MODELING AND OPTIMIZATION OF A HEAT RECOVERY SYSTEM APPLIED TO A DIESEL ENGINE

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Abstract. *The purpose of this study is to evaluate the capacity of the proposed system to recover energy from the exhaust gases of a diesel engine, this by modeling the system composed by a turbocharger, a heat exchanger and an air turbine; the system was modeled with the boost pressure and the heat exchanger mass flow as its variables, the maximum efficiency point is selected as the parameters to optimize the heat exchanger using a bacterial chemotaxis algorithm to minimize pressure drops and heat exchanger volume and maximize overall heat transfer coefficient, the selected heat exchanger was then introduced in the system model to have a more reliable response of the efficiency and recovery capabilities.*

Keywords*: Diesel exhaust heat recovery, diesel engine model, bacterial chemotaxis algorithm, bioinspired optimization.*

1. INTRODUCTION

The diesel engine is one of the most used system for power generation in small electric plants and in transport systems, so it becomes a very sensitive problem to maximize its power production efficiency (Abusoglu and Kanoglu, 2009), the most common way to increase the power that is generated is introducing more air fuel mixture into the engine so that it produces more power (Durgun and Sahin, 2009). This is achieved when the energy of the exhaust gases drives a turbine which is used to compress the intake air so more fuel can be burned inside the combustion chamber (Hugues *et al*, 2009). A not so common way to increase efficiency is to recover exhaust gases heat energy (Giakoumis, 2007)(Yang *et al*, 2003), this can be done by transferring the heat from the gases to a medium (Ibrahima *et al*, 2008) in a heat exchanger and transforming heat from that medium (Kim and Favratb, 2009) into a useful form of energy (Hountalas, 2008).

A system that combines both recovery methods is modeled and optimized to find its maximum recovery capability; a phenomenological model is used for the engine and other components, in which the effect of the recovery systems over the engine is given by a number of pressure rises at the engine exhaust which after a certain point begins to reduce the engine efficiency; and a bio inspired chemotaxis algorithm is used to find the group of optimized solutions for the heat exchanger objective functions of pressure drops, heat transfer global coefficient and heat exchanger approximate volume.

2. MODELS AND PROCEDURE

The proposed recovery system as seen in Fig.1 is composed by a diesel engine, a heat exchanger, a turbocharger and an air motor, air is introduced by a compressor in the system. Part of this mass flow goes to the engine and the other part is directed to the heat exchanger, afterwards the air is burned in the engine and the exhaust gases (Parlak *et al*, 2005) are introduced into the heat exchanger where part of its heat is transferred to the air (Huang *et al*, 2005)(Talbi and Agnew, 2002), and then passes through the turbine which drives the compressor, afterwards the heated air is introduced into the air motor (Huang *et al*, 2005) where it expands and produces power.

Since the heat exchanger and turbine represent a restriction to the expansion of gases inside the engine, it reduces its efficiency (Heywood, 1989), the purpose then is to maximize the global efficiency, taking the compressor output pressure and the heat exchanger mass flow as the design variables, and the tubes' diameters and quantity as the design variables for the heat exchanger.

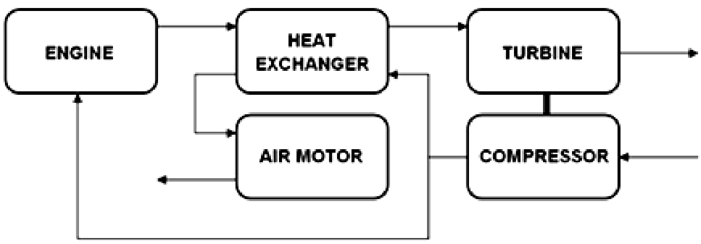

Figure 1. Proposed system diagram.

A phenomenological model was used for the engine, which only depends on the theoretical behavior points (Çengel and Boles, 2001); a correction with normal efficiency for every process in the cycle was taken to achieve more accurate results. The expansion pressure and temperature are found using the pressure drops and pressure used to drive the recovery system Eq.(1) where $Pmul$ is the exhaust multiple pressure, $\eta comp$ is the compressor efficiency, $\eta turb$ is the turbine efficiency, $mgas$ is the exhaust gas mass flow, $mcomp$ is the compressor mass flow, $Tamb$ is the ambient temperature, $Poucomp$ is the compressor output pressure, $Pouturb$ is the turbine output pressure, $kexp$ and $kcomp$ are the specific heat relations.

If the sum of pressure is above the engine free expansion pressure Eq.(2) where $Pcomb$ and $Padm$ are the combustion and ambient pressure respectively and $Tcomb$ is the combustion temperature; then it is taken as the expansion pressure and the temperature is calculated, but if the expansion pressure is above the pressure needed in the turbine plus the pressure drops across the system as shown in Fig. 2., expansion pressure and temperature are equal to the ones for free expansion (Eastop and McConkey, 1993)

$$Pinturb = \left(\left(Pmul \times \eta comp \times \eta turb \times mgas \times Cp \times Tesc/mcomp \times Cp \times Tamb \times \left(1 - \left(\frac{Poucomp}{Pouturb}\right)^{\frac{kexp-1}{kexp}}\right)\right) + 1\right)^{\frac{kcomp}{kcomp-1}} \quad (1)$$

$$Pesclibre = Pcomb \times \left(\frac{Padm \times Tcomb}{Pcomb \times Tadm}\right)^{kexp} - Pressure\ drops\ through\ valves \quad (2)$$

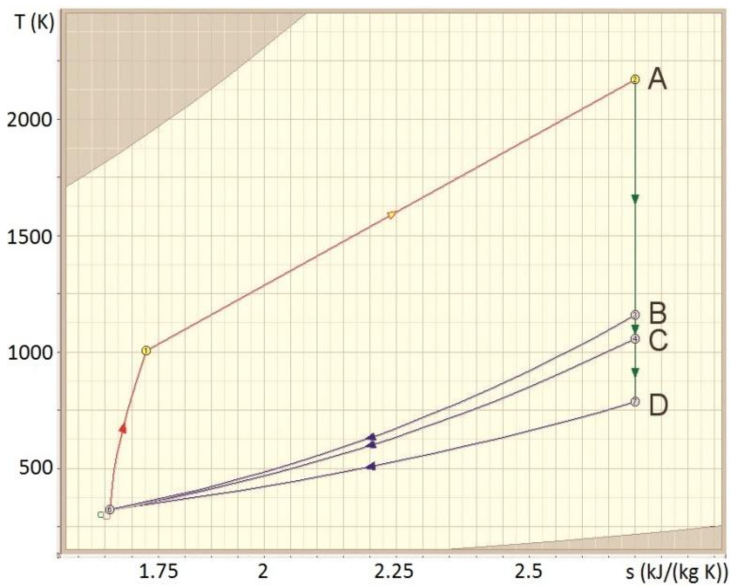

Figure 2. Motor cycle T-s diagram. A: Combustion pressure. B: Free expansion pressure. C: Turbine needed pressure. D: Ambient pressure.

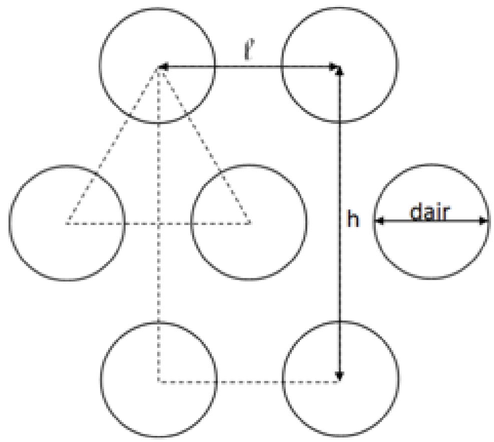

Figure 3. Heat exchanger distribution.

The global efficiency is found for every point using Eq.(3) where ηmec is the fixed mechanical efficiency, $mair$ is the air mass flow that enters the engine, $Texp$ is the expansion temperature, $mgas$ is the gas mass flow that expands inside the engine, $mfuel$ is the fuel mass flow, $Wmot$ is the work that is generated by the air motor and is added to the work generated by the engine, and divided by the net inlet heat.

$$\eta glob = \frac{\eta mec \times (mair \times Cp \times (Texp - Tcomb) + mgas \times Cp \times (Tcomp - Tadm)) + Wmot}{mfuel \times LHV} \quad (3)$$

After the residual gas mass (Rakopoulos *et al*, 2004) is stable, the global efficiency maximum and the value of compressor output pressure is also determined, as well as mass flows through the heat exchanger, temperature at the exhaust and temperature of the intake air for the heat exchanger. With these values the heat exchanger can be designed (Marín and Mestizo, 2009), a counter-flow heat exchanger type was selected for the design.

For the heat exchanger optimization three objective functions were taken, the global heat transfer coefficient U to maximize Eq.(4), in function of heat transfer coefficient h for air and gas (Çengel, 2007), the sum of the pressure drop Eq.(5) in the air and gas side to minimize, where pressure drops are calculated by the Darcy-Weisbach equation and friction factor is calculated by the Colebrook equation, and the approximate total heat exchanger volume Eq.(6) to minimize; and three variables were taken to calculate the objective functions, the air tubes diameter, the gas tubes equivalent diameter Eq.(7), and the number of tubes Fig. 3.

$$Max\left(U = 1/\left(\frac{1}{hgas} + \frac{1}{hair}\right)\right) \tag{4}$$

$$Min\left(dP = dPgas + dPair\right) \tag{5}$$

$$Min\left(Vol = \frac{\sqrt{3}}{2}\left(2 \times ntubes \times \pi \times \frac{(dgas^2 - dair^2)}{\sqrt{3}}\right) \times L\right) \tag{6}$$

$$dgas = \sqrt{\left(4 \times \left(\frac{\sqrt{3}}{2} \times l^2 - ntubos \times \pi \times \frac{dair^2}{4}\right)\right)/ntubos \times \pi} \tag{7}$$

The optimization problem was solved by a bioinspired bacterial chemotaxis based multiobjective algorithm (Guzmán *et al*, 2010), in which a number of bacteria that represent the candidate solutions are randomly positioned in the variable space, for every bacterium the objective functions are evaluated in a heat exchanger model in which simple heat transfer equations were used to calculate the necessary variables, then the bacteria apply simple chemotactic rules of movement in order to explore and explode the search space.

Applying the fast non-dominated sorting procedure (Deb et al. 2002), bacteria are classified as strong (non-dominated) and weak bacteria (dominated). By comparing their previous position with the current, bacteria decide the next movement based on non-dominance criterion, looking for positions in the search space that represent non dominated candidate solutions, so a Pareto Optimal Front can be found.

3. RESULTS

After the optimization algorithm has reach the pareto front, a discretization of every solution variables is done by changing the variable value for the closest one in a set of discrete values (Belegundu and Chandrupatla, 1999), and again objective functions are evaluated for the new variable discrete value Fig. 4. Then a solution is selected for the heat exchanger, this solution can be randomly selected from the pareto front, so a closest to the origin selection criteria was programmed, thus the air tubes diameter, gas tubes equivalent diameter and number of tubes are selected automatically and can now be introduced into the real model and calculate a more reliable response to the recovery capability of the system, including the pressure drop and heat exchanging efficiency.

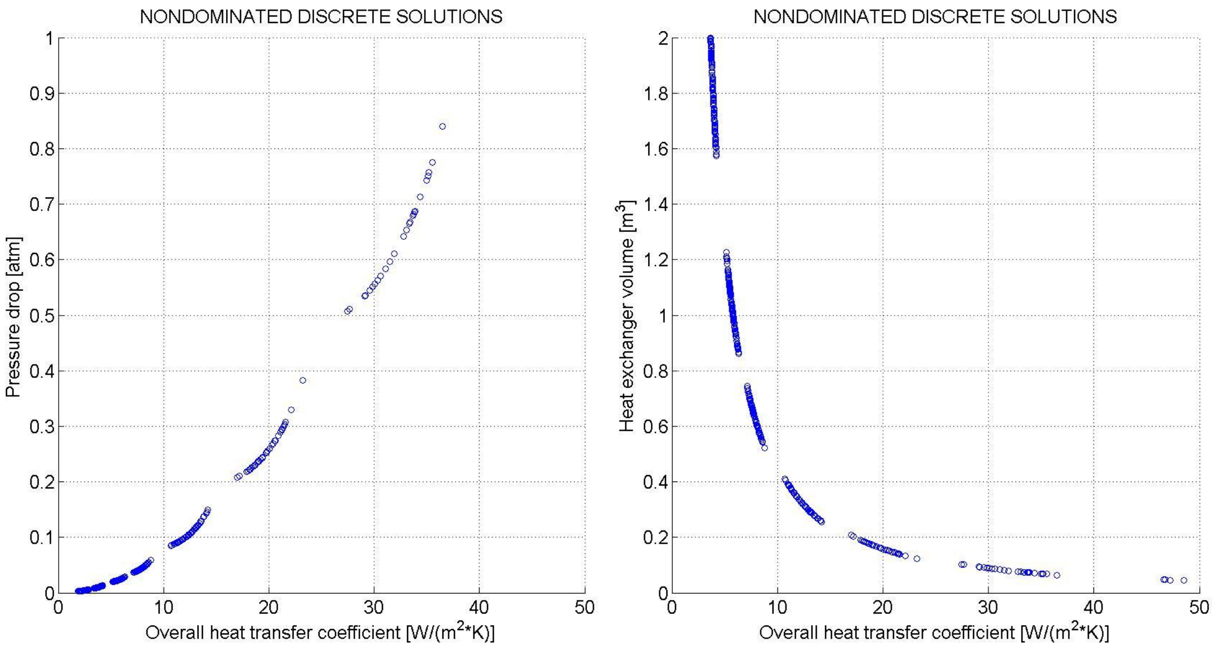

Figure 4. Discrete Pareto front. As seen distribution is not continuous neither a surface as expected for three variables and objective functions.

The global efficiency depends on the heat exchanger selection, this is very important because large pressure drops can occur across this component due to its variable selection (Sánchez-Horneros, 2009), the optimization algorithm gives the group of non dominated solutions, from which a unique solution must be chosen, but as the objective functions are not always in a permissible range only one of the most representative ones must be selected, as a common method to select a stable solution the closest to the origin is selected, this method was implemented even knowing that every axis has different units, this due to a lack of a better way to select a unique answer.

The same procedure of the ideal model is used for the real model, but in this point a pressure drop and a heat exchanger effectiveness is calculated for every point of the variable analysis space and again the iterative process for the residual gas mass is done until its value is stable. With the residual gas mass value stable, the engine and global real efficiency is found as is shown in Fig. 5, and the values of the compressor output pressure and heat exchanger mass flow in the maximum efficiency point can be easily computed.

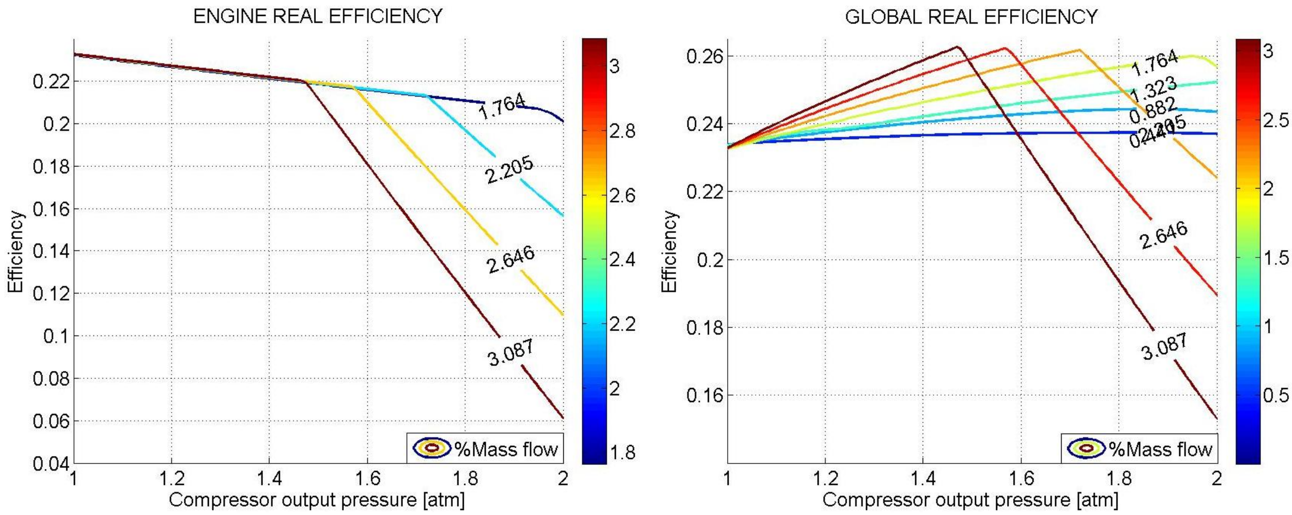

Figure 5. Real efficiency results. Engine efficiency decreases as global efficiency increases due to pressure drops induced by the recovery system on the motor. Global efficiency for every % mass flow increases to a maximum and then decreases. % Mass flow represent the relation between heat exchanger mass flow and engine mass flow.

The maximum global efficiency increase for the ideal model was 3.188% with a 1.614% engine efficiency decrease; these values were found at 224% the mass flow to the engine, and at 1.693 atm. The maximum global efficiency increase for the real model was 2.865% with a 1.231% engine efficiency decrease, this values were found at 350% the mass flow to the engine, and 1.406 atm of compressor output pressure, this point has a much larger value than the maximum point for the ideal model, and has the greater efficiency increase for the real model having a lower engine efficiency decrease than the ideal model maximum point.

4. DISCUSSION

In both the ideal and real model results the global efficiency tends to increase with the pressure and with the mass percentage (Fig. 6), when the compressor output pressure grows above the free expansion pressure the global efficiency starts to decrease rapidly; the real system model efficiency lines for fixed mass flow percentage starts with smaller slope than in the ideal model, and the point of free expansion pressure is reached at lower intake pressures due to pressure drops across the system. The engine efficiency tendency for both ideal and real models is decreasing from the start Fig. 7, with the real one having a higher negative slope than the ideal model also due to the pressure drops across the system, and the same as in the global efficiency when the compressor output value reaches the free expansion pressure the efficiency starts to decrease rapidly until it becomes zero.

With this study it was proved that in a purely theoretical way it is possible to combine both recovery ways to increase the global efficiency even when reducing the motor efficiency itself, as the maximum real efficiency was found in the top mass flow percentage, the maximum efficiency is constrained with the maximum compressor flow capacity; when the mass flow percentage increases power produced and efficiency also increases which gives an advantage to the hybrid system over using a single turbocharger system.

The optimization of the heat exchanger with and without the discretization on the last step gives a solution for the three objective functions as shown in Fig. 6 and 7 where a line instead of a surface shows that for a given variable the optimization of the functions occurs when it reaches one of its constraints, as seen in the results the variable that always reaches its upper limit is the number tubes inside the heat exchanger, when the upper limit of the number of tubes is raised the bacterial number of tubes variable value keeps reaching the limit for all solutions in the pareto front.

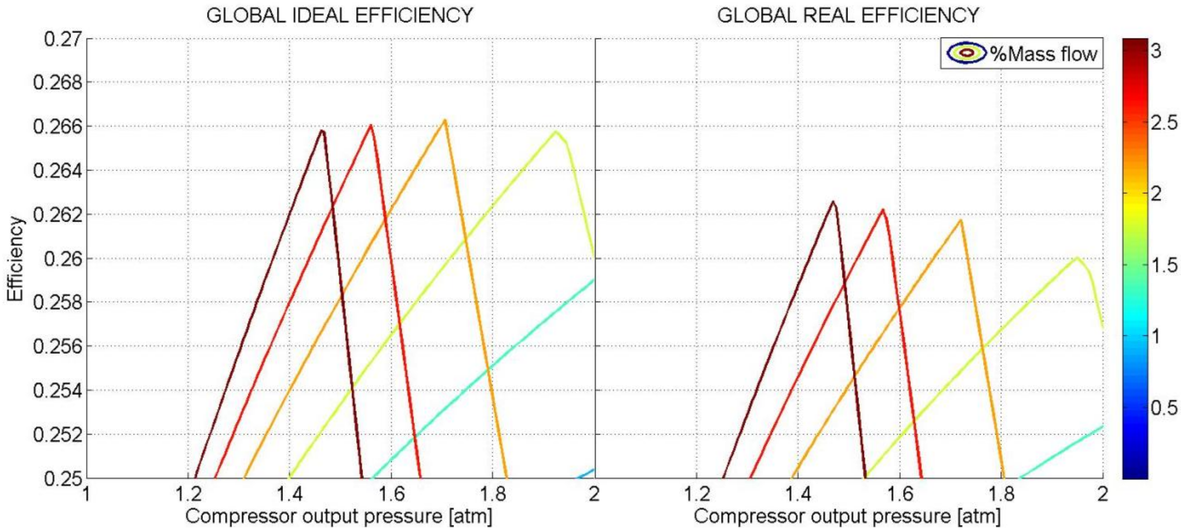

Figure 6. Global efficiency comparison. Efficiency decreases due to pressure drops across heat exchanger. % Mass flow represent the relation between heat exchanger mass flow and engine mass flow.

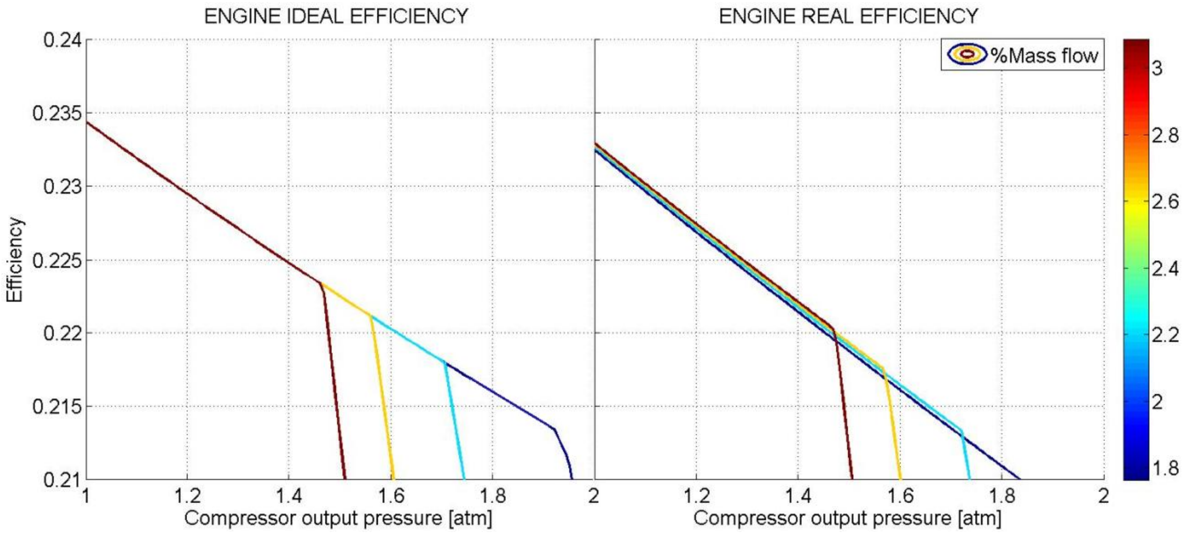

Figure7. Engine efficiency comparison. Efficiency decreases due to pressure drops across heat exchanger. % Mass flow represent the relation between heat exchanger mass flow and engine mass flow.

After the real model maximum efficiency point has been calculated a new heat exchanger should be selected and the maximum efficiency recalculated, and this as for the residual gas percentage should be done iteratively until the maximum efficiency point and its value stops changing between the two last iteration, this should be evaluated with a minimum convergence tolerance which is the maximum allowable difference between those last iteration values.

Despite the efficiency increase achieved with the model, with a more accurate model results may change and efficiency increase may not be enough to consider the real system implementation so further studies may apply a more complete model to know with a better approximation the real system recovery capabilities.

5. CONCLUSIONS

As seen in the results the maximum global real efficiency was found in the top mass flow percentage, and a tendency of every mass flow percentage line peak efficiency to increase with the flow percentage and to move towards a lower compressor output pressure, states that the maximum global efficiency point is restricted by the compressor mass flow capacity and that with a higher percentage of mass flow the maximum efficiency should be reached at a lower compressor output pressure.

The optimization process gives some heat exchanger parameters that applied to the engine model for the variable design space results on a different maximum efficiency point, this implies that a heat exchanger parameter selection must be done for every point of the engine variable space, this to assure that every point was the best parameter selection and the maximum efficiency possible for it, or an iterative heat exchanger parameter selection could be done until maximum efficiency point and value stop changing between the two last iterations.

A theoretical recovery capacity was proven for the proposed system, it has an advantage over the single turbo compressor because its efficiency rise along with the power generated. A more exact model of the engine behavior should be implemented for a more reliable results of the real recovering capacity of the system, along with a dedicated discrete optimization algorithm to compare the heat exchanger parameter selection and its effect on the recovery capacity.

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