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Cold Climates Heat Pump Design Optimization

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Abstract

Heat pumps provide an efficient heating method; however they suffer from severe capacity and performance degradation at low ambient conditions. This has deterred market penetration in cold climates. There is a continuing effort to find an efficient air source cold climate heat pump that maintains acceptable capacity and performance at low ambient conditions. Systematic optimization techniques provide a reliable approach for the design of such systems. This paper presents a step-by-step approach for the design optimization of cold climate heat pumps. We first start by describing the optimization problem: objective function, constraints, and design space. Then we illustrate how to perform this design optimization using an open source publically available optimization toolbox. The response of the heat pump design was evaluated using a validated component based vapor compression model. This model was treated as a black box model within the optimization framework. Optimum designs for different system configurations are presented. These optimum results were further analyzed to understand the performance tradeoff and selection criteria. The paper ends with a discussion on the use of systematic optimization for cold climate heat pump design.

INTRODUCTION

Air-source heat pumps (ASHPs) provide efficient heating by augmenting their energy consumption with heat collected from the ambient air that is “pumped” to the required supply temperature. Reverse cycle air-conditioners were presented in the 1930’s as means to efficiently provide heating in buildings (Kerr Jr. et al. 1934, Neeson 1938, Brace and Crawford 1938, Labberton 1939). However, these systems were not introduced to the market before the 1950’s and only started to be reliable and economically feasible in the 1960’s as described by Hiller (1976). This industry received increased interest following the 1973 oil embargo prompting significant research to improve performance, particularly for cold climate locations.

In late 1975 Carrier Corporation initiated an extensive heat pump research effort (Groff and Reedy 1978 and Groff et al. 1979). Four residential split-system ASHPs based on the vapor-compression (VC) refrigeration cycle located in Seattle, Minneapolis, Syracuse, and Boston were instrumented and monitored for a full year. The field tests illustrated that these heat pumps achieved significant energy saving compared to electric resistance heating systems. Additional studies pointed out the benefits of increasing heat pump capacity for colder climate locations despite negative impacts on cooling season performance (Groff et al. 1978 and Bullock et al. 1980). The average efficiency of residential heat pumps sold in USA increased 2.5% per year in 1980s (Calm 1987). In 1995, EPA introduced Energy Star specifications for residential heating and cooling products, including ASHPs. Today’s Energy Star label is only awarded to ASHPs with a Cooling Seasonal Coefficient of Performance (COP_s) of 4.25 or higher (SEER of >14.5) and Heating Season COP (COP_{HS}) of 2.4 or higher (HSPF of >8.19). The current most efficient air-source heat pumps have COP_s of 5.9 (SEER >20.13) while the COP_s of a heat pump in 1979 was just 2.1 (EERs = 7.17). Thus the energy efficiency of modern air-source heat pumps is almost three times higher than those available 30 years ago. This great efficiency achievement has resulted from technical advances in vapor

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compression systems and components (e.g. compressors, heat exchangers, flow control devices, etc.) as well as microprocessor-based control, variable-speed motors, etc., all achieved while making the switch from ozone depleting refrigerants to HFCs (Karen and Herold 1993). However, several issues still negatively impact ASHP heating performance under cold ambient conditions (Roth et al. 2009). First, ASHP heating capacity and COP significantly decrease as ambient temperature decreases. Second, ASHPs have the drawback of accumulating frost on outdoor coils requiring periodic defrosting (usually by temporarily reversing the VC cycle refrigerant flow), both of which degrade energy efficiency and thermal comfort.

Over the last several decades, a number of technologies and design modifications have been proposed to improve the COP and heating capacity of cold climate heat pumps. Homes and buildings in cold climates usually require higher space-heating design loads than space-cooling design loads. In an effort to increase ASHP energy efficiency in cold climates, US manufacturers are gradually introducing new products specifically designed for better cold weather performance (Hadely et al. 2006 and PTR 2007). These new products use a combination of innovative technologies coordinated by the control systems to enhance their performance. The strategy of the multi-stage vapor injection compression cycle (with multiple compression stages) is becoming attractive to improve the COP and heating capacity of heat pumps at cold operating conditions. Previous theoretical and experimental results reported that multi-stage vapor injection compression cycles achieved higher COP and capacity than single-stage cycles at cold ambient conditions (Domanski 1996, Bertsch and Groll 2008, Wang et al. 2009, and Mathison et al. 2011).

In this paper, engineering optimization is used to optimize the performance of a cold climate heat pump based on currently available heat pump products with minor system and heat exchanger modifications. These modifications include using a vapor injected compressor with either a flash tank or an intermediate pressure economizer as well as varying the fin density on the indoor and outdoor heat exchangers. Furthermore, optimization is used to maximize the efficiency and satisfy minimum acceptable capacity at low ambient conditions of -26.11°C (-15°F).

METHODOLOGY

Engineering optimization is a powerful design tool. It provides a systematic way of finding a global optimum design. In this paper we are using optimization in its rigorous definition (Radermacher and Abdelaziz 2008). In this paper, GenOpt®[1], a publicly available optimization tool box is used to optimize the design of cold climate vapor injected heat pump systems. GenOpt® is an optimization program for the minimization of a cost function evaluated by an external simulation program (Wetter 2009). In this paper, we use an in-house validated vapor compression system simulation tool[2] to evaluate the performance of the cold climate heat pump. A wrapper was developed to provide seamless coupling between GenOpt® and the in-house system simulation tool. In this case, the wrapper, shown in Figure 1, generates the required script files to run GenOpt® and the cold climate heat pump model based on the problem definition. The cold climate heat pump model is treated as a black box simulation tool that runs in batch mode. The following subsections include brief description of engineering optimization and cold climate heat pump modeling.

Engineering Optimization

Engineering optimization problems involve *at least one* objective function that is either minimized or maximized. Examples include maximizing energy efficiency, capacity, reliability, or minimizing cost, energy consumption, materials, leaks, etc. The result is a global optimum design that satisfies all the problem constraint. Optimization problems are usually subject to equality and inequality constraints within a specified design space as shown in Equation (1). In Equation (1), f refers to the objective function of the optimization problem, x is a vector of d design variables, g_i represents the i^{th} inequality constraint, I is the number of inequality constraints, h_j is the j^{th} equality constraints and J is the number of equality constraints. The vectors x_k^L and x_k^U indicate the lower and the upper bounds of the variable x_k respectively. In common

[1] http://simulationresearch.lbl.gov/GO/
[2] http://www.ornl.gov/~wlj/hpdm/MarkVII.shtml

engineering problems the equality constraints are replaced with inequality constraints representing the acceptable upper and lower bounds of the constrained value.

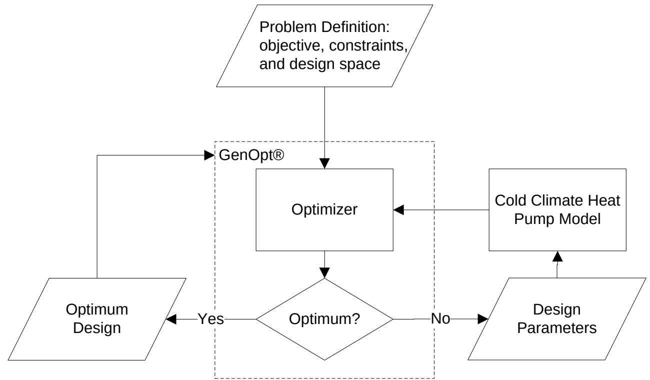


Figure 1: Cold Climate Heat Pump Optimization Flow Chart

$$\begin{aligned} &\underset{x}{\text{minimize}} \quad f(x) \\ &\text{subject to } g_i(x) \le 0 \qquad i = 1,....., I \\ &\qquad\qquad\quad h_j(x) = 0 \qquad j = 1,....., J \\ &\qquad\qquad\quad x_k^L \le x_k \le x_k^U \quad k = 1,....., d \end{aligned} \tag{1}$$

In this paper we used GenOpt® to perform the optimization. GenOpt® has 6 built-in optimization algorithms and 2 parametric study algorithms. Available optimization algorithms are: Generalized Pattern Search algorithm (Hooke-Jeeves and Coordinate Search algorithm), Particle Swarm Optimization algorithm, hybrid global optimization algorithm (Particle Swarm Optimization for the global optimization, and Hooke-Jeeves for the local optimization, Discrete Armijo Gradient algorithm, Nelder and Mead's Simplex algorithm, and Golden Section and Fibonacci algorithms for one-dimensional minimization. Parametric studies algorithms in GenOpt® can be done either using mesh generator to evaluate the function on all points that belong to a mesh with equidistant or logarithmic spacing between the mesh points or parametric search where only one independent variable is varied at a time. More details about these algorithms are discussed by Wetter (2009). GenOpt® allows the coupling of any simulation program with built-in optimization algorithms. This coupling is achieved by properly setting the optimization problem using the configuration file. GenOpt® communicates with simulation programs using text files. GenOpt® automatically generates input files for the simulation program based on predefined templates that include keywords describing the problem variables. The configuration file includes all the information regarding the simulation program: how it can be started and how to retrieve the current value of the cost function. As such, GenOpt® can be used with any simulation program that can read its input from text files and write the cost function value plus any possible error messages to a text file. In the case of constrained optimization as is the case with most engineering applications, Genopt® provides a default algorithm for box-constraints. Users can also provide their own penalty or barrier functions as part of the problem definition.

Cold Climate Heat Pump Modeling

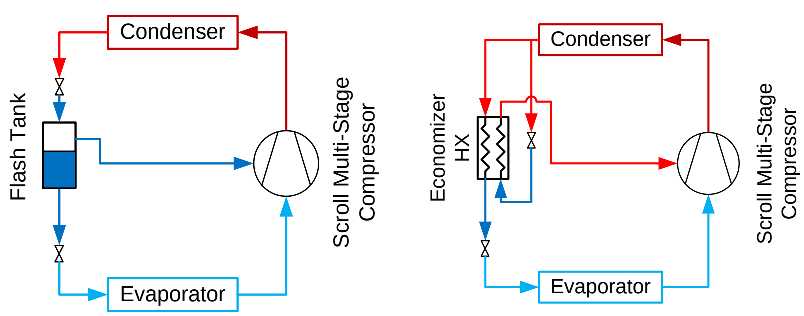
A research tool for steady-state simulation and design analysis of air-to-air heat pumps and air conditioners (Rice 2001) was used for the heat pump system modeling described in this paper. The vapor-compression cycle is modeled with empirical representations for compressor performance based on a 10 coefficient performance maps (as described in ANSI/AHRI

Standard 540, [AHRI 2004]) and first-principle region-by-region modeling of the heat exchangers. Recent advances in this model allow for flexible system simulation. The model is a component based simulation tool with a Newton-Raphson non-linear solver. Different component models are being used in this paper. A segmented Heat Exchanger (HX) model is used to model the evaporator and condenser fin-&-tube coils. A modified compressor map is used to model the multi-stage compressor. An overall UA/effectiveness HX model is used to model the refrigerant-to-refrigerant economizer. Finally, a simple flash tank model is used to evaluate the injection flow rate based on phase separation.

Multi-stage vapor injection compression cycle can be classified into two fundamental configurations: (a) Flash tank cycle and (b) Economizing heat exchanger cycle. Figure 2 shows the schematics of a 2-stage cycle for each configuration. In a two-stage cycle with flash tank, two-phase refrigerant is separated into saturated liquid and vapor by a flash tank after the first expansion. It has the advantage of feeding 100% of saturated vapor to the compressor injection port. However, the amount of refrigerant going to the injection port is difficult to control and is solely determined by the high side pressure.

The two-stage cycle with economizing heat exchanger allows part of the liquid refrigerant at the condenser outlet to pass through an expansion valve before entering the economizer HX to further subcool the mainstream refrigerant coming from the condenser. The superheated intermediate pressure refrigerant leaving the economizer HX enters the intermediate compressor port. As a result, the separation with economizer HX will never be 100% as compared to the flash tank separation due to the limited surface area involved. In the meantime, the subcooled main-stream refrigerant is expanded by a second expansion valve, and then enters the evaporator. The refrigerant flow rate and pressure entering the intermediate compressor port can be easily controlled using thermostatic expansion valves since there exist enough superheat to operate the thermostatic expansion valve. As such, this two-stage cycle has been widely investigated. Wang *et al.* (2009) demonstrated that two-stage cycle with economizer HX achieves performance improvement comparable to that of two-stage cycles with flash tank. The former has a wider operating range of injection pressure due to its freedom of setting the injection refrigerant superheat at the injection port. A few commercial heat pump products have been available based on the concept of two-stage cycle with economizer HX

This model supports flexible system configuration. Component connections are described in a system configuration file; hence any system configuration can be simulated. We have developed 2 system configuration files; one for the flash tank cycle described in Figure 2 (a) and one for the economizer cycle described in Figure 2 (b). Indoor and outdoor HXs have been sized based on an existing 17.59 kW (60 kBTU) heat pump that is available on the market. The compressor is a prototype 3.73 kW (5 HP) R410A scroll compressor with vapor injection ports.



(a) (b)

Figure 2: Schematics of two-stage cycles with flash tank (a) and two-stage cycle with economizing heat exchanger (b).

Problem description

Cold climate heat pump design provides a unique challenge. The cold climate heat pump has to meet rating criteria at

AHRI rating conditions and provide acceptable performance at low ambient conditions. Heat exchanger designs need to be optimized at the rating conditions; however at low ambient conditions we can only modify fan speeds and compressor capacity using a 2 speed fan and a variable capacity compressor. Hence a 2 step optimization procedure is used. The result from the first optimization problem is used as the baseline for the second optimization problem. Flash tank and economizer vapor injected cycles are considered in this study.

In the first optimization study, the fin materials on both indoor and outdoor HXs are minimized in order to reduce the initial equipment cost and compressor capacity is modulated to meet the required performance targets. Minimum capacity and COP constraints are used to meet the requirement at the AHRI rating conditions, i.e. 8.33°C (47°F) ambient. This problem is described as shown in equation (2) below. In this problem, the HX tubing and circuitry were not modified in order to avoid the need for retooling; however, fin density (FD) is varied on both indoor and outdoor HX. A compressor capacity multiplier ($F_{compressor}$) is used as a means to describe the modulated capacity of a variable capacity compressor, which is a multiplier to correct the suction mass flow rate and power consumption predicted by the compressor map. At the AHRI rating conditions, indoor and outdoor airflow rates are assumed to be 1.12 and 2.24 kg/s (8880 and 17760 lbm/h) respectively. Compressor heat loss is assumed to be 20%. Indoor and outdoor fan powers were evaluated based on empirical fan curves and included in the evaluation of the system COP. Furthermore, the evaporator outlet superheat is assumed to be 1.11°C (2°F), considering addition of a suction line accumulator to maintain a near-saturated evaporator exit, the condenser outlet subcooling is assumed to be 5.55°C (10°F), and the economizer outlet superheat is assumed to be 5.55°C (10°F).

$$\begin{aligned} \underset{FD_{indoor}, FD_{outdoor}, F_{compressor}}{\text{minimize}} \quad & A_{\text{fin,indoor}} + A_{\text{fin,outdoor}} \\ \text{subject to} \quad & COP \geq 3.81 \\ & Q_{HP} \geq 17.59\,kW \\ & 164 \leq FD_{indoor} \leq 984 \ \text{Fin/m} \\ & 164 \leq FD_{outdoor} \leq 984 \ \text{Fin/m} \\ & 0.5 \leq F_{compressor} \leq 3 \end{aligned} \qquad (2)$$

In the second optimization study, the fin materials on both indoor and outdoor HXs are kept constant at the optimum values obtained from the initial optimization study however the compressor capacity and outdoor HX fan speed are allowed to change to reach maximum COP at the cold ambient conditions of -26.11°C (-15°F). A minimum capacity constraint is used to reduce the need for electric strip heating. This problem can be described as shown in equation (3) below. At low ambient conditions, the compressor heat loss assumption was modified to 40%.

$$\begin{aligned} \underset{F_{compressor}, \dot{m}_{out}}{\text{maximize}} \quad & COP \\ \text{subject to} \quad & Q_{HP} \geq 8.8\,kW \\ & 0.5 \leq F_{compressor} \leq 3 \\ & 1.47 \leq \dot{m}_{out} \leq 4.9 \ \text{kg/s} \end{aligned} \qquad (3)$$

RESULTS

Optimization results suggest that the flash tank cycle has the best performance; it requires the least amount of fin surface and results in the maximum COP at low ambient conditions. Figure 3 shows the values for the optimum design variables and corresponding total fin surface areas for indoor and outdoor HXs. In this optimization study, the compressor capacity factor for optimum design was 1.0 suggesting that the selected compressor is properly sized for the rating conditions. The flash tank cycle required significantly lower fin density for the indoor coil (480 fin/m instead of 709-752 fin/m for the economizer cycle) and slightly lower outdoor coil fin density (285 fin/m versus 289 to 356 fin/m for the economizer cycle) while meeting the design requirement for COP of 3.81 (EER = 13) and capacity of 17.59 kW (60 kBTU). Higher economizer effectiveness resulted in lower fin density requirements. The economizer outdoor fin density approached that of the flash tank cycle at higher economizer effectiveness. The significant difference in indoor HX fin density is attributed to the larger amount of vapor injection required in the economizer cycle that results in larger refrigerant mass flow rate through the condenser. The evaporator refrigerant mass flow rate is calculated based on the compressor map. The outdoor HX size variation will be greatly limited due to the low variation in evaporator mass flow rate dictated by the compressor map. As a result, the total fin surface area for the flash tank cycle is 17% smaller than that of the 90% effective economizer cycle.

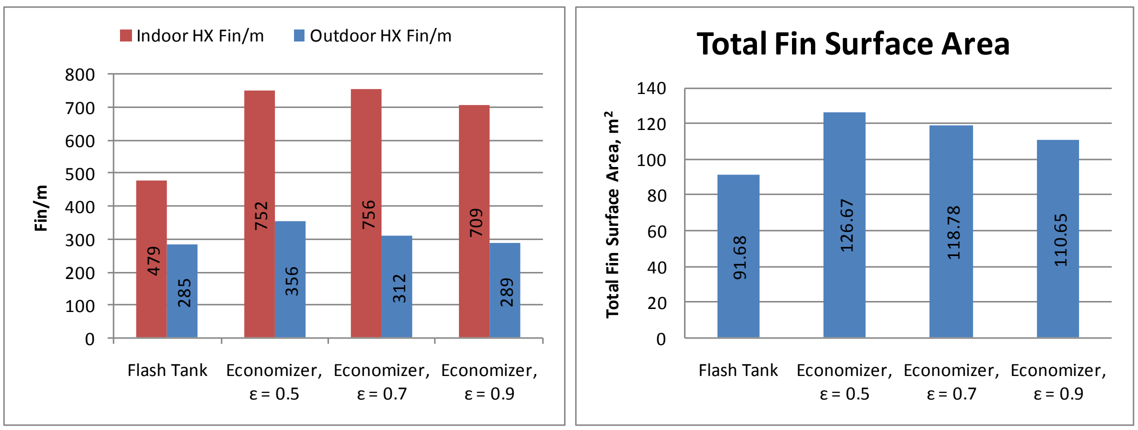


Figure 3: First Optimization step: HX optimization at rating conditions - (a) design variables, (b) objective function value

The second optimization step was targeting compressor and outdoor fan capacity modulation to provide at least 8.8 kW heating capacity while maximizing the COP at low ambient conditions. Figure 4 shows the values for the optimum design variables and corresponding COP at -26.1°C (-15°F). In this optimization study, the compressor capacity multiplier for optimum design was 1.24 for the flash tank cycle and 1.3 for the economizer cycle. The outdoor fan flow rate multiplier varied between 0.65 and 0.9 for the different cycle configurations. The most interesting result is the lower outdoor airflow requirements at low ambient conditions. This is mainly due to HX pinching which limits the evaporator capacity. As such, the optimizer was able to identify lower ambient airflow rates that reduce the fan power requirements while maintaining enough airflow to satisfy the evaporator load. The flash tank cycle required a 4% lower compressor capacity multiplier than did the economizer cycle; mainly due to the higher COP. The economizer effectiveness had no impact on the compressor capacity multiplier. Finally, higher economizer effectiveness resulted in lower ambient airflow requirements as shown in Figure 4 (a).

The maximum COP for the vapor injected cycle at low ambient conditions is shown in Figure 4 (b) for different configurations. The flash tank cycle had a maximum COP of 2.02 (EER = 6.9); 7% better than the 50% effective economizer cycle. The economizer cycle COP shows some dependence on the economizer effectiveness. The 90% effective economizer cycle showed 5% COP improvement over the 50% effective economizer cycle. Increasing the economizer effectiveness from 50% to 90% require more than tripling the overall HX conductance (UA). Assuming the same heat transfer surface configuration will be used, this means 3 times larger heat transfer area. The modest gain in efficiency might not warrant the additional material required to build the larger economizer. This would require the development of an accurate feasibility assessment model that accounts for the initial equipment cost and the expected annual energy consumption in order to estimate the payback period.

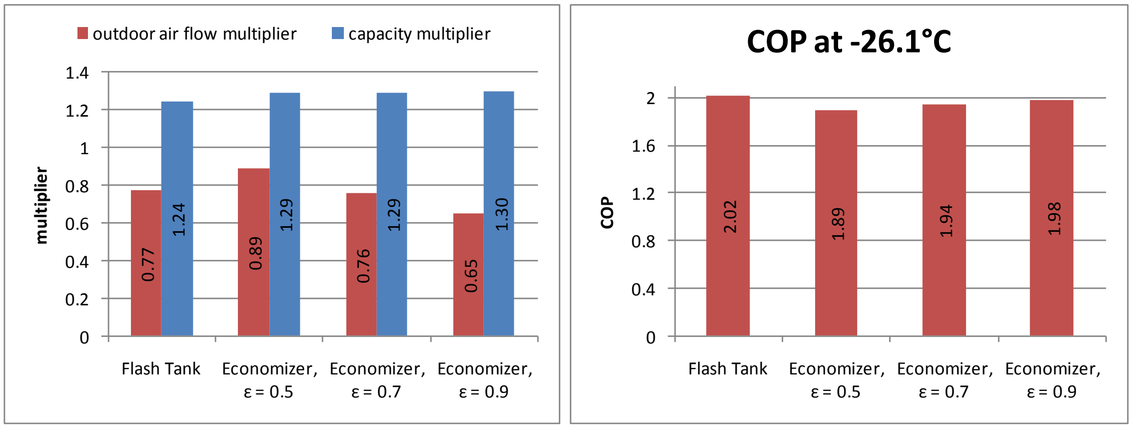


Figure 4: Second Optimization (a) design variables, (b) objective function value

CONCLUSION

A step-by-step approach for the design optimization of cold climate heat pumps was presented. A 2 step optimization approach was presented in this paper: the system was first optimized at the standard AHRI 8.33°C (47°F) ambient rating conditions in order to minimize the amount of fin material, then the optimized system capacity modulation was optimized a low ambient conditions to serve a minimum acceptable load while maximizing the COP. The optimization problem was described formally: objective function, constraints, and design space. An open source publically available optimization toolbox was used along with publically available VC system simulation tool to optimize the cold climate heat pump design. In this paper, VC system model validation was not presented for brevity; however we strongly recommend simulation tool validation prior to engineering optimization. Optimum designs for two different system configurations (a flash tank cycle and an economizer cycle) were presented and analyzed. The optimization results were intriguing and suggest the need for increase compressor capacity and lower ambient airflow rates at low ambient conditions. The optimization results also suggest that the flash tank cycle is superior in performance to the economizer cycle.

Results presented in this paper indicate the power of systematic optimization. Engineering time was spent early on to properly set the optimization problem. Optimization was then performed seamlessly without user intervention; running the computer does not require significant engineering time. Finally, additional engineering time was spent to analyze the optimization results in an effort to understand why the design tradeoff that resulted in this optimum design. The counter-intuitive optimum result of reducing the ambient airflow rates at low ambient conditions stimulated the discussions about the limiting factors for the cold climate performance.

Further cold climate heat pump design optimization work is required in order to assess the technical and economic feasibility of various systems at different climate conditions. This would require the development of new system configurations (e.g. the ejector cycle, booster compressor cycle, etc.), accurate component cost models, and an accurate annual energy consumption model.

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NOMENCLATURE

AHRI = Air-conditioning Heating and Refrigeration Institute

COP = Coefficient of Performance

ε = Heat exchanger effectiveness

$F_{compressor}$ = Capacity multiplier

FD = Fin Density

HX = Heat eXchanger

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