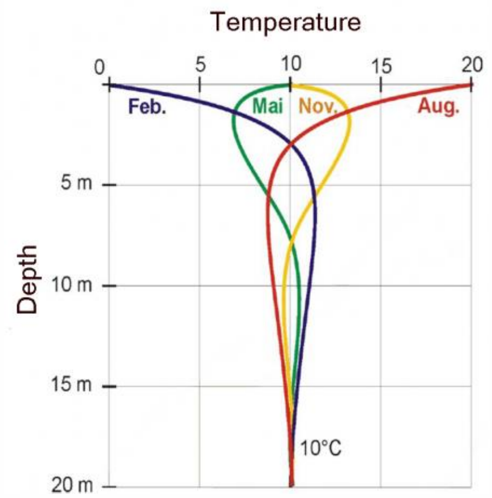

Figure 1 Diagram of the subsoil Temperature according to the depth of soil layer (Rivas, 2020)

Nibe F1155 (6kW, 1x230V)

In this project, we will analyze the operating principles and characteristics of NIBE F1155 (Figure 2). Because the energy center is equipped with a NIBE F1155 geothermal pump that was installed on 07/19/2019 (GEONOVIS energia geotermica, 2019). This pump is an intelligent, inverter-controlled geothermal heat pump with no integrated hot water tank, making it easy to install in locations with low ceilings. A separate hot water tank is selected based on hot water requirements. NIBE F1155 provides optimal savings as the heat pump automatically adapts to the heating demand of the building.

Figure 2: NIBE heat pump of the center with a buffer tank and fan coil

This pump was installed after the construction of the energy center was completed. This is why the engineers encountered problems installing the geothermal pipes of the system, and it was so expensive to build and cost almost 30,000 euros. This pump is a small system and for experimental purposes only.

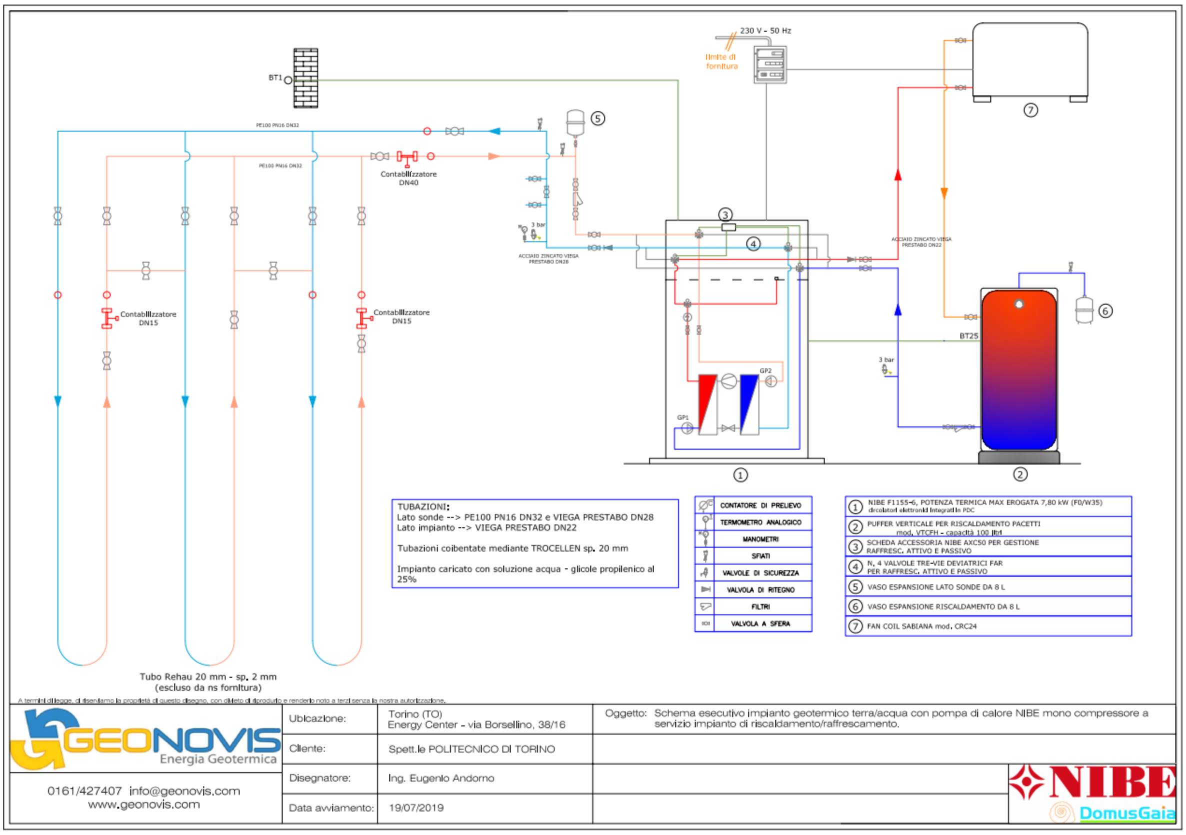

Figure 3 Plant project of the pump installed in the center (GEONOVIS energia geotermica, 2019)

Operating principle

The F1155 system contains a heat pump, an immersion heater, circulation pumps, and a control system. The pump is connected to the brine system and heat transfer fluid systems. And the whole system consists of 3 circuits: brine circuit, refrigerant circuit and heat transfer fluid circuit (figure 4).

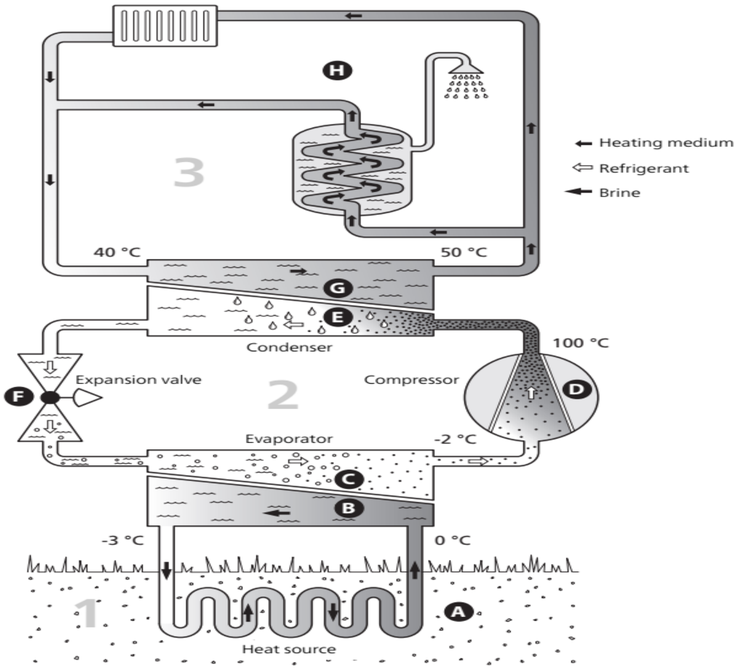

Figure 4: Scheme of the heat pump (The temperatures are only examples and may vary between different installations and times of the year.)

1) Brine circuit

A. This loop provides heat from the ground heat source (ground, rock pool) where the solar energy is stored. The heat transfer medium is brine, which is water mixed with glycol, antifreeze, or ethanol. In our case, this mixture is a 25% propylene glycol water solution that circulates in very flexible plastic pipes and draws its energy from the ground. The probes are certified and have a diameter of 20 mm and a thickness of 2.0 mm. In the centre, the wall extends for 7.50 m and reaches a depth of 4.6 m. Three different pipeline circuits of 2.5 m each have been installed, two horizontal and one vertical. (figure 5)

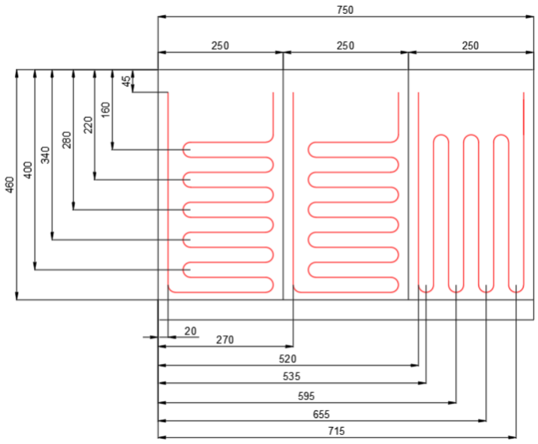

Figure 5: Geothermic arrnagement of coils with dimensions

The coils, thanks to a system of valves, have the possibility of being controlled selectively in series or in parallel by the fluid. The flow is managed by a pump (GP2). Energy from the heat source is stored by heating the brine a few degrees, from about –3°C to about 0°C.

B. The heated brine flows into the heat pump evaporator and releases thermal energy into the refrigerant and cools a few degrees, then returns to the heat source to recover heat from the ground (Figure 4).

2) Refrigerant circuit

C. An intermediate circuit is inside the pump and here the very low boiling point refrigerant R407C absorbs energy from the brine and begins to boil to a saturated vapor phase in the evaporator.

D. The refrigerant turns into gas and is sent to an electric compressor. When the compressor compresses the gas, the pressure and temperature rise sharply from about 5°C to 100°C

E. The compressed gas at high temperature and pressure passes to the condenser where it releases its energy to the end user via a heat exchanger, after which the refrigerant cools and condenses back into the liquid phase.

F. The fluid becomes liquid but it is still under high pressure, so here we use a pressure reducer to decrease the pressure of the fluid and bring it back to its point of origin. And so the cycle continues.

3) Heat medium circuit

G. In this circuit, the hot water inside the Pacetti mod. The 100L VTCFH with thermal flywheel function recovers energy from the refrigerant in a condenser heat exchanger and distributes it to the end-user. Here the water flow is managed by a pump (GP1)

H. The hot water heats the environment through heat exchangers, in this case it is the Sabiana mod CRC24 fan coil (Figure 2). This branch of the circuit can also be used for other purposes, such as conducting hot water to the building's sanitary installations, heating a swimming pool, etc. (NIBE, Instructions for use)

The system is also equipped with a whole series of valves, which allow its control and the different regulatory safety functions for your protection and that of the people who may be in your environment. (Round, 2020)

Sensors

The sensors that characterize the Nibe geothermal plant mainly compost from temperature sensors. Those useful for the purposes of this study will be listed below, which are graphically visible in figure 6a:

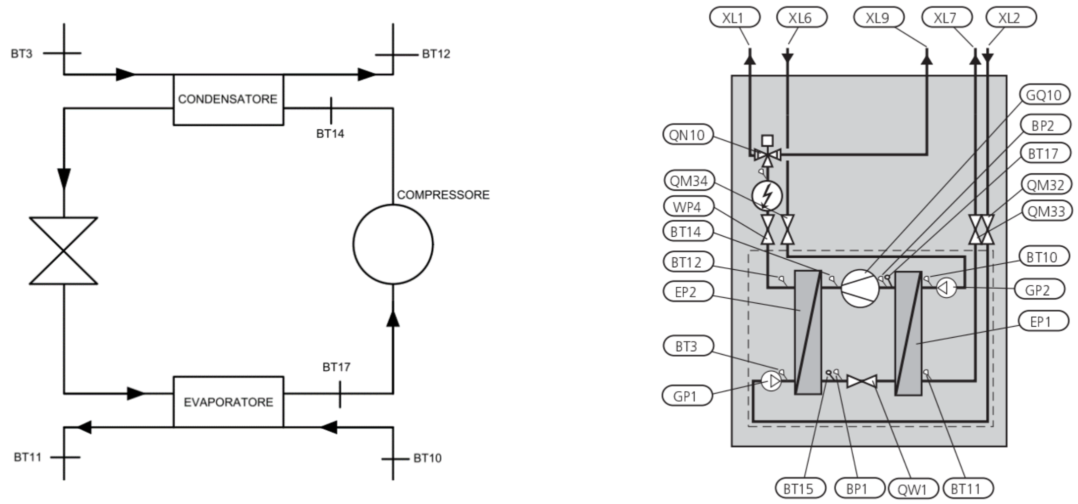

Figure 6 (a)Schematic drawing of the arrangement of the temperature sensors, and (b)other components of the pump

- outside temperature, BT1;
- return temperature of the heating fluid, BT3;
- the temperature of the return fluid from the probes, BT10;
- the temperature of the flowing fluid towards the probes, BT11;
- flow temperature from the condenser of the heating fluid, BT12;

- hot gas temperature, BT14;
- suction gas temperature, BT17.(figure 21a)

Case study

The main objective of this experiment is a comparison between the values calculated based on the data of the ".txt" file automatically generated by the Nibe F1155 electronic control unit and the results obtained from the "Coolpack" software. inserting the same data. And we will also analyze and compare the obtained results with the documentation values given in the technical data of NIBE F1155. By comparing the results, we can ensure that there are operational problems that prevent the use of this type of heat pump in a residential environment. All the energy values, the powers exchanged, the flows, the yields, any type of result, were calculated in instantaneous magnitudes for each of the approximately 4,500 measurements with the compressor on (the total measurements are around 20,750). These amounts were then averaged according to certain criteria, which vary from case to case, depending on what needs to be proven in the given circumstance.

The data used for the study comes almost entirely from the ".txt" file automatically generated by the electronic control unit of the Nibe F1155. The control unit performs a measurement at an adjustable regular frequency (in this case every 30 seconds) in which it stores the values measured by the sensors present inside the machine. We collect data from the switchboard for 8 days (from 10/24/2019 to 10/31/2019). We analyze the heat pump in heating mode (winter season), so we use old data that was measured in 2019. The useful values for the study are the operating speeds of the GP1 and GP2, presented in percentage and the temperatures. described above. In the current state of the machine, there are no sensors that allow instantaneous pressure measurement (an accessory must be purchased), but the manufacturer has specified that the high-pressure branch must operate with a pressure between 16 bar and 30 bar, while low pressure in the range of 7-11 bar. In the thesis, we calculate the values for four different pressure ranges to understand the effect of pressure on the performance of heat pumps. These pressure ranges are 7-16 bar, 8-16 bar, 7-20 bar and 7.23 bar, which fully corresponds to the permitted working pressure range of NIBE heat pumps. The last fundamental data is the one resulting from the measurement of the flowmeters present in the branch of the primary circuit, that is, the one that leads to the underground probes. The flow rate value of 460=l/h was used, (Tofalo, Sperimentazione di muri energyi, 2019) an average of the values measured by the flow meters installed in the system. By exporting the big data of the control unit to an Excel file, we can use all the temperature values of the system to calculate the necessary values (COP, mass flow, power exchanged between the circuits, etc.). In the data file, all temperatures are given in the format T(℃)*10 in order to clearly see the variations, since the system collects data from the sensors every 30 seconds.

To display a duty cycle, the period starting at 07:41:08 on 10/27/2019 with the compressor on, goes to 08:34:03, when the compressor is off, is taken as the sample ends at 12:02:34 again on 10/27/2019, a moment before the compressor restarted. The following figures show the temperature trends during this period of operation. As expected, the temperatures are different when the compressor starts, when the compressor stops, the upstream and downstream temperatures become identical, in the case of the exchangers (figure 7), while they decrease and become closer in the case of exchangers. compressor (figure 8).

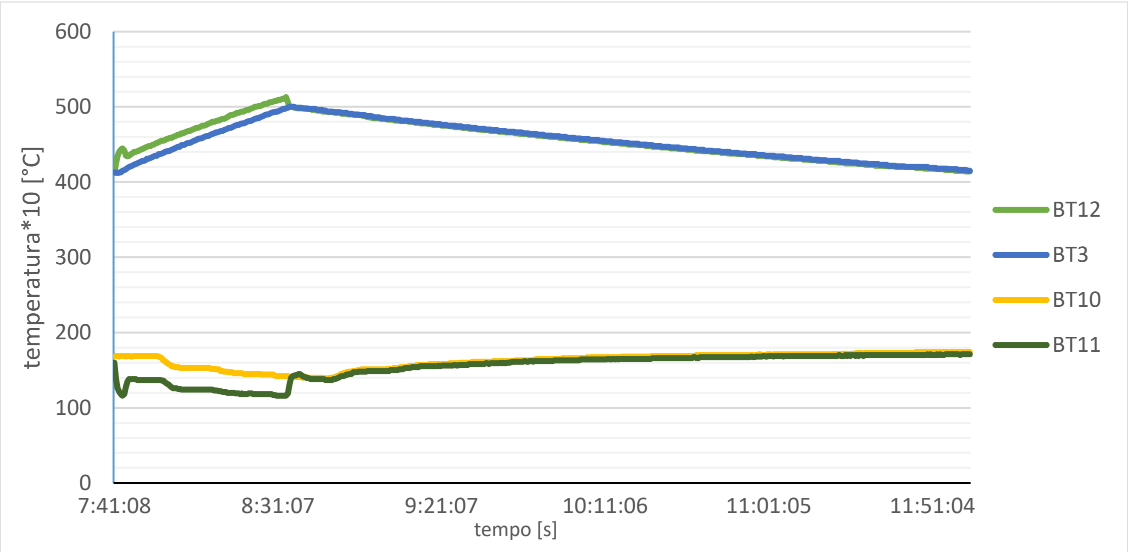

Figure 7: temperature trends of two heat exchangers.

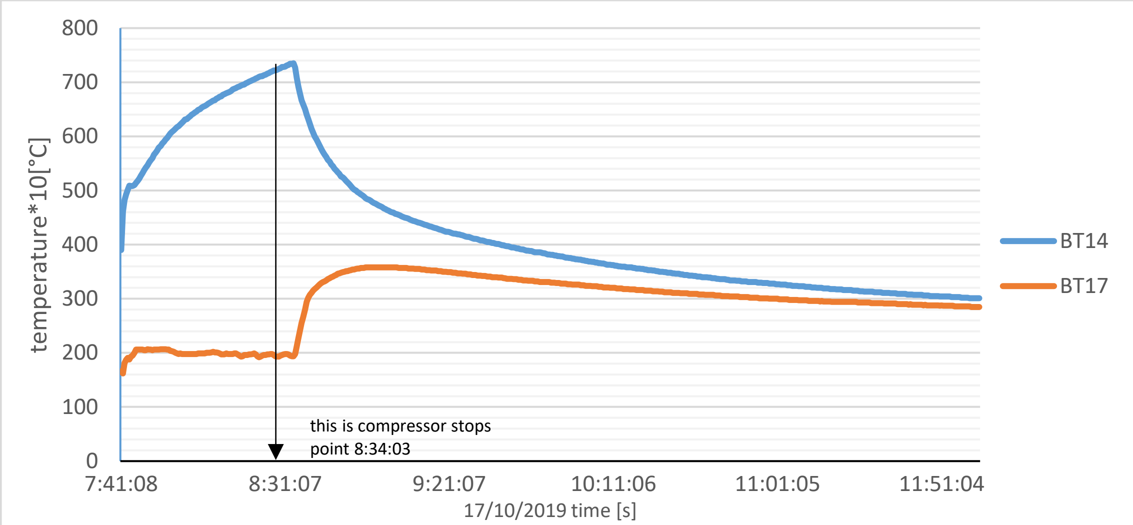

Figure 8: The temperature trends of compressor inlet (BT17) and outlet (BT14)

COP of the system

The system coefficient of performance (COP) is a typical energy performance used when our heat pump is operating in heating mode and is the ratio of the useful heat provided by the system to the work done by the compressor of the heat pump.

$$COP = \frac{\dot{Q}_H}{W_i} = \frac{q_{H,R407C}}{l_{i,R407C}} \ [-]$$

$q_{H,R407C}$: specific heat exchanged in the condenser (hot side) $\left[\frac{kJ}{kg}\right]$

$l_{i,R407C}$: is the work done per unit mass by the compressor and this is supply energy by external (electric energy) $\left[\frac{kJ}{kg}\right]$.

$\dot{Q}_H$: Heat exchanged in the condenser [kW]

W_i: Electric power used in the compressor [kW]

To calculate the COP of the heat pump, we have to use specific amounts of energy because the mass flow rate of the refrigerant is not known. Thermal losses can be neglected by approaching the compressor system as an adiabatic system, we can consider that the compressor works only on the pressure and temperature of the fluid, keeping its speed unchanged, finally, since the machine does not move during its operation, even the difference in gravitational energy can be considered zero. Then:

$$l_{i,R407C} = c_{p,R407C}(T_{BT14} - T_{BT17})$$

Here $c_{p,R407C}$ is the specific thermal capacity of the refrigerant

It is important to verify the temperatures of the BT17 sensor, since when observing the available data and comparing it with the diagram (log(p) -h), certain operating points are detected at the compressor inlet (determined by temperature and pressure) . within the limit of the curve. This means that, at least in part of the transformation, the compressor could end up working with a partially liquid fluid instead of a totally gaseous fluid, which would cause malfunctions or breakdowns. This fact depends on the assumption of constant pressure, obligatory due to lack of data, but which inevitably leads to critical problems. The actual operation of the compressor will have an initial transient, during which the pressure will gradually increase, allowing it to always operate with a completely gaseous fluid. That said, the point of the curve is determined by the upper limit pressure of the low pressure branch to which a temperature will correspond, T_{satLP}. At this point if $T_{satLP}<T_{BT17}$ equation holds, then surely the compressor will work with totally gaseous fluid. All points that do not meet this condition have been discarded by the calculation of $l_{i,R407C}$ and therefore of the COP. In the same way, it will also be necessary to check the temperature detected by the BT14 sensor: the same procedure followed in the low pressure branch is applied. The high pressure at the outlet of the compressor, in the diagram (log (p) - h) the temperature is read (T_{satHP}) of the point at the intersection of the isobar and the upper saturation curve and, if $T_{satHP}<T_{BT14}$ then it means that the compressor also in this case works with a totally gaseous fluid. Again the points of operation detected by sensors, whose temperatures don`t respect this inequality, must not be considered in the calculation of $l_{i,R407C}$ and the COP. The temperature controls described above have greatly reduced the range of pressure variation. In the event of pressures greater than 23 bars in the high pressure branch or greater than 8 bars in the low pressure branch, the number of operating points to be excluded would be greater than 25% of the total, which would distort the averages. Therefore, the intervals considered in this analysis are 16-23 bar (for high pressure) and 7-8 bar (for low pressure).

We will now show the steps to calculate the specific work done by the compressor during one day and one of the pressure ranges. As an example, we have taken the date of 10/27/2019 and the pressure assumption 7-23bar. But at the end of the analysis, we will give all the days and all the chosen pressure ranges.

First, we find state values for the chosen pressures using a refrigerant calculator

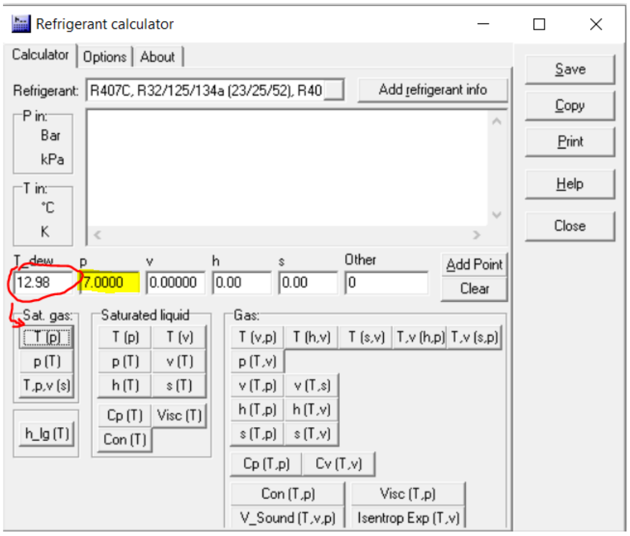

Figure 9: Refrigerant calculator. Input is pressure and output are saturated T and h

$p_L = 7bar$	$T_{sat\_gasLP} = 12.98°C$
$p_H = 23bar$	$h_{sat\_liq} = 286.44\ kJ/kgK$
	$T_{sat\_gasHP} = 56.34°C$
	$T_{sat\_liq} = 51.84\ °C$

To find the c_{pR407C}, we proceed to identify the maximum and minimum temperature values of the high pressure part of the circuit (compressor outlet) and of the low pressure part (compressor inlet), considering the period of one day . . By crossing these temperature values with those of pressure, the c_p was calculated, corresponding to each working point, using the refrigerant calculator. Finally, a daily average has always been made on the values identified. Therefore, the average specific heat will be considered constant during the reference day. We can see in the following table how to find an average c_{pR407C}.

10/27/2019			
$c_{pR407C\_average}$ kJ/kgK	1.059		
T_{minHP} °C	56.4	T_{minLP} °C	19.2
T_{maxHP} °C	73.5	T_{maxLP} °C	21.4

c_{p_maxHP} kJ/kgK	1.273	c_{p_maxLP} kJ/kgK	0.894
c_{p_minHP} kJ/kgK	1.173	c_{p_minLP} kJ/kgK	0.897

Now we can calculate instantaneous work done values of compressor for switching on conditions only. Here we will show average values of temperatures, and $l_{i,R407C}$ for 27/10/2019 date

27/10/2019	
T_{BT14_ave} °C	66.87
T_{BT17_ave} °C	20.31
l_{i_ave} kJ/kgK	49.315

After calculating the specific work done by the compressor, we proceed to consider the condenser, in which heat exchange takes place. In the condenser, the heat exchange with the user (water in our case) is calculated thanks to the enthalpy differences between the input and output state points of the condenser.

$$q_{H,R407C} = h_{BT14} - h_{sat.liq}$$

h_{BT14}: Enthalpy at the compressor output$\left[\frac{kJ}{kgK}\right]$

$h_{sat,liq}$: Enthalpy of saturated liquid (high pressure)$\left[\frac{kJ}{kgK}\right]$

At the end of transformation enthalpy ($h_{sat.liq}$), at the condenser outlet corresponds to the point of intersection between the high pressure isobar and the lower saturation curve and is therefore constant once a certain pressure level has been set. Instead, the enthalpy of the point at the condenser inlet is found through the (log(p)-h) diagram, as a function of temperature and pressure. As there was no software allowing an accurate evaluation of this enthalpy for all the measurements taken every 30s from the Nibe control unit, it was necessary to proceed differently. Operating points were randomly identified, the enthalpy of which was manually calculated through the diagram (log(p)-h). Exploiting a linear interpolation, all the enthalpy values of the other points were then obtained, in an automated way through spreadsheets. By way of example, random values of enthalpies corresponding to appropriate temperature points are chosen here.

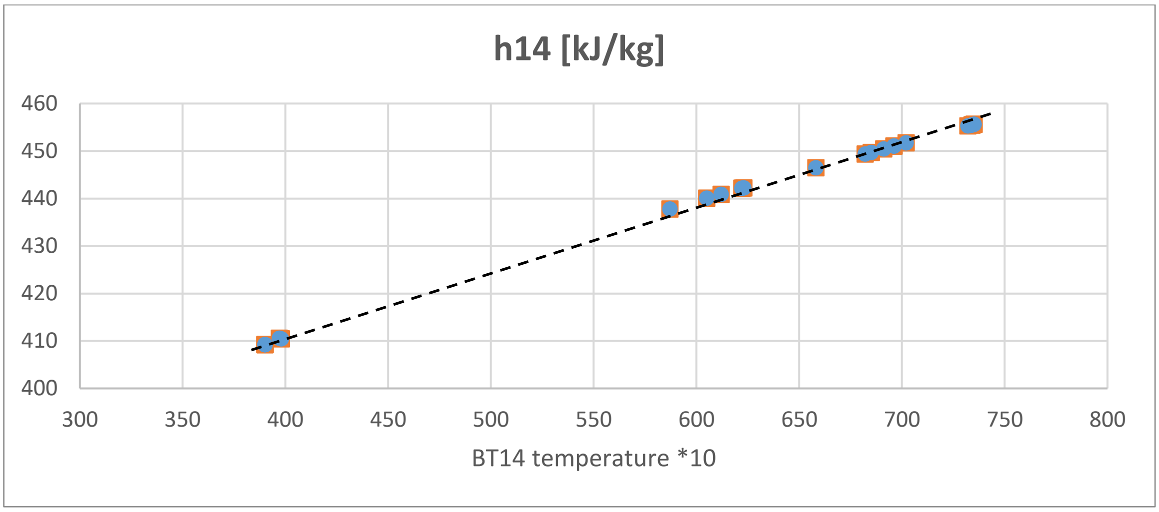

Figure 10: Linear interpolation of randomly calculated enthalpies for 27/10/2019

Finally, by carrying out sample checks, it was seen that the method followed led to having enthalpy values that differ very little from the real ones, therefore fully acceptable values for proceeding with the analysis.

Then we can calculate specific heat released by the condenser for each instantaneous value and then take an average

27/10/2019	
$q_{H,R407C\_ave}$ (kJ/kgK)	161.043

In conclusion, the COP was calculated as the ratio between the heat exchanged and the work for all the points where the compressor was on and for all the points that complied with the temperature controls described above. we have to ignore some "no gas" values that cause misalignment of the mean values. Here we can show that the maximum and minimum COP was taken on October 27 and an average value for one day (Figure 11).

27/10/2019	
COP_ave	3.29
COP_min	2.95
COP_max	3.93

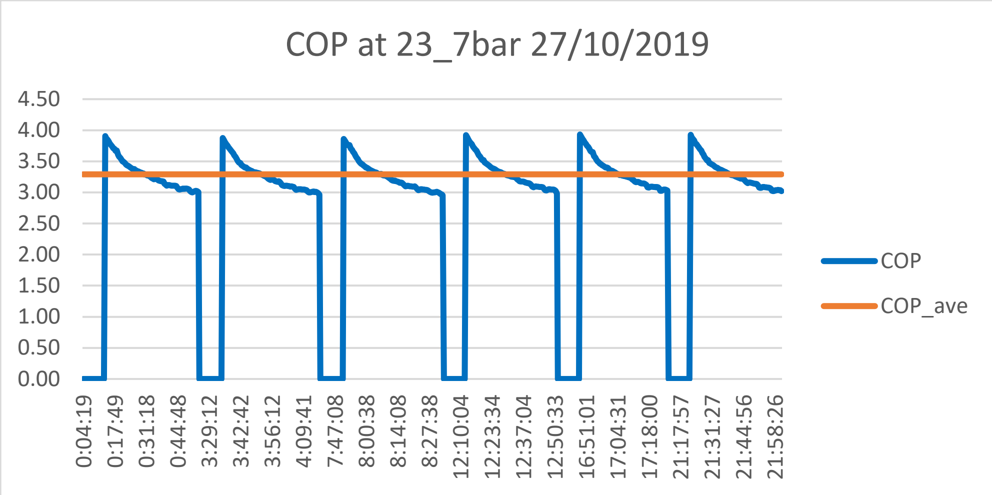

Figure 11: daily and average COP of the pump

Calculation of powers

The same statements made for the COP are also valid for the calculation of the powers: the operating points that involve the presence of liquid fluid inside the compressor are excluded and the calculations are made only for the operating points during which the compressor is on. . It would be useless to calculate the values of thermal power exchanged when the compressor is off because the flow of refrigerant in this particular situation would be zero. In this part of the analysis, the value of the flow measured through the flow meters comes into play. The only circuit whose flow is known is precisely the primary circuit, the power calculation must therefore start from the evaporator whose temperatures d input and output of the primary circuit are known. The heat exchanged between the brine and refrigerant circuits will be calculated using this formula:

$$Q_C = \dot{m}_{glycol} \cdot c_{p,glycol} \cdot (T_{BT10} - T_{BT11})$$

Here we have all data for the brine circuit. (Dusseldorf, 1991)

Density of glycol	ρ_{glycol}	*1026.3*	kg/m^3
Specific heat of glycol	$c_{p,glycol}$	*3.953*	*kJ/kg K*
Specific heat of water	$c_{p,water}$	*4.186*	*kJ/kg K*
Density of water	ρ_{glycol}	*1000*	kg/m^3
Volumetric flow rate	Q_{glycol}	*460*	*l/h*
Mass flow rate	$\dot{m}_{glycol}$	*0.1311*	*kg/s*

Without considering the particular energy losses, it can be said that the heat released by the glycol is the same as that which will be absorbed by the R407C refrigerant. Taking advantage of the power balance, it will also be possible to calculate the flow of refrigerant in the internal circuit of the heat pump by following the following calculation steps:

$$Q_H = Q_C + W_i$$

The heat released to the user is the sum of heat absorbed from the ground and supplied power to the compressor. If we write the equations of each power:

$$Q_H = \dot{m}_{R407C} \cdot q_{H,R407C}$$

$$W_i = \dot{m}_{R407C} \cdot l_{i,R407C}$$

$$Q_C = \dot{m}_{glycol} \cdot c_{p,glycol} \cdot (T_{BT10} - T_{BT11}$$

Now, by combining the above equations we can easily find the mass flow rate of the refrigerant fluid.

$$\dot{m}_{R407C} = \frac{\dot{m}_{glycol} \cdot c_{p,glycol} \cdot (T_{BT10} - T_{BT11})}{q_{H,R407C} -\cdot l_{i,R407C}}$$

By setting equality between the heat released by the refrigerant and that absorbed by the heating fluid, the water flow rate in the secondary circuit can also be easily found, which is constant (as expected given the constant operation of the GP1 pump). Here the temperatures of the secondary circuit can be measured with sensors.

$$Q_H = \dot{m}_{water} \cdot c_{p,water} \cdot (T_{BT12} - T_{BT3})$$

$$\dot{m}_{water} = \frac{Q_H}{c_{p,water} \cdot (T_{BT12} - T_{BT3})}$$

27/10/2019			
The average mass flow rate of refrigerant	$\dot{m}_{R407C\_ave}$	*0.013*	*kg/s*
Average specific heat absorbed in evaporator	q_{C_ave}	*11.075*	*kJ/kg*

By summarizing all results we can get three main powers of the system are:

$$Q_C = \dot{m}_{glycol} \cdot c_{p,glycol} \cdot (T_{BT10} - T_{BT11})$$

$$Q_H = \dot{m}_{R407C} \cdot q_{H,R407C}$$

$$W_i = \dot{m}_{R407C} \cdot l_{i,R407C}$$

27/10/2019			
Thermal power absorbed in evaporator (cold side)	$Q_{C,ave}$	*1.45*	*kW*
Thermal power released in condenser (hot side)	$Q_{H,ave}$	*2.09*	*kW*
Power supply to compressor	W_i	*0.64*	*kW*

Coolpack

Coolpack is a software for refrigeration systems developed by the energy engineering department of the University of Denmark (CoolPack, 1995), in which there are different useful tools for the design or control of refrigerators, air conditioners, pumps heat, anything that works with fluid refrigerant. , for example, an R407C, as in the case of this study. The Coolpack tools that were used for this analysis are called "Refrigerant Calculator", which allows to know the characteristic quantities of a point of the diagram (log(p)-h) having known, for example, the temperature and the pressure, "Refrigeration Utilities", with function of drawing cycles in the diagrams (log(p)h), "Heat transfer fluid calculator", with which it is possible to calculate the characteristics of the primary circuit fluid such as density, heat exchange coefficients or others, "Cycle analysis", with which it was possible to analyze the quantities and energy yields of a thermodynamic cycle.

Cycle analysis

One of the main parts of our study is to calculate the COP of the heat pump through the Coolpack software, and this can be done in the cycle analysis section. To calculate all the desired quantities, we need to enter some state values taken from the refrigerant calculator and calculated using data from the control unit. We built the cycle analysis for 4 pressure ranges and every 8 days, and compared the results with the average values for each day. Here too we only show for one day (27/10/2019) and one pressure range (7-23 bar) as an example.

We already have the status results for the temperatures and the refrigerant mass flow for the corresponding pressures for each day, and we have also entered the power supplied to the compressor and the daily average temperatures upstream and downstream of the compressor which have been calculated in the COP part of the analysis. Here is the input data needed for cycle analysis in the cycle specification window and the software automatically calculates the COP and thermal powers in the evaporator and condenser.

27/10/2019 7-23bar	
T_{sat_gasLP}	12.98℃
T_{sat_gasHP}	56.34℃
T_{sat_liq}	51.84 ℃
W_i	0.64 kW
$\dot{m}_{R407C\_ave}$	*0.013 kg/s*
T_{BT14_ave}	66.87 ℃
T_{BT17_ave}	20.31 ℃

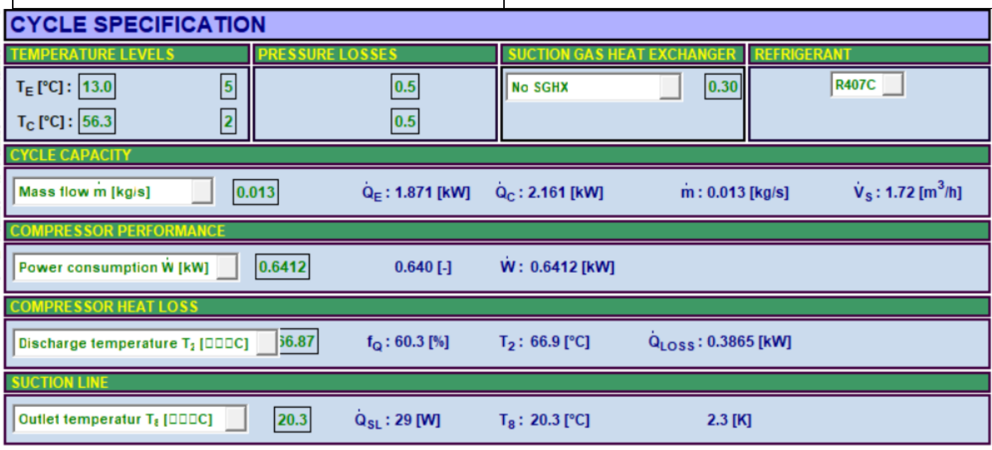

Figure 12: Input data section of cycle analysis.

And by calculating system builds a closed cycle of the system and shows all calculated values on it.

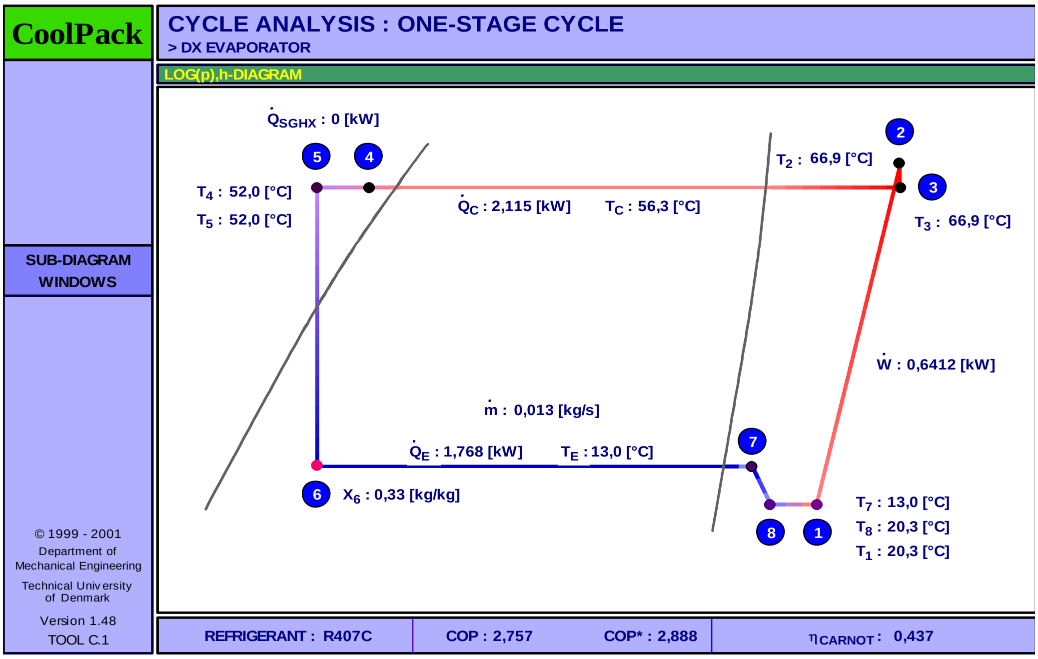

Figure 13: The cycle analysis diagram for 7-16bar and 27/10/2019

Achieved results are:

27/10/2019 7-23bar		
	Calculated in excel	Calculated in Coolpack
$Q_{C,ave}$ kW	*1.45*	1.768
$Q_{H,ave}$ kW	*2.09*	2.115
COP	3.290	3.295

Attention: as we can see on the Coolpack screen, the COP value is calculated, but this COP value has been taken from the ratio between the thermal power absorbed by the condenser and the compressor supply. Because all refrigeration cycles were focused on cooling the environment and their COPs are given based on these criteria. However, our goal is to heat the user by taking energy from the ground and the useful energy is released in the capacitor. In our case, we have to manually calculate the COP from the Coolpack results using the powers calculated in Coolpack.

$$COP = \frac{Q_{H,coolp}}{W_i}$$

It is obvious that the Coolpack results are slightly different from the calculated ones. Because Coolpack automatically takes into account certain energy losses and isentropic efficiencies that we consider to be without losses and without efficiencies.

Comparison of COP values

After the calculations, we get a lot of results from Excel and Coolpack. First of all, we pay attention to the COP of the heat pump in the analyzed periods. And it can be seen that the COP of the system strongly depends on the outside ambient temperature, on the pressures upstream and downstream of the heat pump, as well as on the heat losses during the cycle. Also, we must say that all COP results are less than the design value of COP_d which is given in the installation manual document. For example, for an outdoor temperature of 12℃ COP_d=4.86 (NIBE Installation Manual, NIBE F1155) They must be lower because the theoretical efficiency is usually always higher than the real one, calculated on experimental data. A fortiori, the outside temperatures on which this analysis is based are even higher than the temperature at which the calculation was made to determine the COP_d. This means that if this study had been carried out with outside temperatures around 12°C, surely the calculated COP would have been lower than COP_d,

10/2019	T_ave ℃	7-16bar		8-16bar		7-20bar		7-23bar	
days	external	COP_ave	COP_colp	COP_ave	COP_colp	COP_ave	COP_colp	COP_ave	COP_colp
24	16.2	4.85	4.584	4.66	4.470	3.82	3.757	3.16	3.171
25	16.3	4.70	4.585	4.68	4.565	3.94	3.888	3.29	3.296
26	16.6	4.73	4.613	4.68	4.569	3.95	3.891	3.29	3.295
27	16.7	5.02	4.617	4.97	4.567	3.95	3.896	3.29	3.299
28	17.2	4.77	4.652	4.72	4.307	3.98	3.927	3.32	3.325
29	17.3	4.71	4.587	4.68	4.558	3.93	3.873	3.29	3.295
30	16.9	4.61	4.486	4.57	4.450	3.83	3.775	3.23	3.232
31	13.8	4.41	4.302	4.35	4.250	3.66	3.610	3.10	3.109

Calculated average COP and COP of Coolpack fir each day and each pressure set

If the outdoor temperature is high, the system COP will also have a higher value. We see very quickly how the lowest average daily COP was obtained on the last day of operation (31/10/2019). Looking at the table below, we understand how this is clearly determined by the fact that the outside temperature was significantly lower than the previous days. 13.8℃

The next important factor is that the pressure ranges chosen are directly dependent on the COP and if the lowest and highest pressures are close together then the COP of the system will be higher. In the table it can be seen that the highest COP values correspond to the 7-16 bar pressure setting, while the lowest were taken from the 7-23 bar pressure setting calculations. Moreover, we can easily understand from Figure 14 how the COP depends on the pressure range. The COP trends for the 7-16 bar and 8-16 bar conditions are very similar because their pressure ranges are close to each other. Some dips from zero on the chart are due to "out of gas" conditions

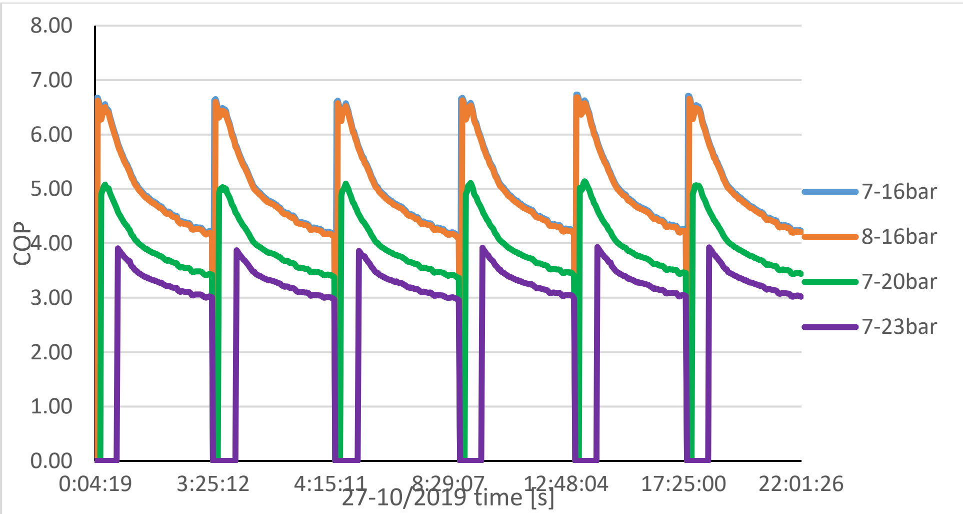

Figure 14: The daily COPvalues of each pressure settings for the 27/10/2019 day

Finally, the main important comparison is between the results calculated in Excel and the Coolpack software. And it is clear that almost all the COPs taken from Coolpack were lower than the average results calculated in excel. But in the last pressure range, the COP of Coolpack is slightly higher than calculated. This is due to the pressure range because at high pressure the COP of the system drops drastically. We can explain this difference by considering losses and assumptions. At first, we consider several assumptions to calculate in Excel, but some irreversibilities are never ignored by Coolpack during cycle analysis and they surely affect the results. Actually, we cannot avoid some losses in the condenser, compressor and evaporator. But the differences are not huge and it may be a factor to consider that our heat pump is working regularly. Here we will show the percentage differences between the measured results and the COP Coolpack results. Finally, the main important comparison is between the results calculated in Excel and the Coolpack software. And it is clear that almost all the COPs taken from Coolpack were lower than the average results calculated in excel. But in the last pressure range, the COP of Coolpack is slightly higher than calculated. This is due to the pressure range because at high pressure the COP of the system drops drastically. We can explain this difference by considering losses and assumptions. At first, we consider several assumptions to calculate in Excel, but some irreversibilities are never ignored by Coolpack during cycle analysis and they surely affect the results. Actually, we cannot avoid some losses in the condenser, compressor and evaporator. But the differences are not huge and it may be a factor to consider that our heat pump is working regularly. Here we will show the percentage differences between the measured results and the COP Coolpack results.

Table: percentage difference of COP between calculated and Coolpack results

	7-16bar		8-16bar		7-20bar		7-23bar	
10/2019 days	26	27	28	30	27	31	27	29
COP_ave	4.73	5.02	4.72	4.57	3.95	3.66	3.29	3.29
COP_colp	4.613	4.617	4.307	4.45	3.896	3.61	3.299	3.295
Diff in %	2.54	8.73	9.59	2.70	1.39	1.39	-0.27	-0.15

We can see from the table that the lower pressure ranges lead to a greater difference in COP between the calculated data and Coolpack. In Figure 39 and Figure 40 we show the trend lines of calculated COP and Coolpack values for randomly selected pressures and days.

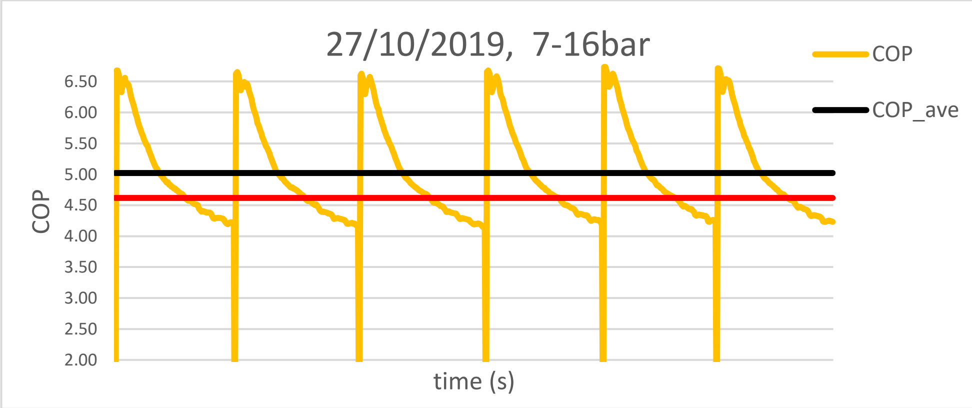

Figure 3:The daily, average, and Coolpack COP for 27/10/2019 at 7-16bar

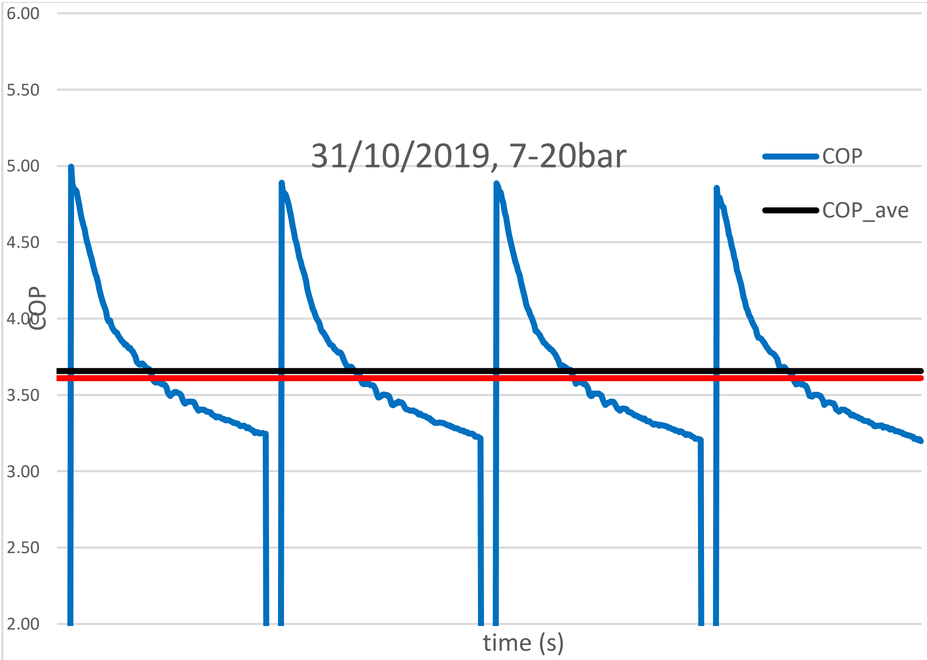

Figure14: The daily, average, and Coolpack COP 31/10/2019 at 7-20bar

Comparison of thermal powers results

Due to the thermal power exchanged in the condenser, we can conclude that a greater pressure difference provides more energy to heat the user because, in the compression process, the refrigerant fluid increases its pressure and temperature and, therefore, the fluid yields more heat. Energy. It is clear that the thermal and electrical powers as well as the COP for the date of 10/24/2019 are relatively higher than the other days, while the results of the other days are not very different. We can explain this condition by looking at the data obtained from the NIBE monitoring system. And on that day the heat pump only turned on 4 times in 24 hours, but on other days the heat pump turned on 7-8 times. When we calculated the average values for each day, the results for October 24 were different. As mentioned above, the compressor was not always active and its working time was from 20 minutes to an hour on average. Then it stopped 3-4 hours and so on.

Table: the average and Coolpack values of thermal power in kW for all days and pressure sets.

10/2019	7-16bar		8-16bar		7-20bar		7-23bar	
	$\dot{Q}_{H,ave}$	$\dot{Q}_{H,coolp}$	$\dot{Q}_{H,ave}$	$\dot{Q}_{H,coolp}$	$\dot{Q}_{H,ave}$	$\dot{Q}_{H,coolp}$	$\dot{Q}_{H,ave}$	$\dot{Q}_{H,coolp}$
24	3.177	3.128	2.909	2.874	3.394	3.414	4.151	3.618
25	1.949	1.947	1.952	1.950	2.020	2.03	2.089	2.111
26	1.949	1.942	1.949	1.947	2.016	2.026	2.086	2.107
27	1.944	1.814	1.928	1.816	2.029	2.038	2.094	2.115
28	1.962	1.959	1.966	1.834	2.037	2.047	2.096	2.118
29	1.961	1.959	1.966	1.964	2.027	2.035	2.105	2.125
30	1.914	1.911	1.923	1.920	1.970	1.978	2.074	2.094
31	1.922	1.920	1.837	1.834	1.902	1.908	2.009	2.028

We can show in figure 15 the released power trends for different pressure ranges for the randomly selected day

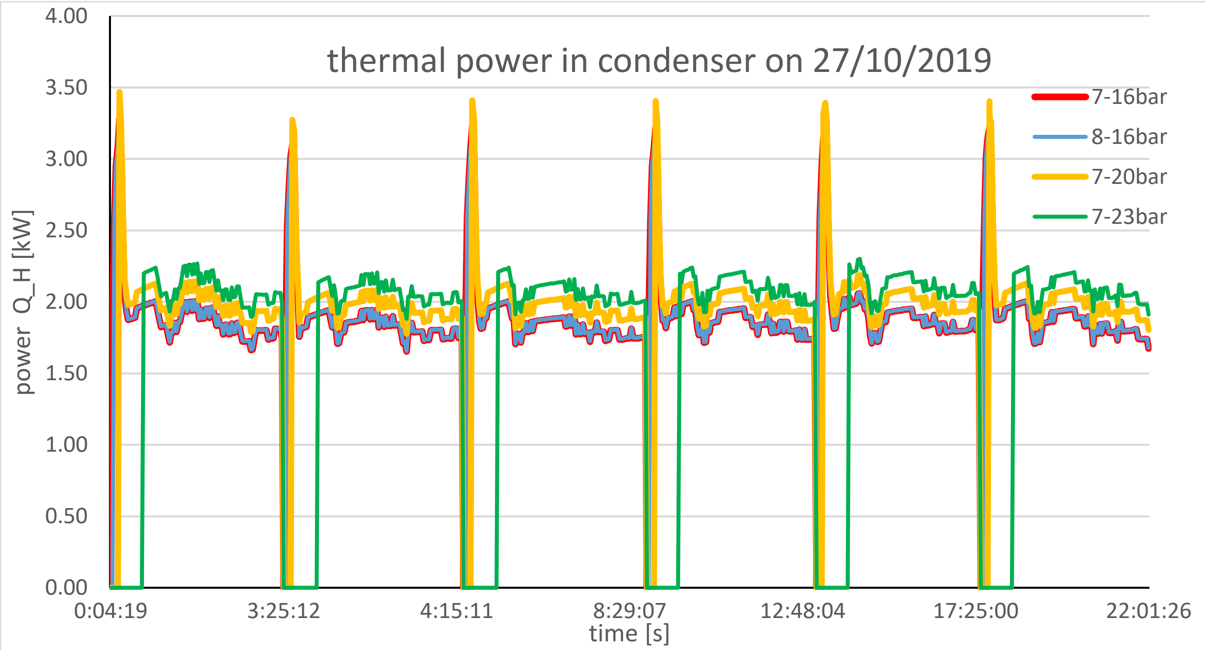

Figure 15: daily thermal power exchanged in condenser on 27/10/2019

For the electrical energy supplied to the compressor, we can give information for different days and different pressures, but not for calculated values and Coolpack. Because we use average power values as input value for Coolpack.

Table: Average electrical power consumed by the compressor for all days and pressures

W_i (kW)				
10/2019	7-16bar	8-16bar	7-20bar	7-23bar
24	0.682	0.643	0.909	1.141
25	0.425	0.427	0.522	0.641
26	0.421	0.426	0.521	0.640
27	0.393	0.398	0.523	0.641
28	0.421	0.426	0.521	0.637
29	0.427	0.431	0.525	0.645
30	0.426	0.432	0.524	0.648
31	0.446	0.432	0.529	0.652

The highest power (highlighted in the yellow cell) was consumed on the last day because the outside temperature dropped drastically compared to the other days and in the case of a high pressure set (7-23 bar). The results of the first day (light green cells) are different from the others due to the lack of active periods of the pump. So we didn't pay much attention that day. On the contrary, if we look at the power trends in Figure 16, we can see how they increase as the pressure level increases, clearly denoting a higher energy demand from the compressor because if we want more compression (higher pressure), we have to give more energy.

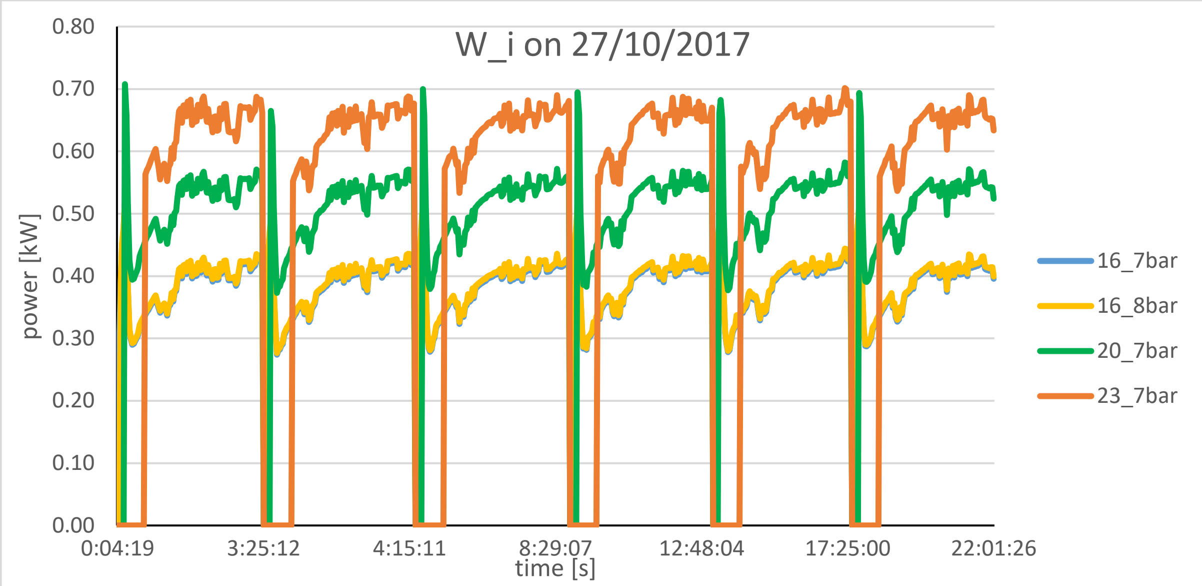

Figure16: The daily power supply trends for different pressure ranges on 27/10/2019

The figures above show the trends of COP, Q_H and W_i with the assumption that the times when the compressor is off are not represented on the x-axis, so everything is condensed for better graphical representation. In any case, the average operating COP calculated for each degree of pressure throughout the time to which these data refer is 4.12, while the average value of COP_coolpack is 4.025 and these values respect what has been said above regarding the COP_d.

Table:percentage increase of thermal and electrical power in different pressures for 28/10/2019

	Pressure levels ratio 8-16bar/7-16bar	Pressure levels ratio 7-20bar/7-16bar	Pressure levels ratio 23-7bar/20-7bar
% increase of average $\dot{Q}_H$	0.22	3.84	2.92
%increase of average W_i	1.12	23.78	22.22

Conclusion

Generally, people think that geothermal heat pump systems are a new and unknown technology, but it is not. Simply, a geothermal system has no visual impact, so it is impossible to understand if a building is powered by this type of system.

The results obtained from this analysis allowed us to conclude that a geothermal heat pump can be used safely even in a residential environment since its operation does not present any type of acute anomaly. Furthermore, the use of such a machine would certainly bring advantages in environments where it is necessary to move harmful emissions. However, its high installed cost would be recovered over time through very low consumption, especially if combined with low-emission power generation methods (such as wind turbine or solar panel installations). These high costs mean that they remain niche installations, despite being part of a growing market. Moving on to the more analytical level of the study, it should be noted that certain assumptions, such as the constancy of pressure, have turned out to be very restrictive, so if you want to go deeper into this analysis, you must mount pressure sensors. in the heat pump circuit. . By having instantaneous data, the precision will surely be better, especially if a program is also used that allows the precise calculation of the enthalpy values at the compressor outlet (remember that in this study they were identified by linear interpolation). With all these precautions, it would also be possible to take into account the transients present when the compressor is on, thus obtaining a complete reading. In addition, the COP of the system can be increased by installing a deeper brine circuit and a ground location with better conductivity. Because in the center the brine circuit pipes were installed after the building was finished and there was not enough space available to mount them.

Bibliography

1 CoolPack. (1995). Retrieved from https://www.ipu.dk/products/coolpack/

2 GEONOVIS energia geotermica. (2019). Retrieved from geonovis.com: https://www.geonovis.com/geotermia/prodotti/nibe-f1155/

3 Geothermal energy. (n.d.). Retrieved from green network: https://greennetwork.it/en/renewable-energy-sources/renewable-sources/geothermal-energy/

4 IOP institute of physics. (n.d.). Ground source heat pump. Retrieved from https://www.iop.org/explore-physics/sustainable-building-design/ground-source-heat-pump#gref

5 Lund1, B. S. (2004). GEOTHERMAL (GROUND-SOURCE) HEAT PUMPS. ResearchGate, 1-2.

6 M.H. Dickson, M. F. (1990). Geothermal energy and its utilization. Retrieved from https://www-sciencedirect-com.ezproxy.biblio.polito.it/science/article/pii/S0375650500000560#BIB9.
7 NIBE. (n.d.). NIBE F1155 Installer manual. Retrieved from NIBE.com: https://www.nibe.eu/assets/documents/27897/331342-6.pd
8 REN21. (2021). Retrieved from https://www.ren21.net/why-is-renewable-energy-important/
9 Rivas, P. (2020). GEOTHERMAL HEATING. RENEWABLE SOIL ENERGY. Retrieved from instalacionesyeficienciaenergetica.com: https://instalacionesyeficienciaenergetica.com/geothermal-heating/
10 Rondina, R. (2020). Analisi sperimentale di una pompa di calore geotermica in ambito residenziale. Tesi, 12.
11 Tofalo, F. (2019). Sperimentazione di muri energetici. Tesi di Laurea Magistrale, 86.
12 U. d. (2011). Guide to Geothermal heat pumps.
13 Experimental Analysis of Geothermal heat pump thesis. Madumar Musurmonov (2021).