

# FINNED-TUBE CONDENSER DESIGN OPTIMIZATION USING THERMOECONOMIC ISOLATION

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## ABSTRACT

Using a detailed system model as a comparison, this study shows that isolating the condenser component and optimizing it independently by minimizing the entropy generation in the condenser component alone, also known as thermoeconomic isolation, is a practical way to design the condenser for optimum air-conditioning system efficiency. This task is accomplished by comparing the optimum design determined by maximizing the entire system's COP, an undisputed method, with the optimum design determined by minimizing the entropy generation in the isolated condenser component, with consistent constraints used for the two methods. The choice of constraints used in the isolated model is critical and discussed in detail.

The resulting optimum designs from the isolated model produced a COP within 0.62% to 1.7% of the designs found by optimizing the COP using an entire system model. Additionally, a 65% reduction in computation time was achieved.

## INTRODUCTION

The performance of most energy systems is dominated by heat exchanger design. A primary example is the air-cooled condenser heat exchanger used in single-family air conditioning systems with a cooling capacity that is typically 8.8 kW (2.5 tons). In the U.S., these air conditioning units must meet U.S. DOE minimum efficiency standards based on annual performance weighted with operating hours over a wide ambient temperature range. With the upcoming ban in new equipment of R-22, new condenser designs are necessary using environmentally friendlier refrigerants.

Optimization of these heat exchangers is a complex and difficult task requiring the determination of approximately ten different finned tube heat exchanger design interrelated parameters, with appropriate constraints. Additionally, the competing effects of increased heat transfer with increased air and refrigerant frictional pressure drop make it difficult to determine the relative goodness of a design. Therefore, the appropriate selection of a figure-of-merit is very important. The most common figure-of-merit for an air-conditioning system is the efficiency, which is expressed as the coefficient of performance (COP). Minimum entropy generation for the entire cycle can also be used because there is a direct correspondence between COP and entropy generation. Both of these methods require the entire system, including fans, to be modeled in order to determine how changes in the condenser design affect the system performance.

Alternatively, many researchers have optimized the design of heat exchangers using a component based figure-of-merit. The problem with these techniques is that quite often the figures-of-merit used are purely heuristic and have no theoretical basis. For instance, the use of minimum heat transfer area, cost, friction factor or a heat transfer parameter like Colburn j-factor as figures of merit gives little information about the merits of the heat exchanger designs regarding overall system performance.

The concept of thermoeconomic isolation is a method that also falls into the component based category. In this case the figure-of-merit is minimum entropy generation in the isolated condenser component. The idea that the design obtained from minimizing the entropy generation in an isolated component, coincides with the design from maximizing the system performance (COP) stems from the fact that system entropy generation and COP are related through fundamental relations (Alefeld 1990). Since the isolated model is lacking feedback from the rest of the

system as the design changes, it is very difficult if not impossible to predict that the resulting component design is the best from the system performance viewpoint by just looking at an individual component. This has been the issue with thermoeconomic isolation, and why its theoretical development has been limited. However, von Spakovsky & Evans (1988) wrote that although thermoeconomic isolation is an ideal condition that real-world systems can only approach, if approached closely enough both detailed and practical component and system optimizations are possible. The current study shows that thermoeconomic isolation can be used as a practical design tool for optimizing finned-tube condenser heat exchangers with the goal of optimizing the COP of the air-conditioning system they are designed for, but only if appropriate constraints are used in the isolated model.

## 1. LITERATURE REVIEW

### 1.1 Entropy Generation Minimization

The technique of Entropy Generation Minimization (EGM) has been in use since the 1970's and is discussed in detail in Bejan (1982). According to Bejan & Pfister (1980), "the 'energy conservation' value of a heat transfer augmentation technique can be best measured in terms of the technique's ability to reduce the rate of entropy generation (irreversibility, exergy destruction) in the heat transfer device in which it is implemented." The central theme of the EGM method is that by minimizing entropy generation, performance is optimized. Bejan (1980, 1982 & 1996) gives many examples of situations in which EGM can be used to optimize performance, however most cases are very basic in nature and have constraints that make their application to a real world design problem impractical.

Klein & Reindl (1997) considered the effect of heat exchanger optimization in relation to an entire cycle. The work explores the optimum allocation of heat exchanger area for both the reverse Carnot and the vapor compression refrigeration cycle models and a comparison was made between using system performance and minimum entropy generation as the figure of merit. The model used is very simplistic compared to the current study, using specific operating conditions to eliminate details. It was concluded that minimizing the total system entropy generation rate does not always result in the same design as maximizing system performance. However, Alefeld (1990) showed that COP and system entropy generation are directly related through fundamental equations. Additionally, Bejan (1996) suggests that the point of minimum system entropy generation should coincide with optimum system performance. It is believed that the discrepancies found in the Klein & Reindl study are due to the evaporator cooling capacity not being fixed throughout the analysis.

Many other authors have assumed that thermoeconomic isolation is valid, investigating entropy generation minimization in just the heat exchanger component without regard to the system in which it is placed (Saboya & da Costa 2000; Hesselgreaves 2000; Sekulic 1986; McClintock 1951; Witte & Shamsundar 1983; San & Jan 2000). Most of these analyses are for very simple heat exchanger types, not for finned tube heat exchangers. However, Lin & Lee (1998) used the analysis of Bejan & Pfister (1980) applied to a wavy plate finned-tube heat exchanger. Their analysis only studied a comparison of the inline vs. staggered tube array. All of these studies assumed that by optimizing the heat exchanger component the entire system would be optimized, but none attempted to address the validity of this issue.

### 1.2 Heat Exchanger Modeling and Optimization

Several authors, such as Vardhan and Dhar (1998), Bensafi et al. (1997) and Corberan & Melon (1998), have developed detailed models of finned-tube heat exchangers with good accuracy compared to experimental data. These authors, however, did not use these models for optimization purposes.

Others, such as Tayal et al. (1999) and Jiang et al. (2002), have looked at searching for optimum heat exchangers using combinatorial optimization schemes such as genetic algorithms. Tayal et al. (1999) concludes that genetic algorithms are equally effective as compared to simulated annealing in solving black-box optimization model problems. Their study focuses on shell-and-tube heat exchangers for process industries. Jiang et al. (2002) developed a user-friendly simulation and optimization tool for design of coils also using a genetic algorithm for the optimization scheme. There was a great deal of focus placed on the refrigerant side in this study, allowing for complex circuitry arrangements. Both of these studies used minimum heat transfer area as objective functions in their optimization procedures. Tayal et al. also considered cost as an objective function.

Richardson et al. (2002) developed a program that simulates a vapor compression system. This program is then used to optimize the system level variables using gradient based and genetic optimization routines. Parameters included in this simulation program are: refrigerant charge, COP, weight, capacity, and cost. Optimization objective functions used were COP, capacity and system weight. This study did not optimize the design of the individual components; rather it used available components and found the best combination of them for specified inputs. The heat exchanger models used in this simulation program are simplistic, assuming infinite air flow rate and fixed properties at the inlet regime for refrigerant pressure drop calculations.

Wright (2000) developed a detailed simulation of the components of a typical U.S. residential type air-conditioning system, including the compressor, plain finned-tube condenser, evaporator, expansion valve and condenser and evaporator fans. The condenser was the focus of Wright’s model incorporating the best available simulations for the air-side and refrigerant-side pressure drops and heat transfer coefficients based on R-410a as the working fluid. Wright was not able to perform an exhaustive design optimization search using this model due to computational time limitations. Therefore, Aspelund (2001) added to Wright’s study by implementing a design optimization search technique (using the Simplex Method by Nelder & Mead, 1965) to optimize 10 operational and geometric design parameters of the condenser with a COP figure-of-merit. The current study uses a program based on the preliminary work of Wright & Aspelund. The details of the model and optimization program can be found in Wright (2000) and Aspelund (2001) respectively, while improvements made in the current study are explained in Stewart & Shelton (2003). The methodology used to define the isolated condenser model is discussed in the next section.

## 2. ISOLATED CONDENSER MODEL

In order to perform thermoeconomic isolation, the component of interest must be entirely “isolated” from the rest of the system. By doing this, the component ceases to give or receive feedback from the rest of the system. However, in order to produce a properly posed and complete model, some amount of information introduced as operating constraints is required. A total of five constraints are needed to complete the isolated condenser model. Referencing Figure 1, the options from which to choose are:

$$P_{1air}, P_{2air}, P_{2r}, P_{3r}, T_{1air}, T_{2air}, T_{2r}, T_{3r}, \dot{m}_r, \dot{Q}_{cond}, \Delta T_{SH}, \Delta T_{SC}, \text{ and } T_{sat}.$$

where  $\Delta T_{SC}$  is the degrees of subcool exiting the condenser, which is a parameter used to determine the refrigerant charge in the system. Also,  $T_{sat}$  is the saturation temperature in the condenser and  $\Delta T_{SH}$  is the degrees of superheat entering the condenser. Note that this is not the normal definition of superheat in an air conditioning system, which usually describes the superheat exiting the evaporator.

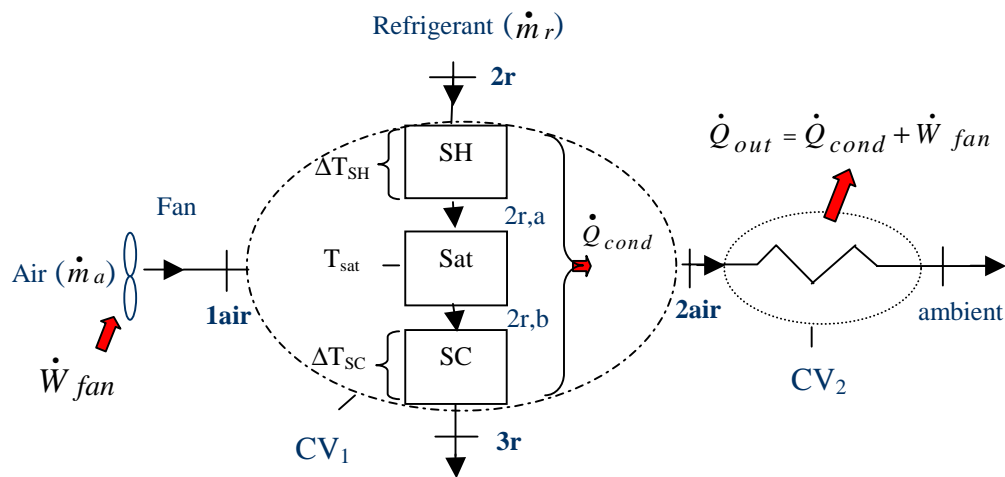


Figure 1: Detailed Condenser Flow Diagram

For the condenser of a residential air-conditioning system, two of the needed five constraints are rather straightforward; i.e. the inlet air pressure and temperature are assumed to be ambient. Therefore, the  $P_{1air}$  is taken as 1 bar, and  $T_{1air}$  was taken as 27.8°C (82°F), which was found by Wright (2000) to give a nearly identical COP as the seasonal COP used by the U.S. Department of Energy in rating residential air conditioners. These values were used in the overall system model as well. The pressure drop and temperature increase on the air side are then calculated by the model.

Three additional constraints are needed to complete the isolated model. The appropriate selection of these three constraints is extremely important, therefore the decision making process is detailed here. Ideally, one parameter would give information about the refrigerant side entrance conditions ( $P_{2r}$ ,  $T_{2r}$ ,  $T_{sat}$ ,  $\Delta T_{SH}$ ), the second parameter would give information about the refrigerant side exit conditions ( $P_{3r}$ ,  $T_{3r}$ ,  $T_{sat}$ ,  $\Delta T_{SC}$ ), and the third would give information about the heat transfer scaling between the air side and the refrigerant side. While pretty much any combination of the parameters will result in a working model, certain combinations will not give as good of results due to their variations with designs. To find the best combination of parameters the following requirements should be met: 1. the parameters should not require detailed system information to arrive at a value, and 2. they should not vary greatly with different heat exchanger designs when operating in the system. Because of these two requirements, explicit temperature and pressure information are not good options, therefore  $P_{2r}$ ,  $P_{3r}$ ,  $T_{2r}$ ,  $T_{3r}$  and even the saturation temperature in the condenser are taken out of consideration as it will vary with heat exchanger design. Therefore, with  $\Delta T_{SH}$  and  $\Delta T_{SC}$  being fixed, the only decision left to be made is the size scaling parameter, which could be either the refrigerant mass flow rate or the heat transfer rate between the refrigerant and the air. Both of these situations were investigated in more detail.

Fixing the mass flow rate of the refrigerant in the isolated condenser model does not fix the total heat transfer rate in the heat exchanger. This is due to the fact that as the saturation pressure varies between heat exchanger designs, the heat of condensation varies. Taken with fixed inlet superheat and exit subcool, the heat transfer rate increases as the saturation pressure decreases. In the system model, the refrigerant mass flow rate varies to compensate for this, while the total heat transfer rate stays relatively constant. Therefore, fixing the heat transfer rate is a more appropriate size scale constraint.

As shown by equations (1) and (2), the heat transfer rate can be approximated quite easily using the definition of COP and the desired cooling capacity, e.g., size, of the system that the condenser is being designed for:

$$COP = \frac{\text{Desired Output}}{\text{Required Input}} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} = \frac{\dot{Q}_{evap}}{\dot{Q}_{cond} - \dot{Q}_{evap}} \quad (1)$$

Rearranging and solving for  $\dot{Q}_{cond}$ :

$$\dot{Q}_{cond} = \dot{Q}_{evap} \left( 1 + \frac{1}{COP} \right) \quad (2)$$

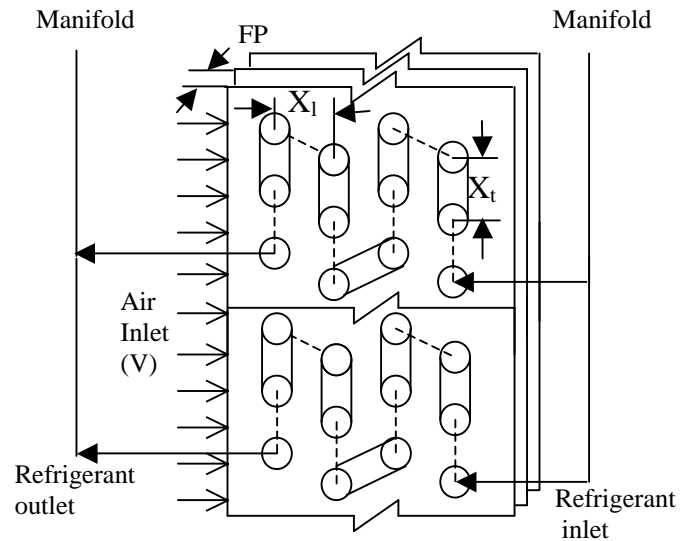
With a refrigerating capacity fixed at 8.8 kW and a typical COP of approximately 4, this gives  $\dot{Q}_{cond} = 11\text{kW}$ .

The value for the subcool used in the system model was set at 5°C (9°F) at an ambient temperature of 35°C (95°F). The optimum subcool is actually 0°. However for practicality reasons, systems are usually designed with some excess subcool to be certain that for varying operating conditions no vapor will enter the expansion valve. Using this design condition the system model then calculates the subcool for the actual ambient condition of 27.8°C (82°F). The resulting calculated subcool at 27.8°C was slightly higher than 5°C depending on the condenser design, so for the isolated model the value was fixed at 5°C for simplicity.

The value for the condenser inlet superheat was set at 25°C (45°F) in the isolated model. This value was determined by looking at some typical standard system designs.

**Table 1: Design Parameters**

<b>Geometric:</b>
Condenser Material Cost (C)
Frontal Area ( $A_f$ )
Number of Rows (N)
Tubes per Row (tpr)
Fins Pitch (FP)
Number of circuits (circ)
Transverse Tube Spacing ( $X_t$ )
Longitudinal Tube Spacing ( $X_l$ )
<b>Operational:</b>
Air Velocity (V)
<b>Fixed:</b>
Aspect Ratio (AR) = width/height
Tube Diameter (D)
Fin thickness (t)



**Figure 2: Condenser Design Parameters  
(Shown: 4 rows, 2 circuits, 6 tubes per row)**

### 3. DESIGN PARAMETERS

Table 1 shows all of the condenser heat exchanger design parameters considered in this optimization study. Many are also depicted in Figure 2. The number of tube rows multiplied by the longitudinal tube spacing determines the depth of the condenser in the direction of air flow, while the tubes per row multiplied by the transverse tube spacing determines the heat exchanger height. The number of circuits determines the number of parallel paths the refrigerant mass flow rate is divided between in the heat exchanger.

For all of the optimizations the tube diameter was fixed at 7.94 mm (5/16 in.). This is because Aspelund (2001) found that the smaller the tube, the better the COP, with little improvement beyond 7.94 mm. Also, the fin thickness was fixed at 0.15 mm (0.006 in.). When left in as a search parameter, the solution always converged to a design with thinner and thinner fins while making the fin spacing closer together and decreasing the air velocity to account for the increased pressure drop. This makes sense theoretically, however, in reality extremely thin fins are not structurally durable, and when the fin spacing is too close dirt and dust will clog the fins. Because of this, the fin thickness was fixed as a reasonably minimum value of 0.15 mm.

Additionally, the overall heat exchanger aspect ratio (width divided by height) was fixed as three; since the outside A/C condensing/compressor unit is typically a cube in shape with the condenser bent around three sides. If the aspect ratio were not specified, the design converges to one row of a single very long finned tube. This is because tube bends at the ends have pressure drop but no heat transfer. Also, if the frontal area were not constrained, the design would always converge to a single row heat exchanger with a frontal area large enough to accommodate the single row. Therefore, larger frontal area is better, if space and packaging allows.

Heat exchanger price is typically based on a multiple of material cost assuming mass production. Therefore, the material costs of the condenser and evaporator were calculated using the cost of copper (US\$1.76/kg) and aluminum (US\$1.54/kg) from the London Metals Exchange. The volume of each metal was calculated and multiplied by its density and cost per pound to determine the material cost for each heat exchanger.

### 4. RESULTS & DISCUSSION

The resulting system and isolated condenser models were run under the conditions described in the previous section using the optimization search scheme developed by Aspelund (2001). For each case, there is a fixed frontal area and

a fixed cost. The resulting optimum designs for the system model and the isolated condenser model are compared in the figures below. For the COP calculation, the optimum design determined by the isolated condenser optimization was then run in the system model to determine the system COP for that design.

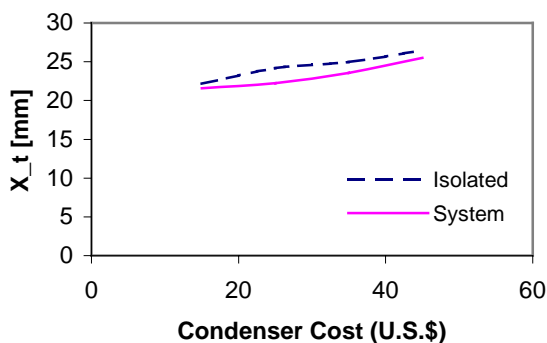
Figures 3-5 show the transverse tube spacing, the longitudinal tube spacing, and the fin pitch respectively vs. varying condenser material cost for a  $0.75\text{m}^2$  frontal area. It can be seen from these figures that the resulting optimum designs from both the system and the isolated model optimizations are quite similar to each other in both trend and value.

The isolated model optimization search scheme was developed to choose an optimum design based on minimizing the entropy generation in the condenser. The components of this entropy generation are heat transfer through a finite temperature difference and both air and refrigerant pressure drop terms. These temperature and pressure effects compete with each other, i.e. as the pressure drop decreases the heat transfer coefficient decreases. Because of this, the optimum design will converge to the case with a minimum of irreversibility due to the trading off these effects. Therefore, as the isolated optimization scheme reduced the air side velocity (Figure 6, for  $0.75\text{m}^2$  frontal area), it reduced both the air and refrigerant-side pressure drops. This is because in order to maintain the constant condenser heat transfer rate, there was a slight increase in the average saturation temperature and pressure in the condenser and a slight increase in the refrigerant side mass flow rate, but this flow rate was split into more (18.2%-29.3%) parallel circuits (number of circuits) resulting in a lower pressure drop on the refrigerant side compared to the corresponding system optimized design (e.g. at \$25,  $0.75\text{m}^2$ : 69.84 kPa vs. 119.1 kPa). The net result of these effects reduced the entropy generation in the condenser, which was the figure of merit for the isolated model.

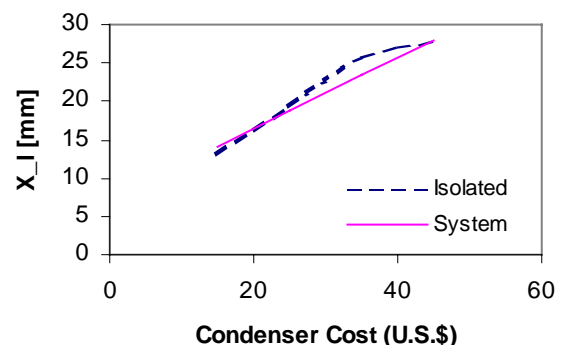
However, when this isolated optimum design is placed in the context of the overall system, the reduction in pressure drop in the condenser creates the need for a larger irreversibility in the expansion valve. Also, the increase in saturation pressure in the condenser (with a fixed  $\Delta T_{SH}$ ) and the increased refrigerant mass flow rate requires a larger compressor power, also creating more irreversibility. The net result of these effects offset the lower entropy generation in the condenser.

The overall effects of these model differences can be seen in Figure 7, which shows COP vs. varying condenser material cost for a fixed frontal area of  $0.75\text{m}^2$ . It can be seen that the COPs from the system model optimizations are slightly higher than the isolated condenser optimized designs. The variation ranges from 0.62% at \$45 to 1.7% at \$15. So, even though some of the design parameters differed by as much as 29.3% between the two methods, the isolated condenser optimization did produce designs very close in COP to those produced by using an entire system model to optimize the condenser design. The 65% reduction in computation time for the isolated model makes this a very attractive and practical option for the design optimization of finned-tube condenser heat exchangers.

Alternatively, at a fixed cost of \$25, and varying frontal area the same design trends are seen. The only difference is shown in Figure 8 where COP is plotted versus frontal area. Note that the COP will always increase with increasing frontal area until there is a single row.



**Figure 3: Transverse Tube Spacing vs. Condenser Cost ( $0.75\text{m}^2$  Frontal Area)**



**Figure 4: Longitudinal Tube Spacing vs. Condenser Cost ( $0.75\text{m}^2$  Frontal Area)**

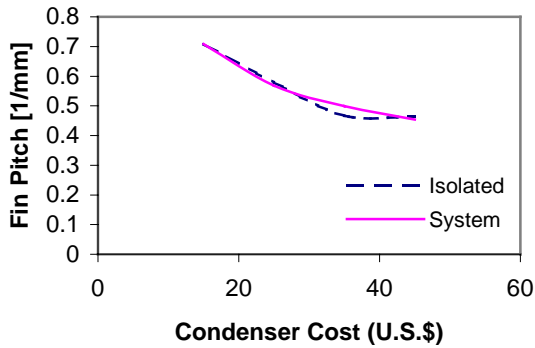


Figure 5: Fin Pitch vs. Condenser Cost (0.75m<sup>2</sup> Frontal Area)

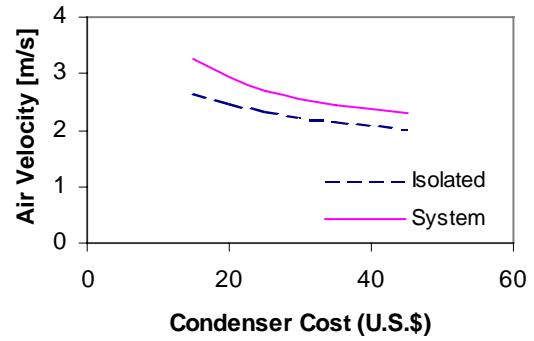


Figure 6: Air Velocity vs. Condenser Cost (0.75m<sup>2</sup> Frontal Area)

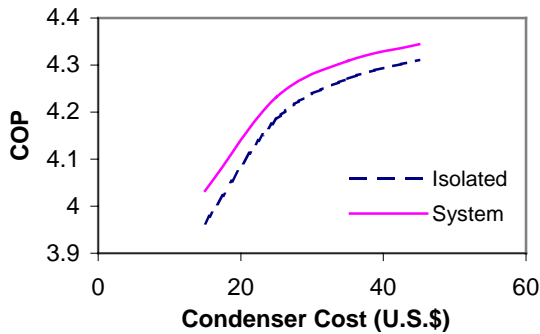


Figure 7: COP vs. Condenser Cost (0.75m<sup>2</sup> Frontal Area)

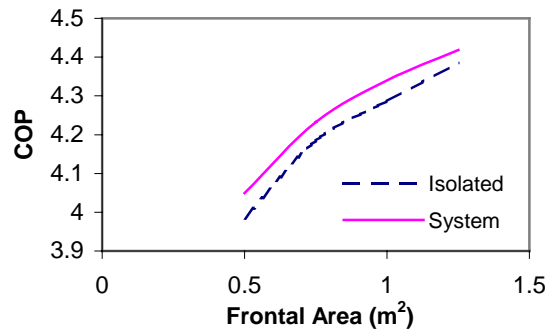


Figure 8: COP vs. Frontal Area (Fixed Cost of \$25)

## CONCLUSIONS

Using a detailed system model as a comparison, this study shows that isolating the condenser component and optimizing it independently by minimizing the entropy generation in the condenser component alone (also known as thermoeconomic isolation) is a practical way to design the condenser for optimum air-conditioning system efficiency. This task is accomplished by comparing the optimum design determined by maximizing the entire system's COP, an undisputed method, with the optimum design determined by minimizing the entropy generation in the isolated condenser component, with consistent constraints used for the two methods. It was found that the most important aspect required for a successful isolated model is a proper set of constraints. In the current study, the condenser heat transfer rate, condenser exit subcool, and the condenser entering superheat were chosen as fixed parameters. These parameters vary little in the system model with changing condenser designs and they do not require detailed system information in order to specify them within good accuracy.

The resulting optimum designs from the isolated model produced a COP within 0.62% to 1.7% of the designs found by optimizing the COP using an entire system model. With a 65% reduction in computation time using the isolated model, this provides a very practical and effective method for designing finned-tube condenser heat exchangers.

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RESUME: Nous présentons une étude comparative pour concevoir un condenseur optimisant l'efficacité d'un système de climatisation. Nous avons utilisé la méthode du système global et la méthode de minimisation de l'entropie du système isolé. Cette tâche est accomplie en comparant les résultats obtenues grâce à ces deux méthodes en leur imposant des contraintes consistantes. Dans le cas du système isolé, le choix de ces contraintes est un paramètre critique que nous discuterons en détail.

La conception optimale de condenseur obtenu par le modèle du système isolé produit une efficacité de 0.62 à 1.7 % de celui obtenue par optimisation de l'efficacité du système global. Notons également une réduction du temps de calcul de 65% a été obtenue grâce au modèle du système isolé.