DELIVERABLE 2. MODEL DESCRIPTION

Research project: Design and evaluation of descarbonisation strategies to achieve near zero emissions indoor swimming pools assisted by renewable energies – nZEPools (TED2021-131173B-I00)

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1. Introduction

One of the specific objectives of this research project is the development of a numerical model in TRNSYS which let us dynamically simulate the thermal and energy performance of the heat pump designed in work package 1 (WP1), working alone and coupled with the installation that initially existed in the swimming pool of La Aljorra which was formed by a set of solar thermal collectors and two boilers which satisfied the heating needs of the pool and the domestic hot water (DHW) demanded by the changing rooms. As was already mentioned, through this project a transcritical CO2 heat pump is included in the system which will support the hot water production of the original system. A photovoltaic facility will also be included in this new design to satisfy part of the electricity demand of the installation.

As described in deliverable 1, the heat pump is an air-to-water transcritical CO_2 one, which consists of a compressor, a gas cooler for the production of domestic hot water, another gas cooler to heat the swimming pool water, an electronic back-pressure valve to guarantee optimum high pressure, a liquid receiver, an electronic thermostatic expansion valve to control the superheating, and an evaporator (Figure 1).

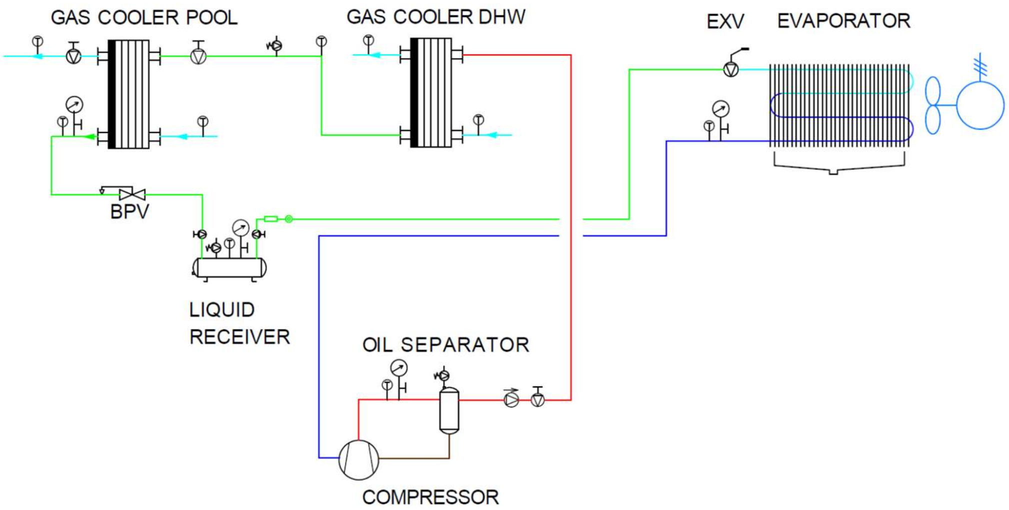

Figure 1. Sketch of the heat pump.

So, through this deliverable, firstly, the different components that make up the heat pump designed in WP1 are explained paying attention to their characteristics, and how they have been programmed. Later on, a description of the rest of the devices comprising the installation included the photovoltaic facility is given.

2. Description of the model

As mentioned, this section describes the different models developed to characterise the behaviour of all heat pump components.

2.1. Compressor

The component created in TRNSYS to characterise the compressor has been carried out by considering the polynomials provided by the manufacturer according to AHRI standard 540 which has the general expression

$$y = c0 + c1 * t_o + c2 * p_{HP} + c3 * t_o^2 + c4 * t_o * p_{HP} + c5 * p_{HP}^2 + c6 * t_o^3 + c7 * p_{HP}^2 + c8 * t_o * p_{HP}^2 + c9 * p_{HP}^3$$

where t_o is the evaporation temperature, $p_{HP}$is the pressure at the compressor outlet and the coefficients (c0 to c9) are polynomial adjustment coefficients whose values depend on the variables of interest calculated with the expression. In this case, the power absorbed by the compressor and the mass flow rate. In Table 1, the coefficients needed to determine these variables are gathered.

Table 1. Polynomial coefficient of the compressor according to AHRI 540.

	$\dot{W}$ [W]	$\dot{m}_r$ [kg/s]
c0	-4900	0.13637282
c1	-251.427652	0.004073265
c2	207.426276	-0.000523728
c3	-3.98329046	5.9823E-05
c4	4.06463936	9.25074E-07
c5	-1.27046659	8.32364E-07
c6	-0.02547129	5.91156E-07
c7	0.02304809	1.74539E-08
c8	-0.01298627	-5.4591E-08
c9	0.00326564	5.80367E-10

Once these variables are estimated, the output variables of the model are obtained taking into account that the input variables are inlet pressure, inlet enthalpy and outlet pressure, so the outlet enthalpy is

$$h_{ro} = h_{ri} + \dot{W}/\dot{m}_r$$

Isentropic efficiency of the compressor is also calculated as $\eta = (h_{rso} - h_{ri})/(h_{ro} - h_{ri})$ where h_{rso} is the enthalpy at the isentropic outlet.

Thermodynamics properties as h_{rso} and those needed to calculate it are determined by means of CoolProp subroutines available with TRNSYS 18 distribution.

2.2. Gas coolers for DHW generation and heating the pool water

Based on the geometric data provided by the manufacturer, we have developed a model in Matlab to characterise plate heat exchangers that function as evaporators, condensers, and in single-phase exchangers. This model allows us to simulate different operating conditions of the exchangers according to their purpose, whether in gas cooler pool or gas cooler domestic hot water.

To achieve this, we have defined a set of input variables (control, p_{ri}, m_{ri}, h_{ri} , P_{seci} m_{seci} and T_{seci}) which, when varied, generate specific outputs in the model. Thus, we have obtained a wide set of operating conditions for each type of exchanger.

Using the results obtained, we performed a least-squares regression in Matlab, generating polynomials that predict the output variables (Q_{gc} , T_{ro}, p_{ro}, m_{ro}, h_{ro} , P_{seco}, m_{seco} and T_{seco}) based on the input variables except P_{seci}. In this analysis, we considered the input variables as dependent variables. To this end, refrigerant flow rates were adjusted within a specific range. For the temperature of water entering the DHW gas cooler, the values considered are $T_{\text{DHWgc,i}}$ 10, 25, 40, and 55ºC, three water mass flow rates are considered in this heat exchanger 0.27, 0.3, and 0.33 kg/s which is ±10% of the nominal value 0.3 kg/s used in the heat exchanger design.

For the temperature of water entering the gas cooler used to heat the pool water, the values considered are $T_{Poolgc,\,i}$ 25, 27, and 29ºC and the water mass flow rate considered was 1.44747, 1.6083, and 1.76913 kg/s which is ±10% of the nominal value 1.6083 kg/s used in the heat exchanger design. Refrigerant inlet pressures considered in both heat exchangers are 75, 80, and 85 bar. Inlet pressure at the water side is considered constant for all test cases and equal to 1 bar. Refrigerant mass flow rate and inlet enthalpies are calculated by considering several inlet conditions for the compressor and by using the polynomial presented above. To do so three superheatings are considered 0, 5, and 10ºC. Four temperatures of air at the evaporator inlet are taken into account 0, 10, 20, and 30. Three temperature differences between the refrigerant at the compressor inlet and the air inlet equal to 3, 4, and 5ºC. This approach allows us to model and predict exchangers efficiency under different operating scenarios accurately and efficiently.

2.3. Metering devices

The electronic valves existing in the system are both modelled as isenthapic valves.

The back pressure valve keeps the discharge pressure at the optimum value. It is calculated by means of the following expression and is kept between 75 and 90 bars.

$$P_{optimum} = min\left(90, \left(c_0 + c_1 T_{gc,out} + c_2 T_o + c_3 T_{ri}\right)\right) \times 1000 \text{ Pa}$$

Where $T_{gc,out}$ is the outlet temperature corresponding to the cooler gas, T_o is the evaporation temperature at the compressor inlet and T_{ri} is the refrigerant temperature at the compressor inlet. In Table 2, the coefficients needed to determine these variables are gathered.

Table 2. Polynomial coefficient of the optimum pressure expression.

	$p_{optimum}$ (Pa)
c0	75158.88299
c1	0.050893266
c2	-28408.79656
c3	-3293.686668

The thermostatic expansion valve guarantees that the value of the superheating is 5ºC.

2.4. Evaporator

The evaporator used in the installation is a Güntner evaporator model GACC CX 040.1/2WN/HJA4A.UNNN, it has been modelled by means of a regression carried out in MATLAB which relates the different inlet and outlet variables of the system. The main parameters we are interested in are inlet pressure, outlet pressure and enthalpy and pressure loss. Total and sensible powers are also calculated, being the latter, less relevant in the analysis.

Initially, a text file with the necessary data is imported, making sure that the format is readable by MATLAB. Each column of the file is assigned to specific parameters such as air relative humidity, air temperatures, refrigerant mass flow rate, superheat, inlet and outlet enthalpy, and inlet and outlet pressures. These variables are obtained by using IMST-ART program. To do this, 4732 test cases are defined. The inputs variables required by this software are relative humidity and air temperature at the inlet. Volume flow rate of air is set to 5325 m^3/h which is the nominal value provided by the manufacturer. Refrigerant mass flow rate, superheating, and subcooling are also needed. To define them, we need to considered a fictitious vapour compressor cycle with a fixed condensation temperature 30ºC whose subcooling will correspond to different gas cooler outlet temperatures of 28, 30, and 32ºC. The values considered for the superheating are 3, 6, 9, and 12ºC. Discharge pressures considered in these tests are 77, 78, 80, 82, and 86 bar.

Mass flow rate is evaluated by taking into account the compressor polynomials presented above and different values of the discharge pressure and the evaporation temperature which is obtained by considering different air inlet temperatures and different temperature differences between air temperature at the inlet and refrigerant temperature at the compressor suction.

The data are then organised in a table to facilitate the application of separate regressions for each relevant parameter. These models are fit using a trial-and-error method to obtain the best possible correlation in each case and MATLAB generates plots and fit coefficients that allow the accuracy of each regression model to be evaluated.

Finally, the expressions and coefficients obtained are integrated into a global model in MATLAB, which will allow us to simulate and analyse the behaviour of the system as a whole.

The component created in TRNSYS to characterise the evaporator has been made considering the polynomial provided by the regressions obtained in MATLAB, which describes the behaviour of parameters such as inlet pressure, outlet pressure, pressure loss, and total and sensible powers. This polynomial allows the evaporator efficiency to be accurately modeled by integrating it into the complete system simulation in TRNSYS, thus facilitating a detailed analysis of its operation under different conditions.

$$y = c_0 + c_1 HR + c_2 T_{seci} + c_3 \dot{m}_{ri} + c_4 SH + c_5 h_{ri} + c_6 HR\, T_{seci} + c_7 HR \dot{m}_{ri} + c_8 * T_{seci} \dot{m}_{ri} + c_9 HR\, T_{seci} \dot{m}_{ri}$$

where *HR* is the relative humidity, T_{seci} is the air inlet temperature and finally, $\dot{m}_{ri}$ and h_{ri} are the refrigerant mass flow rate and the refrigerant enthalpy at the inlet. The coefficients c_0 to c_9 are polynomial adjustment coefficients which depend on the variable considered, namely; refrigerant pressure drop, refrigerant inlet pressure or the evaporator sensible heat. Their values for each variable are gathered in Table 3.

Table 3. Polynomial coefficients of the evaporator.

	Δp_r	p_{ri} [Pa]	Q_s[W]
c_0	-167.91	3127.7	19.988
c_1	0.7922	-7.4552	-0.012677
c_2	-5.7497	89.9	0.394
c_3	0.88171	-5.2746	0.052048
c_4	3.1894	-59.149	0.00047732
c_5	0.017788	3.7602	-0.064815
c_6	0.058347	-0.21241	-0.0036722
c_7	-0.003767	0.044055	-0.00024005
c_8	-0.00054127	0.041054	-0.00046201
c_9	-7.1975E-05	0.0004092	-3.7609E-06

All other thermodynamic properties are calculated by using the 'CoolProp' subroutines included in TRNSYS 18 software.

3. Global model. Integration of the heat pump model in the global model/ Initially existing photovoltaic system and heat pump couplings

The municipal swimming pool of La Aljorra, located in Calle Américo Vespucio, has a surface area of 212.5 m² and a water volume of 261.43 m³. To allow year-round use, it has been covered with a sandwich structure made of steel and fiberglass, and both the pool basin and the surrounding environment have been air-conditioned.

Initially, the heating system had vacuum-tube solar collectors on the roof, supported by two diesel-fired boilers (151.16 kW for domestic hot water and 348.8 kW for the pool), with a 5,000-liter fuel tank. In recent years, the average consumption of diesel fuel has been around 40,000 liters per year, although the goal is to reduce this figure by including a heat pump.

The solar installation has 30 Trisolar solar collectors of 2.4 m² each, instead of the 40 Viessmann collectors originally planned in the original swimming project. The collectors have a slope of 45° and face south. In addition, the system has a 3,000-liter solar storage tank and a 738-liter DHW storage tank, which stores and distributes the heated water.

The original system layout is depicted in Figure 2, which, as can be seen, includes solar collectors, two water boilers for the pool and the changing rooms, and storage tanks, configured to optimise the supply of thermal energy to the pool.

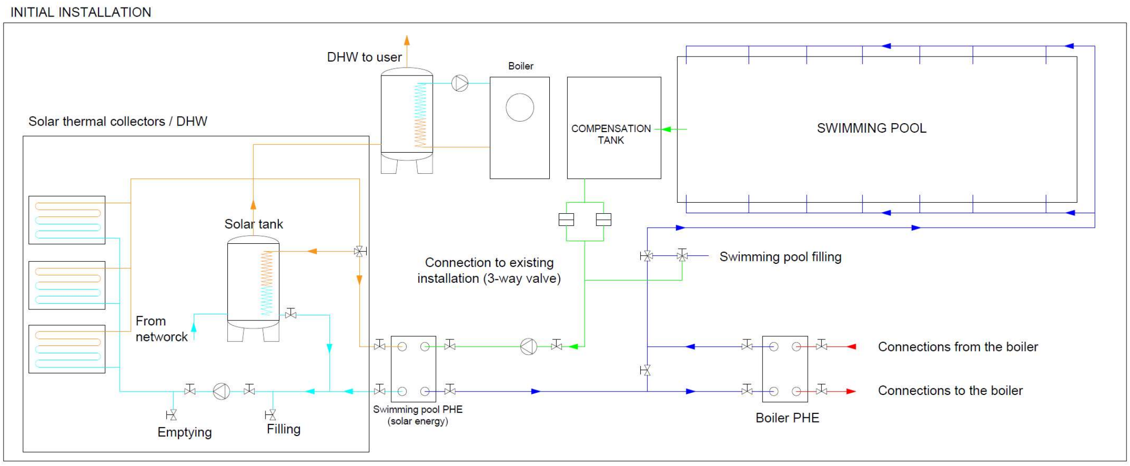

Figure 2. Schematic of the initial installation at the Aljorra swimming pool.

On the other hand, Figure 3 shows the installation after including the heat pump.

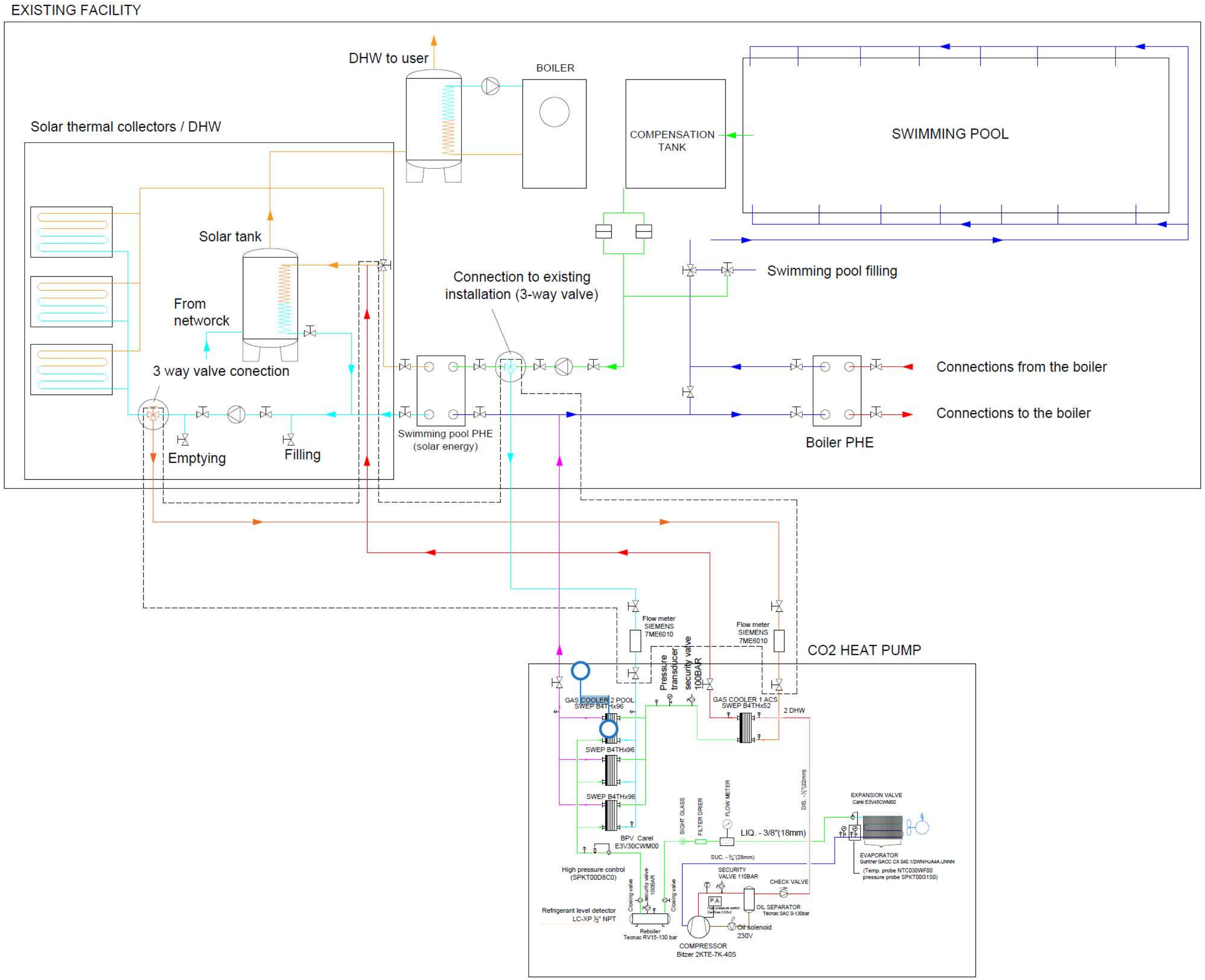

Figure 3. Installation including the heat pump.

It includes the heat pump and shows how it is coupled to the initial installation. As shown, the first gas cooler provides heat for the DWH generation when it is necessary and the set of three plate heat exchangers heat up the water coming from the swimming pool jointly with solar field. If the solar collectors and the heat pump are not able to heat up the water, the boilers will provide the necessary heat to reach the temperature set points of each circuit. Figure 4 shows two different views of the heat pump as it is in the swimming pool engine room.

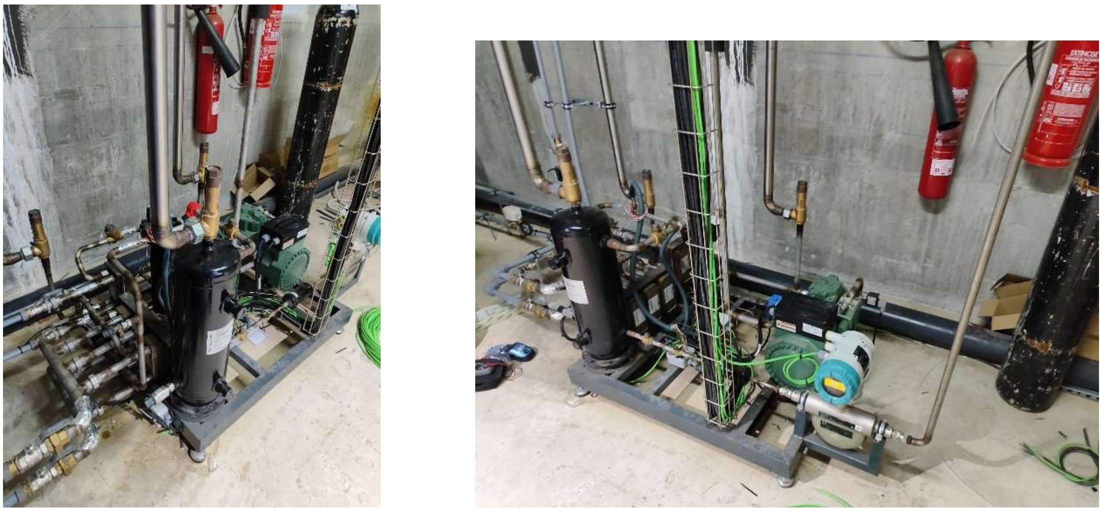

Figure 4. Two views of the heat pump constructed in the swimming pool.

In the framework of this research project, a photovoltaic generation system has been installed. This has been designed to cover approximately the electrical demand of the heat pump installed, it includes compressor, impulsion pumps and dehumidifiers. This initial facility has been modelled in TRNSYS18, the model used for the original swimming pool facility is that presented in (Velasco et al., 2024). The photovoltaic panels are modelled by means of Type 103 and the inverter by Type 48. Their properties have been obtained from the manufacturer's catalogues. Table 4 describes their main properties. It is formed by 14 Jinko Solar panels, together with a 50 kW Huawei inverter.

Device/System	Manufacturer/Model	Specifications
14 Photovoltaic solar panels (550 kWp)	Jinko Solar JKM550M-72HL4-V	7,7 kWp
1 Inverter	Single phase Huawei 6KTL L1	50 kW
2 Batteries	Huawei LUNA2000	10 KWh

Table 4. Properties of the photovoltaic installation.

Figure 5. View of the photovoltaic panels mounted at the swimming pool roof.

The individual models of each system have been integrated in a TRNSYS global model of the facility which is depicted in Figure 6.

References

AHRI Standard 540, Standard for performance rating of positive displacement refrigerant compressors, 2020.

F.J.S. Velasco; J. Giménez-Villa; J.R. García-Cascales; F. Illán-Gómez; J.P. Delgado Marín; R.A. Otón-Martínez, Comparative analysis of the behaviour of the hot water production system of an indoor swimming pool, Journal of Physics: Conference Series, Eurotherm 2024, 9th European Thermal Science Conference, 10-13/06/2024, Bled, Slovenia.

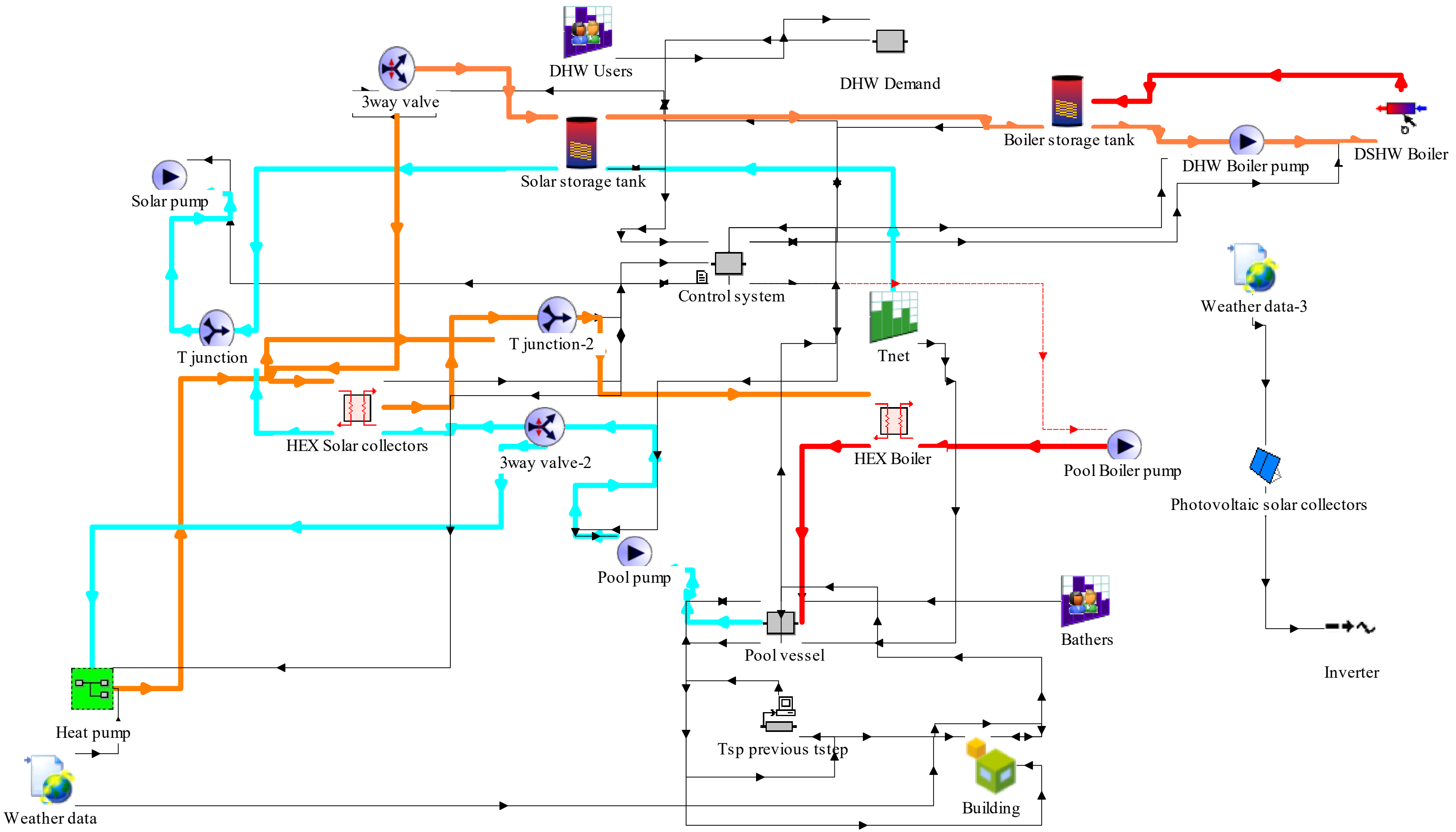

Figure 6. TRNSYS model of the swimming pool facility including the heat pump.