



A new concept of industrial multi-energy systems with a multi-stage R718 high temperature heat pump

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Abstract

To reach global climate change targets, new concepts for industrial energy systems are needed. High-temperature heat pumps (HTHPs) for process heat electrification are a promising and efficient waste heat recovery and process heat generation technology. However, the limits of the achievable temperature, the need for a high COP, the temporal mismatch between waste heat availability and process heat demand and between electricity demand and renewable electricity generation, and the limited capacity of the electricity grid are challenges that delay broad industrial adoption of HTHPs.

This paper proposes a new industrial multi-energy system (MES) with a multi-temperature multi-stage R718 HTHP systematically integrated with thermal energy storage (TES) and photovoltaics (PV). The methodology consists of two stages. Firstly, high energy efficiency and delivered temperature for the industrial MES are ensured at the component level. For this, single-stage centrifugal MVC heat pumps with direct flash evaporation and condensation are arranged in cascade between TES tanks, upgrading the waste heat to provide multi-temperature process heat. Secondly, high operational flexibility of the industrial MES is demonstrated at the system level through an illustrative case of a dairy plant where specific process heat demands (hot water and steam) are given. Accordingly, an appropriate R718 MVC HTHP is designed. A dynamic model for HTHP optimization is developed using thermodynamic relations with CoolProp interface and regression modeling. Using time profiles of the process heat demands and the PV generation as inputs, the model is used to optimize system operation with the goal to minimize the stress on the electricity grid. The results show that a high COP of above 3 (25–35 % higher than conventional closed-cycle HTHP) can be reached with the proposed new HTHP, together with a high share of local renewable electricity to power the HTHP, reduced grid stress, and improved flexibility of the integrated system.

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Selection and/or peer-review under the responsibility of the organizers of the 15th IEA Heat Pump Conference 2026.

Keywords: R718 multi-stage high temperature heat pump; Flash evaporation and condensation; Regression model; Operational optimization; Multi-energy system; Operational flexibility.

1. Introduction

Using High-Temperature Heat Pumps (HTHPs) for electrification of process heat is a promising solution to improve energy efficiency and increase the use of renewables in industry. They represent an efficient and effective technology to upgrade waste heat for process heat generation. For this, they use electrical energy to upgrade waste heat to higher temperatures, making it suitable for reuse in industrial processes. Hereby, especially in process industries such as food and beverages, chemical, textile, paper, and wood, large energy savings can be achieved through the application of HTHPs [1–4]. Additionally, the inherent thermal inertia is beneficial for demand-side management [5].

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In the literature, a broader spectrum of work on HTHPs is already given. A comprehensive overview of HTHP studies can be found in our previous work [5,6]. Frate et al. [7] and Andersen et al. [8] describe the HTHP design process. The HTHP operation in an integrated system has been investigated by Walden et al. [9] (for Brayton HTHP) and Padulles et al. [10] (for R717 single-stage HTHP). To investigate the operation of a component in a wider system with significant time-variability in energy flows, dynamic models are required (Motylinski et al. [11]). For this reason, Mugnini et al. [12] developed a dynamic model of a heat pump representing the COP as a function of sink and source temperature and heat pump operating load using polynomial regression based on manufacturers' data. Furthermore, Walden et al. [9] developed a regression model based on simulation data.

While the mentioned literature indicates that there is a large potential for HTHPs, still several challenges must be addressed to enable their widespread deployment: i) reach the required heat temperature and temperature lift, ii) efficiency, iii) temporal mismatch between waste heat and heat demand, iv) temporal mismatch between renewable generation and electricity consumption, and v) grid congestion. These challenges directly affect the success of HTHP deployments in industrial systems and motivate the need for system-level, optimization-oriented modelling approaches.

This paper proposes a novel framework for the innovation and optimization of HTHP-integrated industrial multi-energy systems (MESs). The main elements of the proposed framework are a concept of an original TES-integrated, open-cycle, multi-stage, R718 HTHP, its dynamic modeling in non-linear algebraic form, and the use of the derived algebraic model in optimization studies. This framework demonstrates improved efficiency and operational flexibility through an illustrative case of a dairy plant. Beyond the dairy industry, the proposed methodology can also be applied in other energy-intensive sectors such as food and beverages, chemicals, textiles, paper, and wood, where process heat occurs across several temperature levels, and where process heat at all temperature levels, as well as low-temperature waste heat, and renewable electricity are highly time-dependent. Such a combination of multi-level high-temperature requirements and strongly time-dependent energy flows poses significant challenges for the design and operation of HTHP-integrated industrial multi-energy systems, which have so far not been adequately addressed in the literature.

2. System description

This section describes the derived novel concept of a multi-stage R718 HTHP and its integration with TES in industrial MESs.

Firstly, the energy flows are described. Waste heat from industrial processes, refrigeration systems, and compressed air stations is collected at 40°C and stored in the first thermal energy storage (TES1). This thermal energy storage is in the form of water sensible heat, with the tank temperature varying between 35°C and 45°C. Process heat is delivered to industrial processes from the thermal energy storage tanks at 40°C (TES1), 67°C (TES2, varying between 64°C and 70°C), and 95°C (TES3, varying between 90°C and 98°C), as well as in the form of steam at 150°C (optionally also at 125°C).

A schematic diagram of the open-cycle, multi-stage R718 HTHP is given in Figure 1. Saturated vapor at state 1 (evaporation temperature 37°C) enters the compressor (C1), where its temperature and pressure rise to state 2. The first part of the flow after C1 is directed to a direct-contact flash condenser (Co1), where it delivers heat at 70°C. The pressure in Co1 is sub-atmospheric and equal to the outlet pressure of compressor C1. In Co1, the refrigerant is condensed (to state 4) by injecting water (at state 3) from the TES2 at a temperature lower than the saturation temperature for the Co1 pressure. Thereby, the water from TES2 heats up from state 3 to state 4, and, using a pump, is returned to TES2 at atmospheric pressure. The refrigerant is throttled from state 4 and enters a direct-contact flash evaporator (E1) at state 5, where it evaporates (to state 1) at 37°C. The pressure in E1 is sub-atmospheric and equal to the compressor C1 inlet pressure. In E1, the refrigerant evaporates by injecting water (at state 6) from the TES1 at a temperature higher than the saturation temperature for the E1 pressure. Thereby, the water from TES1 cools down from state 6 to state 11, and is returned to TES1 at atmospheric pressure using a pump.

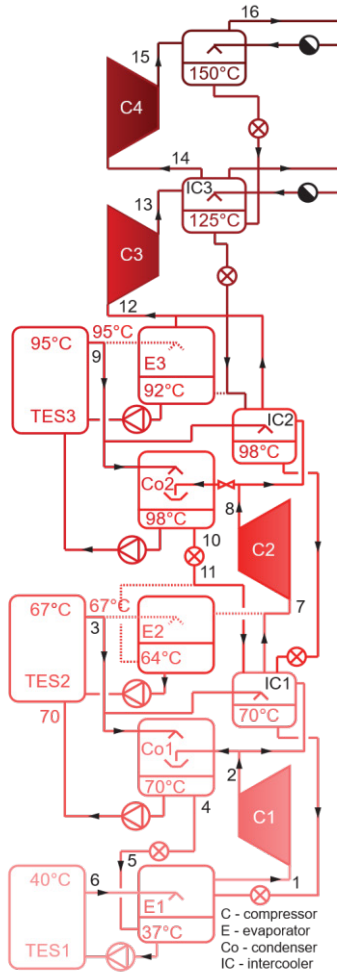


Figure 1 Scheme of a TES-integrated multi-stage R718 HTHP

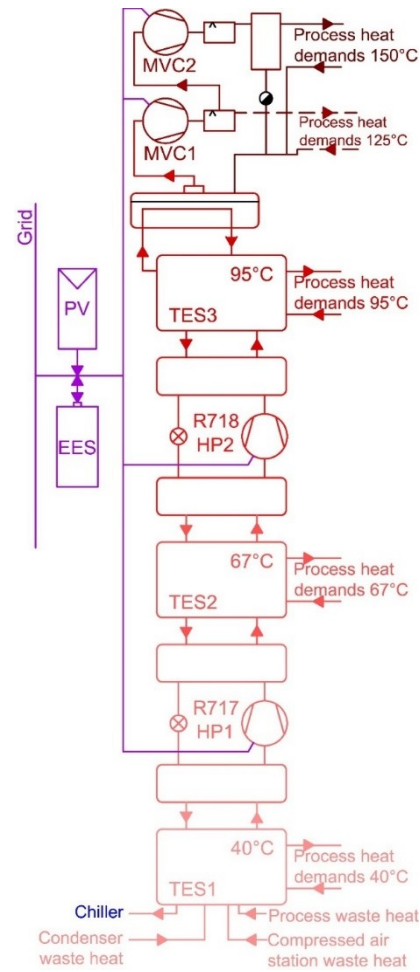


Figure 2 Scheme of a closed-cycle cascade R717/R718 heat pump topped with MVC [5]

The other part of the flow after C1 is intercooled in the intercooler (IC1) to state 7 and directly suctioned by the compressor C2, which further raises the pressure and temperature to state 8. One part of the flow after C2 is directed to the direct-contact flash condenser (Co2), where it delivers heat at 98°C. The other part of the flow after C2 is intercooled in the intercooler (IC2) to state 12 and directly suctioned by the compressor C3, raising its temperature and pressure to state 13. After intercooling in the intercooler (IC3), a final compression happens in the compressor C4 to state 15. High superheating occurs, and hence the temperature is decreased to the saturation temperature at state 16 in the final cooler [13]. The saturated steam at 150°C is delivered to heat consumers.

Optionally, two additional evaporators (E2 and E3) can be integrated, whose main role is to improve the system's flexibility at the cost of slightly increased temperature lift and, consequently, reduced COP. When the temperature in the TES3 reaches its upper limit, C1 and C2 are shut down, and E3 is turned on. When the temperature in the TES2 reaches its upper limit, C1 is shut down and E2 is turned on.

A conventional closed-cycle cascade HTHP is shown in Figure 2 (proposed by Sharevska et al. [5]). The first key difference is that the system proposed in Figure 1 is a multi-stage R718 heat pump, rather than a cascade R717/R718 heat pump topped with a mechanical vapor compression (MVC). The second key difference is that the system in Figure 1 proposes an open-cycle heat pump in which R718 serves as both the TES medium and the refrigerant. The advantages of multi-stage over cascade HTHPs and of open-cycle over closed-cycle HTHPs are described by Sharevska et al. [6].

3. Methodology

This section gives a methodology for modeling and optimization of the proposed new TES-integrated multi-stage R718 HTHP described in Section 2. Firstly, a physics-based model is developed to determine the system's design parameters and to generate training data. Next, surrogate models are developed and validated. Finally, the operation optimization problem is formulated with the surrogate models as constraints and grid stress minimization as an objective. The methodology is applied to an illustrative dairy plant aiming to increase energy efficiency and reduce grid stress (Figure 3).

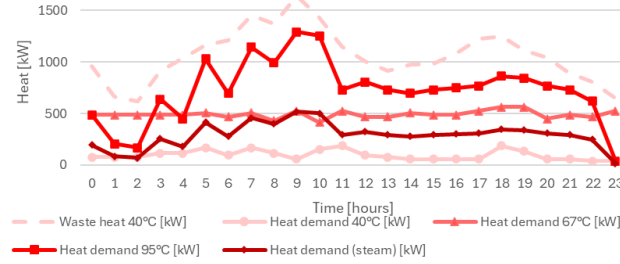


Figure 3 Daily heat demands and waste heat for the dairy industry

3.1. Physics-based model

This subsection describes the physics-based model of the original system. To create a well-defined setting for the presented methodology, the following modelling assumptions are made:

- Compressors, flash evaporators, and flash condensers reach a steady state instantly.
- Saturated vapor exits the evaporator and saturated liquid exits the condenser.
- Adiabatic operation of the compressor is assumed.
- Centrifugal compressor isentropic efficiency in design conditions is 80 %.
- Centrifugal compressor isentropic efficiency in off-design conditions is a quadratic function of the flow rate and pressure ratio, as given in [5].

A model of the original HTHP configuration is developed in MATLAB with a CoolProp interface. The thermodynamic states of the system are defined according to Figure 4 for given temperatures in TES1, TES2, and TES3, given heat demands at 67°C and 95°C, and given steam demand. Based on this, the specific heating effect, specific compression work, specific cooling effect, flow rates, and COP are calculated.

The condenser 1 (Co1) is designed for twice the mean heat demand at 67°C ($Q_{HeatDemand67^\circ C}$) and the flow rates of the refrigerant ($\dot{M}_{Co1}$) and the water ($\dot{M}_{WCo1}$) are given in Eqs. (1-3). Even though both streams are water and interact directly in the direct-contact flash condenser, the model distinguishes between the flow rate that undergoes heating from state from 3 to 4 and the flow rate that undergoes condensation from 2 to 4.

$$q_{Co1} = h_2 - h_4 \quad (1)$$

$$\dot{M}_{Co1} = \frac{Q_{HeatDemand67^\circ C}}{q_{Co1}} \quad (2)$$

$$\dot{M}_{WCo1} = \dot{M}_{Co1} \frac{h_2 - h_4}{h_4 - h_3} \quad (3)$$

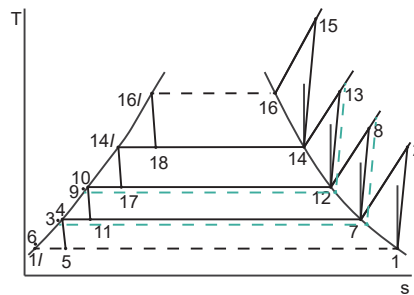


Figure 4 Temperature-entropy diagram of open-cycle multi-stage R718 HTHP

The condenser 2 (Co2) is designed for twice the mean heat demand at 95°C ($Q_{HeatDemand95^\circ C}$) and the flow rates of the refrigerant and the water are given in Eqs. (4-6).

$$q_{Co2} = h_8 - h_{10} \quad (4)$$

$$\dot{M}_{Co2} = \frac{Q_{HeatDemand95^\circ C}}{q_{Co2}} \quad (5)$$

$$\dot{M}_{W_{Co2}} = \dot{M}_{Co2} \frac{h_8 - h_{10}}{h_{10} - h_9} \quad (6)$$

The compressor 4 (C4) is designed for the maximum steam demand ($Q_{SteamDemand}$) and the flow rate is given in Eq. (7).

$$\dot{M}_{C4} = \frac{Q_{SteamDemand}}{h_{15} - h_{16l}} \quad (7)$$

The flow rate through compressor 3 (C3) has to account for the additional injection of liquid (y_3 with h_{14l}) for intercooling between C3 and C4 (from h_{13} to h_{14}) and for the vapor fraction after throttling (x_{18}) in Eq. (8).

$$\dot{M}_{C3} = \frac{\dot{M}_{C4}}{\left(1 + \frac{h_{13} - h_{14}}{h_{14} - h_{14l}}\right)(1 + x_{18})} = \frac{\dot{M}_{C4}}{(1 + y_3)(1 + x_{18})} \quad (8)$$

The flow rate through compressor 2 (C2) consists of a part directed towards Co2 and a part directed towards C3 in Eq. (9). The part directed towards C3 accounts for the liquid for intercooling (y_2) and the vapor fraction after throttling (x_{17}).

$$\dot{M}_{C2} = \dot{M}_{Co2} + \frac{\dot{M}_{C3}}{\left(1 + \frac{h_8 - h_{12}}{h_{12} - h_{10}}\right)(1 + x_{17})} = \dot{M}_{Co2} + \frac{\dot{M}_{C3}}{(1 + y_2)(1 + x_{17})} \quad (9)$$

The flow rate through compressor 1 (C1) consists of a part directed towards Co1 and a part directed towards C2 in Eq. (10). The part directed towards C2 accounts for the liquid for intercooling (y_1) and the vapor fraction after throttling (x_{11}).

$$\dot{M}_{C1} = \dot{M}_{Co1} + \frac{\dot{M}_{C2}}{\left(1 + \frac{h_2 - h_7}{h_7 - h_4}\right)(1 + x_{11})} = \dot{M}_{Co1} + \frac{\dot{M}_{C2}}{(1 + y_1)(1 + x_{11})} \quad (10)$$

The flow rate through evaporator 1 (E1) is the same as the flow rate through C1, whereas the flow rate of water through E1 ($\dot{M}_{W_{E1}}$) is given in Eq. (11). As the design temperature difference between state 6 and state 11 decreases, the flow rate of water increases.

$$\dot{M}_{W_{E1}} = \dot{M}_{C1} \frac{h_1 - h_5}{h_6 - h_{11}} \quad (11)$$

To calculate the total evaporation heat and compression work Eqs. (12-17) are used.

$$l_{C1} = h_2 - h_1 \quad (12)$$

$$q_E = h_1 - h_5 \quad (13)$$

$$l_{C2} = h_8 - h_7 \quad (14)$$

$$Q_{E1} = \dot{M}_{C1} q_E \quad (15)$$

$$P_{C1} = \dot{M}_{C1} l_{C1} \quad (16)$$

$$P_{C2} = \dot{M}_{C2} l_{C2} \quad (17)$$

Because the generated heat is provided at three different temperature levels, the total COP would be an unfair indicator of the performance. Hence, the COP for each stage is determined using Eqs. (18-20) and the total exergy efficiency is calculated by Eq. (21). Also, because the heat source is available at 40°C, the COP for each temperature lift (COP_1 for the lift from 40°C to 67°C, COP_2 for the lift from 40°C to 95°C, and COP_3 for the lift from 40°C to 150°C) is given in Eqs. (19a), and (20a).

$$COP_1 = \frac{h_2 - h_4}{h_2 - h_1} = COP_{1'} \quad (18)$$

$$COP_2 = \frac{h_8 - h_{10}}{h_8 - h_7} \quad (19)$$

$$COP_{2'} = \frac{h_8 - h_{10}}{(h_8 - h_7) + (h_2 - h_1) \frac{1}{(1 + y_1)(1 + x_{11})}} \quad (19a)$$

$$COP_3 = \frac{h_{15} - h_{16l}}{(h_{15} - h_{14}) + (h_{13} - h_{12})} \quad (20)$$

$$COP_{3'} = \frac{h_{15} - h_{16l}}{l_{C4} + \frac{1}{(1+y_3)(1+x_{18})} (l_{C3} + \frac{1}{(1+y_2)(1+x_{17})} (l_{C2} + \frac{1}{(1+y_1)(1+x_{11})} l_{C1}))} \quad (20a)$$

$$\eta_{ex} = \frac{Q_{Co1} \left(1 - \frac{T_0}{T_{TES2}}\right) + Q_{Co2} \left(1 - \frac{T_0}{T_{TES3}}\right) + Q_{SteamDemand} \left(1 - \frac{T_0}{(150 + 273)}\right)}{P_{C1} + P_{C2} + P_{C3} + P_{C4} + Q_{E1} \left(1 - \frac{T_0}{T_{TES1}}\right)} \quad (21)$$

The energy balance equations for the three TES are given in Eqs. (22-24).

$$M_{TES1} c_w \frac{dT_{TES1}}{dt} = Q_{WasteHeat}(t) - Q_{HeatDemand40^\circ C}(t) - Q_{E1}(t) \quad (22)$$

$$M_{TES2} c_w \frac{dT_{TES2}}{dt} = Q_{Co1}(t) - Q_{HeatDemand67^\circ C}(t) \quad (23)$$

$$M_{TES3} c_w \frac{dT_{TES3}}{dt} = Q_{Co2}(t) - Q_{HeatDemand95^\circ C}(t) - Q_{E3}(t) \quad (24)$$

3.2. Surrogate model

To determine the electric power load and other operational parameters in each time step that minimize the grid stress, optimization methods are needed. However, if the model described in the previous subsection is used, this would require complex simulation-based optimization techniques (e.g., [14]). Hence, a different path is chosen for this study. Polynomial surrogate models are developed, with the regression coefficients evaluated via simulations of the physics-based model from the previous subsection. Such polynomial surrogate models are then used as algebraic constraints in a nonlinear optimization problem in the next subsection.

Firstly, data is generated using a design-of-computational-experiments procedure to obtain a representative sample of the design and solution spaces. For thirteen equally spaced levels of 5 operational variables (the temperatures in TES1, TES2, and TES3, flow rate in Co1, and steam demand) the complex physics-based model is run and the corresponding value for the response variables (specific condensation heat in Co1 and Co2, the specific compression work of C1, C2, C3, and C4, and the specific evaporation heat in E1) are calculated. The data is split into training data and validation data.

Next, the training data is used to train polynomial regression models in MATLAB with varying polynomial orders ranging from 1st to 9th order polynomial. Note that not all operational variables are used for each response variable (e.g., T_{TES3} is omitted from the model for specific compression work of C1). The residuals have been visually analysed using residuals-against-fitted-values plots, which indicated some inequality of variance when using the original design matrix. To address this issue, the response variables and the operational variables are rescaled. Concretely, the power was converted from W to kW and the flow rate was converted from kg s^{-1} to g s^{-1} . After this transformation, the residuals appeared to be approximately normally distributed around zero [15]. The obtained polynomial models are validated on the independent validation dataset.

Finally, to identify a suitable polynomial order, the root mean squared error on the training data (RMSE) and on the validation data (RMSE_{val}), and the adjusted R² (which penalizes adding terms that are not helpful) were evaluated and compared (e.g., for specific compression work of C1 given in Figure 5). Additionally, for each response variable, the polynomial order was compared with physical expectations. The results showed a consistent decrease in RMSE and RMSE_{val}, indicating an improved fit. However, beyond the second order, the gain in accuracy of the model on the validation dataset became marginal. Therefore, the second-order polynomials were selected as the final regression model for the subsequent optimization study.

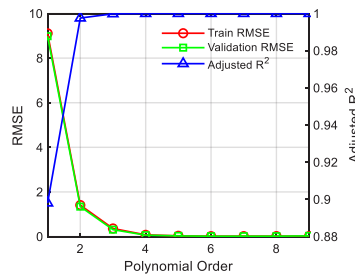


Figure 5 Model selection for the specific compression work of C1

3.3. Optimization formulation

The developed polynomial regression models accurately reproduce the thermodynamic model outputs, as low RMSE on training and validation datasets are achieved. Such a representation of the complex underlying thermodynamic relationships now enables a quick evaluation of the HTHP response under varying operating conditions and makes the regression models suitable for optimization studies on the system operation. For such a study, an optimization methodology is used to determine the proper operating schedule of the proposed system with the aim to minimize the grid stress.

Since peaks in the energy streams are the primary cause of grid congestion, the chosen objective function in Eq. (25) is to minimize the sum of the squared grid power (P_{grid}) over all timesteps [16].

$$f(X) = \sum_{t=1}^N P_{grid}(t)^2 \quad (25)$$

The decision variables are the refrigerant flow rate through the first condenser (Co1) and the On/Off status of the third evaporator (E3) at each timestep.

The HTHP is modelled with the regression models (f_{RM}) in Eqs. (26-32).

$$Q_{Co1} = \dot{M}_{Co1} * q_{Co1} = \dot{M}_{Co1} * f_{RM1}(T_{TES1}, T_{TES2}, \dot{M}_{Co1}, Q_{SteamDemand}) \quad (26)$$

$$Q_{Co2} = \dot{M}_{Co2} * q_{Co2} = \dot{M}_{Co2} * f_{RM2}(T_{TES2}, T_{TES3}, \dot{M}_{Co1}, Q_{SteamDemand}) \quad (27)$$

$$Q_{E1} = \dot{M}_{E1} * q_{E1} = \dot{M}_{E1} * f_{RM3}(T_{TES1}, T_{TES2}) \quad (28)$$

$$P_{C1} = \dot{M}_{C1} * l_{C1} = \dot{M}_{C1} * f_{RM4}(T_{TES1}, T_{TES2}, \dot{M}_{Co1}, Q_{SteamDemand}) \quad (29)$$

$$P_{C2} = \dot{M}_{C2} * l_{C2} = \dot{M}_{C2} * f_{RM5}(T_{TES2}, T_{TES3}, \dot{M}_{Co1}, Q_{SteamDemand}) \quad (30)$$

$$P_{C3} = \dot{M}_{C3} * l_{C3} = \dot{M}_{C3} * f_{RM6}(T_{TES3}, Q_{SteamDemand}) \quad (31)$$

$$P_{C4} = \dot{M}_{C4} * l_{C4} = \dot{M}_{C4}(Q_{SteamDemand}) * l_{C4} \quad (32)$$

The three thermal energy storage tanks are modelled using the energy balance in every time step (Eqs. (33-35)) and over the entire day (Eqs. (36-38)). Moreover, upper and lower bounds are imposed on T_{TES1} , T_{TES2} , and T_{TES3} (given in Section 2).

$$T_{TES1}(t) = T_{TES1}(t-1) + \frac{\Delta t}{M_{TES1}C_w} (Q_{WasteHeat}(t-1) - Q_{HeatDemand40^\circ C}(t-1) - Q_{E1}(t-1)) \quad (33)$$

$$T_{TES2}(t) = T_{TES2}(t-1) + \frac{\Delta t}{M_{TES2}C_w} (Q_{Co1}(t-1) - Q_{HeatDemand67^\circ C}(t-1)) \quad (34)$$

$$T_{TES3}(t) = T_{TES3}(t-1) + \frac{\Delta t}{M_{TES3}C_w} (Q_{Co2}(t-1) - Q_{HeatDemand95^\circ C}(t-1) - Q_{E3}(t-1)) \quad (35)$$

$$\sum_{t=1}^N (Q_{WasteHeat}(t) - Q_{HeatDemand40^\circ C}(t) - Q_{E1}(t)) = 0 \quad (36)$$

$$\sum_{t=1}^N (Q_{Co1}(t) - Q_{HeatDemand67^\circ C}(t)) = 0 \quad (37)$$

$$\sum_{t=1}^N (Q_{Co2}(t) - Q_{HeatDemand95^\circ C}(t) - Q_{E3}(t)) = 0 \quad (38)$$

Finally, the On/Off status of the third evaporator (E3) is modelled by a binary decision variable (X_{E3}). The refrigerant flow rate through the first condenser (Co1) is a semi-continuous decision variable and can take any value between $\dot{M}_{Co1}^{min}$ and $\dot{M}_{Co1}^{max}$ or 0. This can be modelled as inequality in Eqs. (39) and (40).

$$\dot{M}_{Co1}^{min}(1 - X_{E3}(t)) - \dot{M}_{Co1}(t) \leq 0 \quad (39)$$

$$\dot{M}_{Co1}(t) - \dot{M}_{Co1}^{max}(1 - X_{E3}(t)) \leq 0 \quad (40)$$

The resulting nonlinear, non-convex, non-continuous optimization problem is solved in MATLAB using a two-stage hybrid approach. First, a global search is performed with a genetic algorithm (GA). Subsequently, the best GA solution is used as the initial point for the MATLAB *fmincon* solver, employing sequential quadratic programming (SQP) to achieve local convergence. This hybrid approach benefits from the exploration capability of GA and the local convergence capability of SQP. As *fmincon* can only handle

continuous decision variables, a penalty method to relax the binary model is used [17]. The resulting objective function for $fmincon$ is given in Eq. (41).

$$f(X) = \sum_{t=1}^N P_{grid}(t)^2 + \gamma \sum_{t=1}^N X_{E3}(t)(1 - X_{E3}(t)) \quad (41)$$

4. Results

In this section, the efficiency and flexibility of the proposed system are evaluated. For this, the performance in design conditions is presented as well as the dynamic performance over an optimized operation schedule.

The flash evaporators and condensers are an essential feature of the system. They allow for smaller temperature drops in evaporators and smaller temperature rises in condensers compared to conventional heat exchangers, effectively reducing the temperature lift and increasing the COP. This influence of the design temperature drop on the COP is shown in Figure 6. Higher COP is achieved with lower temperature drops. However, the vapor fraction after the flashing process is lower, indicating that higher mass flow rates are required (also shown in the right axis of Figure 6). In this study, the design temperature drop and the design temperature rise for all flash evaporators and flash condensers are set to 3 K. The trade-off between equipment size and efficiency is highlighted by Vieren et al. [18].

The exergy efficiency of the proposed system is 75%, and the COP for each stage is given in Figure 7. The first temperature lift from 40°C to 67°C is achieved with a COP of 7.3. The second stage shows a COP of 9.8. The final stage shows a COP of 5.7. Compared to the system proposed by Sharevska et al. [5], the multi-stage R718 HTHP proposed in Figure 1 shows a higher COP (Figure 7). This is because, firstly, the temperature lift of each stage is reduced due to the replacement of conventional evaporators and condensers with flash evaporators and condensers. Secondly, the refrigerant needed to meet the heat demand at higher temperatures is directly compressed rather than passing through additional condensation and subsequent evaporation processes that introduce irreversibilities in cascade heat pumps. Note that these temperature lifts can be realized with single-stage centrifugal compressors (Sarevski and Sarevski [19]).

Because the heat source is available at 40°C, the COP for each temperature lift is also calculated. The COP_1 for the lift from 40°C to 67°C is 7.3, COP_2 for the lift from 40°C to 95°C is 4.5, and COP_3 for the lift from 40°C to 150°C is 2.9.

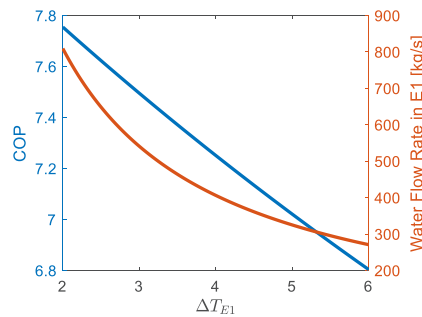


Figure 6 Design temperature drop in evaporator effect on COP and water flow rate

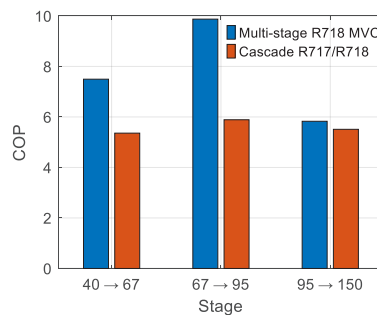


Figure 7 COP for Multi-stage R718 and Cascade R717/R718

The optimized operation over the day is given in Figure 8. Full-load operation coincides with times of renewables generation. This has also been observed in the works by Walden et al. [9] and Padulles et al. [10]. Figure 8 shows the operation schedule for two cases: a) TES₁, TES₂, and TES₃ sizes of 200, 200, 100 tonnes, respectively, and b) TES₁, TES₂, and TES₃ sizes of 250, 250, 150 tonnes, respectively. Note that, the slight drop in load in Figure 8a happens because operating at full power over the entire period of PV availability would violate the constraint for the upper limit of T_{TES3}.

The resulting operation schedule, minimizing the objective function, reduces the mean import of electrical energy and the peaks of imported power. This can be observed from Figure 9a and Figure 9b. The resulting distribution of HTHP power among the four compressors is given in Figure 10.

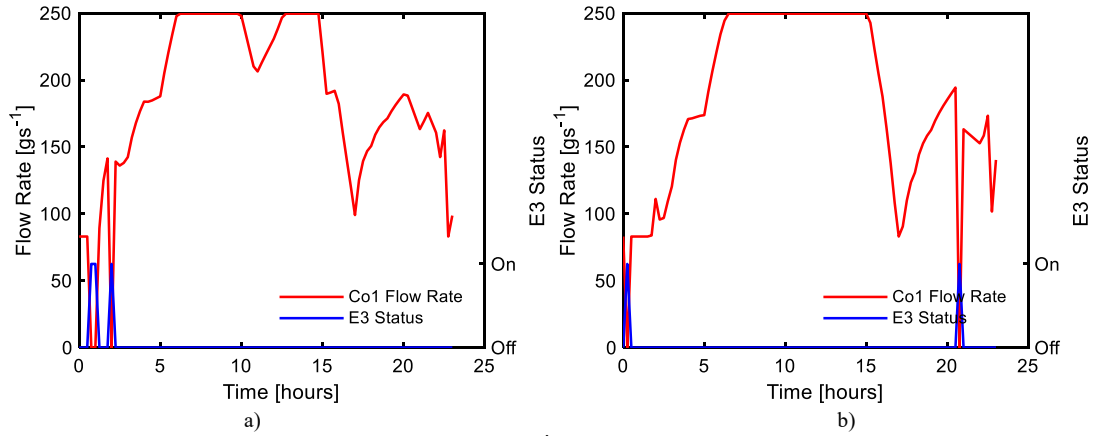


Figure 8 Optimized solution for Co1 flow rate ($\dot{M}_{Co1}$) on left axis and E3 status (X_{E3}) on right axis

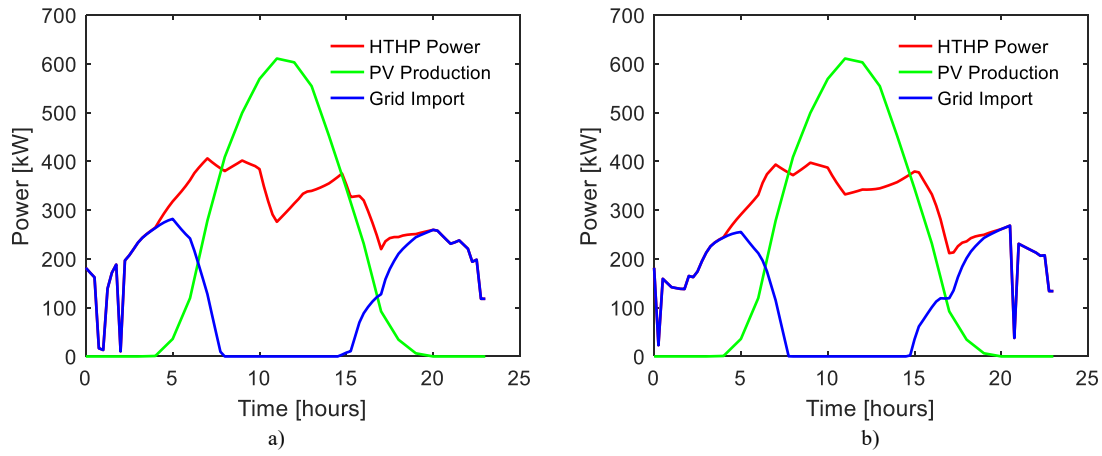


Figure 9 Power profiles for the HTHP, grid import, and PV generation

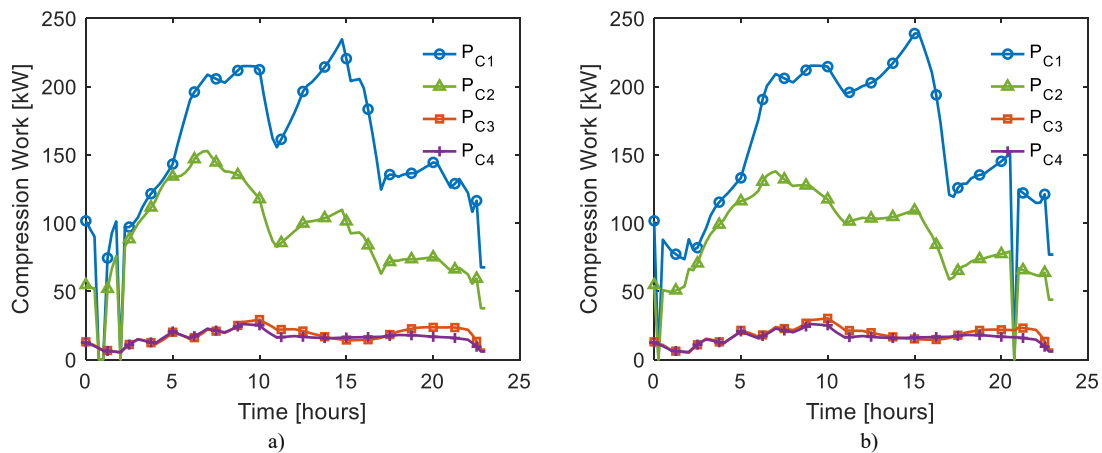


Figure 10 Daily profiles of the compression work

Figure 11 shows that the temperatures of the TESs remain within stable operating range, demonstrating effective thermal coupling between the generated heat (Figure 12) and the heat demand (Figure 3). When the temperature constraints are reached, compressors C1 and C2 are shut down, and no condensation heat is delivered in the TES. However, as the steam demand remains, E3 turns on (Figure 8) to meet this demand. The temperatures of TES₂ and TES₃ rise to store heat during periods of PV generation, while the temperature of TES₁ drops as E1 operates more. This diverging of temperatures increases the temperature lift and although the compressor efficiency is highest, the COP decreases. The resulting COP variation over the day is shown in Figure 13. The figure indicates that the optimization favours efficient operation of the HTHP, meaning that the operating load in each timestep is decided not only based on the PV availability, but also on creating conditions to allow operation across low temperature lift and high compressor efficiency.

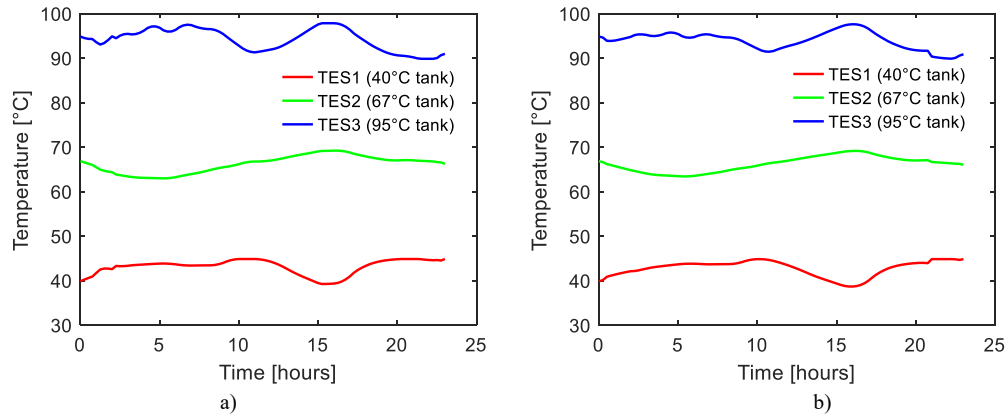


Figure 11 Daily profile of the temperatures in TESs

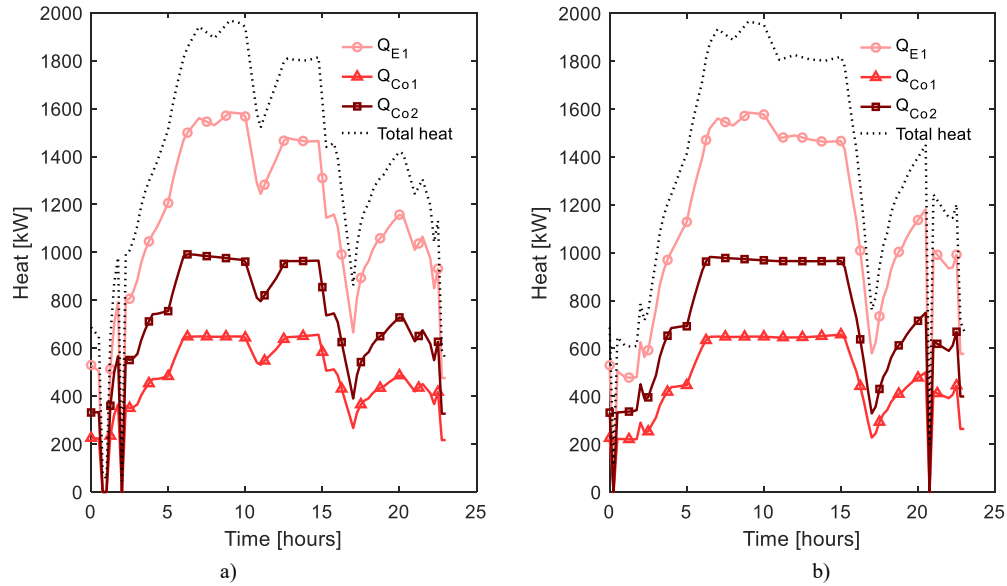


Figure 12 Heat flows in the system

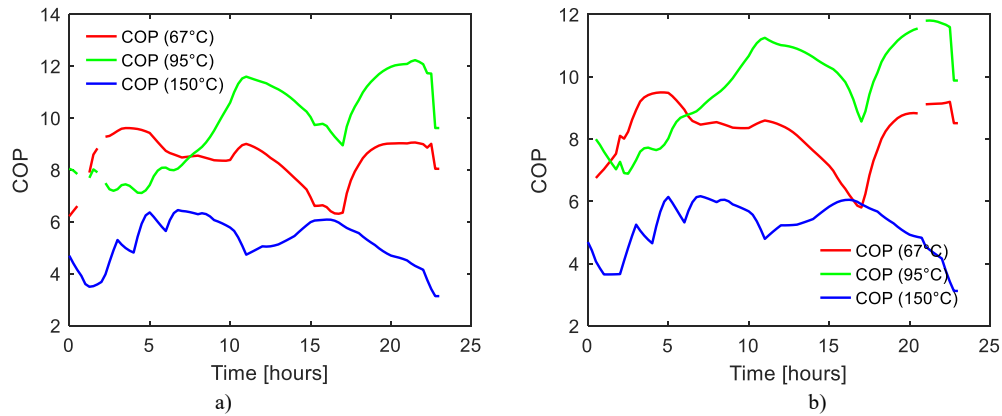


Figure 13 Daily variation of COP

5. Conclusions

This paper proposes a novel framework for the innovation and optimization of HTHP-integrated industrial MESs. An original, open-cycle, multi-stage, multi-temperature, TES-integrated R718 HTHP is proposed as a generalized concept that unifies the roles of the heat storage medium and the refrigerant. Such a concept is shown to improve the efficiency and flexibility of industrial MES based on the results from an original methodology. Firstly, a new physics-based model of this HTHP has been developed. Then, the simulation data is used for developing regression surrogate models that enable fast optimization. Finally, the polynomial regression models are used in optimization of the operation, whereby a genetic algorithm is used for global exploration, followed by a sequential quadratic programming for local refinement.

The results show that the exergy efficiency of the proposed system is 75%, whereas the COP for the first stage (from 40°C to 67°C) is 7.3, for the second stage (from 67°C to 95°C) is 9.8, and for the final stage (from 95°C to 150°C) is 5.7. Moreover, using the methodology an optimized operation is found that successfully couples the electricity and different temperature heat flows and minimizes the grid stress. Hence, the proposed framework offers a promising pathway towards decarbonized industrial energy systems.

Nomenclature

c	specific heat capacity	(J kg ⁻¹ K ⁻¹)
COP	coefficient of performance	(–)
h	specific enthalpy	(J kg ⁻¹)
l	specific compression work	(J kg ⁻¹)
M	mass	(kg)
$\dot{M}$	mass flow rate	(kg s ⁻¹ , g s ⁻¹)
p	pressure	(Pa)
P	power	(W, kW)
Q	heat	(W, kW)
q_{Co}	specific heating effect	(J kg ⁻¹)
q_E	specific refrigeration effect	(J kg ⁻¹)
s	specific entropy	(J kg ⁻¹ K ⁻¹)
T	temperature	(°C, K)
x	quality	(–)
X	On/ Off status	(–)
t	time	(s, h)
T_c	condensation temperature	(°C, K)
T_e	evaporation temperature	(°C, K)
T_0	ambient temperature	(°C, K)
η	compressor efficiency	(–)
η_{ex}	exergy efficiency	(–)
Π	compressor pressure ratio	(–)

Acknowledgements

This research is conducted within the EIGEN (Energy Hubs voor Inpassing van Grootschalige Hernieuwbare Energie) project, subsidised by the Netherlands Enterprise Agency, Dutch Ministry EZK, Netherlands, under Grant Number MOOI 52101.

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