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Plate-fin and Tube Heat Exchangers Refrigerant Circuiting Optimization in Vapor Compression Refrigeration Systems

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ABSTRACT

Refrigerant circuitry in condensers and evaporators has a significant effect in the performance of refrigeration systems. The optimized project of the refrigerant circuits in refrigeration systems with plate-fin heat exchangers is not trivial, due to the complexity of their representation as well as the high number of possible combinations, even when methodologies of intelligent optimization are used. The present work proposes a new methodology for the simultaneous optimization of refrigerant circuiting in air-air refrigeration systems with plate-fin and tube heat exchangers. This new methodology proved to be more efficient than traditional methods. The method was applied, in conjunction with a full refrigeration system simulator for the optimization of a high performance commercial air-conditioning unit, considering the use of heat exchangers with tubes of different diameters. Overall, a predicted COP enhancement between 5.5% and 8.3% for system with R-22, and 6% - 6.5% for R410A, was observed with the optimization method.

1. INTRODUCTION

A most effective way of improving refrigeration cycle efficient is, of course, to improve the efficiency of its components. Concerning air-to-air heat pump and refrigeration cycles, among possible options for optimization (Fowler *et al*. 1997; Matos *et al*. 2004), one advantageous measure is to search for an optimum refrigerant circuitry of the condenser and evaporator coils (trough appropriate junction of the coil tubes). This approach, besides practical and low-cost, can provide a significant increase in heat transfer capacity and, consequently, cycle performance, without major changes in the way of the manufacturing process of the heat exchanger. This has been experimentally verified by Chwalowski *et al*. (1989) and Liang *et al*. (2001). Inasmuch as the number of possible laboratory tests towards circuitry optimization is restricted by time and cost, simulation tools have to be employed (Kaufman e Michalski, 2004). A number of works on coil circuitry optimization can be found in the literature.

Granryd and Palm (2003) proposed a simplified method to determine an optimum number of circuit branches in evaporators, for a given thermal capacity. Correlations were proposed based on a minimum pressure drop.

Circuiting optimization is a particularly complex task, evenmore if air flow distribuition is not uniform. And the experience gained with the optimization for a given refrigerant may not be automatically extended to other substances (Groll, 2008). Considering the typical characteristics of the optimization problem (discrete or continuum domain**,** large search space, complex relationship among the different components, multiple variables and restrictions), the use of genetic algorithm - a well known technique (e.g., Goldberg, 1989) - seems to be an appropriate choice. Presently, cycle optimization of systems with plate-fin and tube heat exchangers is carried out in a sequential manner, i.e., the circuitry of each heat exchanger is individually optimized for maximum thermal capacity (Domanski *et al*., 2005).

Kaufman and Michalski (2000) developed a hybrid algorithm called ISHED1 (Intelligent System for Heat Exchanger Design), to optimize the circuit of a plate-fin and tube heat exchanger for maximum heat capacity. The EVAP-COND program (Domanski, 2001) was used to simulate the heat exchangers. The ISHED1method (Kaufman and Michalski, 2000) succeeded to obtain better results than those of a human expert (Cervone *et al*., 2000), and has been used with relative success in the optimization of evaporators (Kaufman and Michalski, 2004) and condensers (Domanski and Yashar, 2007a) for refrigerants R600a, R134a, R290, R22, R410A and R32.

To overcome the defficiencies of genetic operators in the applications of heat exchanger circuiting optimization, a thecnique called Learnable Evolution Model (LEM) was implemented to increase the velocity of the evolution process (Michalski, 1998; Cervone *et al*., 2000). The history of the population was taken into account (Saleem and Reynolds, 2000; Vervoce *et al*., 2003). A similar methodology was presented by Wu *et al*. (2008). The same circuiting representation of Kaufman and Michalski (2004) and Domanski and Yashar (2007b) was employed.

The use, for example, of an evaporator with an optimized circuit is bound to provide a better performance to the refrigeration system (Domanski *et al*., 2005). However, such approach should be used with caution, since isolated optimization of the components may lead, occasionally, to cases with deteriorated system global performance (Granryd and Palm, 2003). Such situations could be avoided should the optimization be carried out over a complete refrigeration cycle simulation, aiming at the system global performance. To the authors` knowledge, no work in the literature has been found where system performance maximization was conducted with the simultaneous optimization of both evaporator and condenser. In fact, to conduct a simultaneous optimization, as opposed to sequential optimization, with the genetic algorithm techniques described so far in the literature (Kaufman and Michalski, 2000 and 2004; Domanski and Yashar, 2007a; Wu *et al*., 2008), would result in a unrealistic number of possible solutions to be processed. In order to overcome such limitation, an improved optimization method, based on genetic algorithm, is described in the present work.

2. CYCLE SIMULATION

To simulate the vapor compression cycle, a computational package, Genesym v1.0 (Yana Motta, 2001), was employed. It has three modes of operation: heat pump, air conditioning and air-cooled chiller. For the purposes of the present work, it simulated an air-to-air vapor compression cycle air-conditioner operating in steady-state condition. The simulation model comprises a number of models, one for each component. The compressor is modeled based on compressor operating maps and fundamental equations. The maps are made available in the program data-base for a number of compressor models. Tube-by-tube local models are employed to simulate the condenser and evaporator. Air-side convective heat transfer coefficient is determined for different types of areas (flat, wavy, lanced or louvered fin) from a number of correlations available in the literature (Wang, *et al*., 1998, 1999a, 1999b, 2000a, 2000b and 2001). Refrigerant-side local condensing and evaporating heat transfer coefficient were determined from Cavallini *et al*. (1999), and two-phase flow pressure drop, from Choi (1999). The expansion device can be: (i) a capillary tube, for which mass flow rate correlations were taken from Yana Motta (1999); (ii) a short tube orifice (Payne, 1997); or (iii) an expansion valve (Hwang *et al*., 2004; Chen, 2008). Refrigerant thermophysical properties are calculated with built-in libraries from REFPROP v7.0 (Lemmon *et al.*, 2002). Reported validation of the program against experimental data can be found in (Domanski and Payne, 2002), for refrigerants R-22 and R-410a, and (Hwang et al. 2004), for R-22 and R-290. The program has also been used for a number of refrigerant R-22 replacement studies (Spatz and Yana Motta, 2004; Chen, 2008).

Evaporator and condenser coils require the following input data: number of repeating sections of coil (slabs), number of rows (depth), number of tubes per row, coil width, height of the coil, pitch between tubes of the same row, tube depth pitch (distance between rows), distance between coil bottom and last tube, number of fins per unit length, fin thickness, type of fin, finned area specific characteristics, fin and tube materials thermal conductivities, type of inside tube surface, tubes outer and inner diameters. Refrigerant circuitry (which implies coil inlet and outlet tubes, refrigerant flow direction and points of merging and splitting circuits, i.e., the refrigerant flow path through the coil) as well as air flow distribution are also provided as input data. Refrigerant mass flow rate and power consumption maps, as a function of evaporating and condensing temperatures, are provided by user and comprise the compressor input data. Each heat exchanger fan is characterized by its volumetric flow rate and power consumption.Three refrigerant lines are considered: suction, discharge and liquid lines. For them, the following input data are required: tube length, outer diameter, thickness and material thermal conductivity and density, insulation thickness, density and thermal conductivity, and prevailing external ambient conditions (dry bulb temperature, relative humidity and pressure). Indoor and outdoor conditions (dry bulb temperature, relative humidity and pressure) are provided, together with the cycle evaluation mode (options of prescribed degree of superheat, prescribed refrigerant charge, both or prescribed degrees of superheat and subcooling). Finally, for refrigerant cycle input data, operating conditions were also characterized, in addition to the prescribed values of above, by condensing and evaporating temperatures.

Solution was obtained sequentially, evaluating component module by module. During iterations, refrigerant mass flow rate was adjusted so that the prescribed conditions could be met. Evaporator and condenser coils were also solved sequentially, tube by tube. Overall cycle results include: refrigerating capacity, compressor power consumption, COP, refrigerant mass flow rate, air volumetric flow rate in condenser and evaporator, refrigerant charge, thermodynamic states at the inlet and outlet of each relevant control volume. Given the local nature of heat transfer correlations, which requires a local analysis of both condenser and evaporator, local values of the thermodynamic states of air and refrigerant, alongside the refrigerant coil, are also available.

3. CIRCUITRY OPTIMIZATION

The optimization objective could be, for example, the maximization of COP or of the refrigeration capacity. It could also be the minimization of refrigerant charge, of overall weight or of energy consumption. Variables of the problem would be those defining the refrigerant circuitry. For example, considering a heat exchanger comprised by five tubes, numbered 1, 2, 3, 4 and 5, a vector (4, 2, 1, 3, 5) would represent a heat exchanger circuitry with inlet in tube number 4, continuing in tube number 2 up to the last tube, number 5. As in any optimization problem, the final solution is limited by constraints. In the present case they are: (i) a lower limit for thermal capacity, which is the system nominal capacity; (ii) maximum cost (for example, by maintaining the overall geometry and altering refrigerant circuitry solely, one would avoid costly manufacturing changes); (iii) user design constraints, such as inlet and outlet locations or maximum tube connection length (for example, tubes belonging to the same branch would have to be reasonably close to one another); (iv) prescribed heat exchanger volume and overall dimensions.

The objective function is provided by the simulation program and could be formally described as follows:

$$Objective\ Function = GENESYM\ (\underbrace{Circuitry, Geometry}_{\substack{input \\ (variables)}}, ..., \underbrace{COP, \dot{Q}}_{\substack{output \\ (results)}}, ...) \qquad (1)$$

The first step of the method is to determine, even before the genetic algorithm is applied, the optimum number of circuit branches, or parallel sections. In spite of the existence of an optimization method, from Granryd and Palm (2003), a comprehensive search method was preferred. It runs the vapor compression system simulation program for every possible combination of circuits in the evaporator and condenser (from 1 circuit to a reasonable value).

After the number of branches in both heat exchangers has been determined, the circuitry optimization procedure starts. A new methodology was devised and comprises two steps: (i) the previous reduction of the domain, by

imposing the problem restrictions, in a kind of filtering process; (ii) application of the genetic algorithm in the search of the optimal solution within the filtered (and reduced) domain. The application of step (ii) only after the domain has been restricted reduces considerably the computational time spent in the application of the genetic algorithm. By its turn, the genetic algorithm is only applied to solutions that have already been filtered, i.e., that comply with the problem restrictions, thus eliminating the necessity of corrective methodos. The overall approach can thus be established: (step 1) determine number of zones, or circuits; (step 2) filter solutions; (step 3) generate database of all possible circuitries for evaporator and condenser; (step 4) apply the genetic algorithm to find the optimum. Figure 1 depicts an example of evaporator and condenser coils, with schematics of tube connections, circuits and representation of inlet/outlet and connections with adjacent tubes only restrictions.

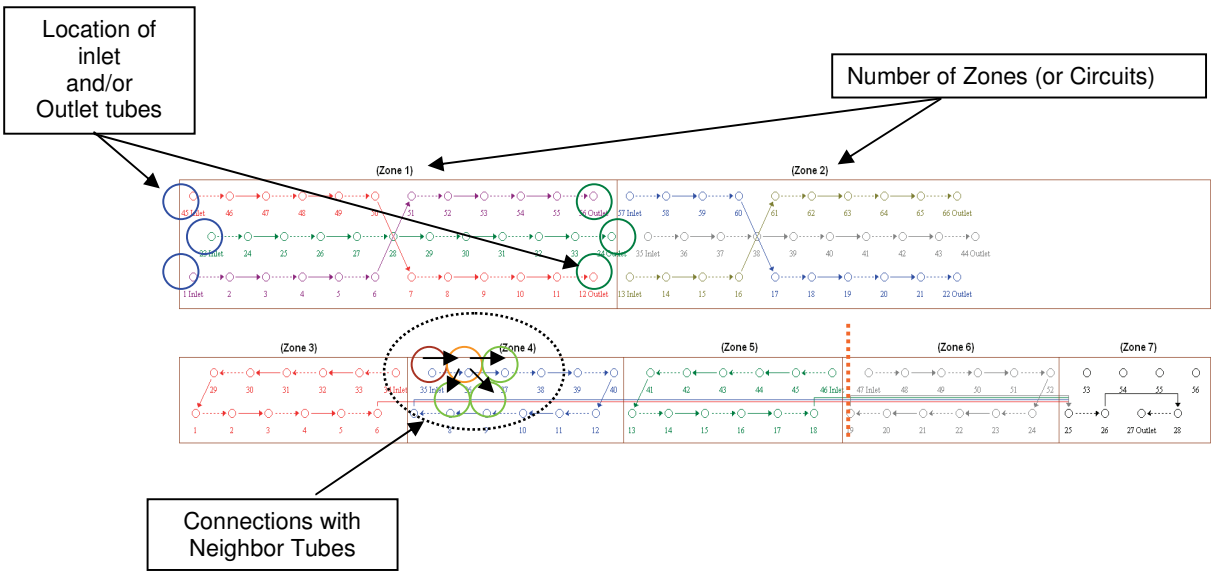

Figure 1: Schematics of condenser and evaporator coils, with refrigerant circuits and inlet/outlet and adjacent tube restrictions.

The basic procedure of application of the genetic algorithm is as follows:

1) Initial population or new circuit: Generate, in a random way, all circuits in the domain;
2) Organize – Generate maximum and minimum throughout the population: Criteria of better capacity and COP are used to organize circuits;
3) New circuits: Originated from best circuits in the previous step, (2), or created ones;
4) Evaluate and organize - Generate maximum and minimum throughout the population: Criteria of better capacity and COP are used to organize circuits;
5) Substitution: Change worse circuits for better ones;
6) Is it the best circuit?
    a. Yes: Circuit delivered.
    b. No: Go back to 2.

More details of the method can be found in (Castillo Martínez, 2009).

4. RESULTS

The method was applied to a commercially available split air conditioner of 3.2 and 3.3 TR of capacity operating with R22 and R410A, respectively. The unit comprises plate-fin and tube 3x20 tubes evaporator and a 2x26 tubes condenser, a hermetic scroll compressor and a thermostatic expansion valve (Domanski and Payne, 2002). Figure 2 shows the original heat exchangers circuitries. Two sizes of tube diameter were tested for the condenser, 7.7 mm and

6.9 mm, and evaporator, 9.4 mm and 7.9 mm. The degree of subcooling was 3.8 K and the evaporator degree of superheating, 7.6 K.

Four optimization runs were performed for each refrigerant, R22 and R410A. The objective function aimed at optimizing the system coefficient of performance. The optimal number of refrigerant parallel circuits in the evaporator was found to be 5. For the condenser this number ranged between 3, 4 and 6, with a subcooling circuit being considered for all of them. Heat exchangers characteristics were as follows, for evaporator and condenser, respectively: length = 0.660 m, 2.045 m; height = 0.508 m, 0.660 m; number of tube rows = 3, 2; number of tubes per rows = 20, 26; number of fins per m = 472.4, 866.1; type of fin = lanced-slit; tube inner surface = microfin (microfin height = 0.2 mm, number of microfins = 60 and 50), smooth. Tube inner diameters for evaporator and condenser varied according to test, as follows: test C3T1 (R22, 9.4 mm and 7.7 mm); C3T2 (R22, 7.9 mm and 7.7 mm); C3T3 (R22, 9.4 mm and 6.9 mm); C3T4 (R22, 7.9 mm and 6.9 mm). Tests C3T5 (9.4 mm and 7.7 mm), C3T6 (7.9 mm and 7.7 mm), C3T7 (9.4 mm and 6.9 mm) and C3T8 (7.9 mm and 6.9 mm) were conducted with R410A.

Tables 1 and 2 sumarize the results, comparing base values of performance parameters, namely: refrigerating capacity, compressor power, COP, refrigerant mass flow rate, refrigerant charge and evaporating and condensing temperatures. One observes that, in all cases, the percentage of COP enhancement was never below 5.5%. A maximum of 8.34% was obtained. These results show that the change in tube diameter did not impact significantly the COP improvement. On the other hand, as a result of a more compact geometry, the optimized configuration provided significant reduction on refrigerant charge (from 19.8% to 30.6%). Figure 3 depicts the resulting optimized evaporator and condenser circuitries for test C3T5 (R410A).

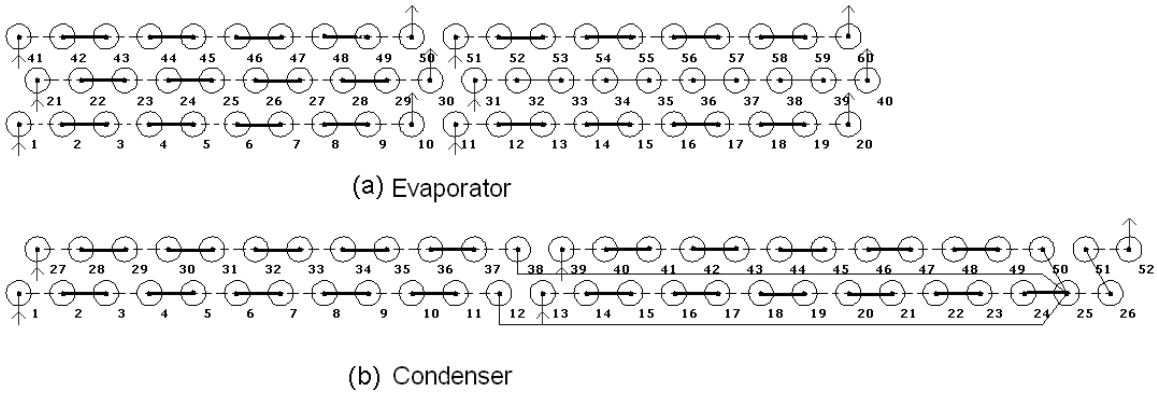

Figure 2: Original refrigerant circuitries of evaporator and condenser.

		C3T1			C3T2			C3T3			C3T4		
		Base	Optimized	Δ[%]	Base	Optimized	Δ[%]	Base	Optimized	Δ[%]	Base	Optimized	Δ[%]
capacity	[kW]	11,313	11,861	4,84	11,017	11,412	3,59	11,335	11,86	4,63	11,045	11,457	3,73
compressor power	[kW]	2,518	2,479	-1,55	2,517	2,471	-1,83	2,556	2,469	-3,40	2,554	2,472	-3,21
COP	[-]	4,493	4,7854	6,51	4,3771	4,6178	5,50	4,4342	4,8042	8,34	4,3247	4,6346	7,17
refrig.mass flow rate	[kg/s]	0,06738	0,07021	4,20	0,06573	0,06789	3,29	0,06724	0,07012	4,28	0,06558	0,06783	3,43
refrigerant charge	[kg]	3,02	2,217	-26,59	2,863	2,019	-29,48	2,718	1,946	-28,40	2,563	1,785	-30,36
evaporating temp.	[K]	281,63	283,17	0,55	281,7	283,2	0,53	281,6	283,12	0,54	281,65	283,18	0,54
condensing temp.	[K]	312,33	312,01	-0,10	312,11	312,01	-0,03	312,34	311,92	-0,13	312,11	311,69	-0,13

Table1: Results for optimization tests with R22.

		C3T5			C3T6			C3T7			C3T8		
		Base	Optimized	Δ[%]	Base	Optimized	Δ[%]	Base	Optimized	Δ[%]	Base	Optimized	Δ[%]
capacity	[kW]	11,819	12,447	5,31	11,685	12,245	4,79	11,852	12,446	5,01	11,709	12,231	4,46
compressor power	[kW]	2,485	2,458	-1,09	2,486	2,458	-1,13	2,496	2,46	-1,44	2,496	2,461	-1,40
COP	[-]	4,7553	5,0646	6,50	4,7004	4,9825	6,00	4,7478	5,0583	6,54	4,6906	4,9705	5,97
refrig.mass flowrate	[kg/s]	0,06848	0,07144	4,32	0,06771	0,07021	3,69	0,06837	0,07144	4,49	0,06775	0,07015	3,54
refrigerant charge	[kg]	3,004	2,409	-19,81	2,838	2,201	-22,45	2,64	2,096	-20,61	2,474	1,909	-22,84
evaporating temp.	[K]	281,61	282,96	0,48	281,65	282,98	0,47	281,57	282,96	0,49	281,68	282,98	0,46
condensing temp.	[K]	312,23	311,88	-0,11	312,13	311,72	-0,13	312,09	311,91	-0,06	312,01	311,76	-0,08

Table 2: Results for optimization tests with R410A.

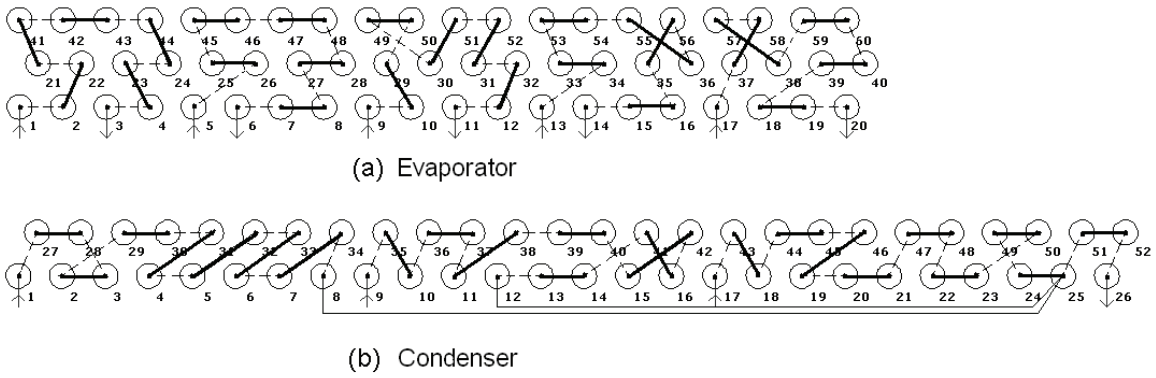

Figure 3: Optimized evaporator and condenser circuitries for test C3T5 (R410A).

5. CONCLUSIONS

A new method for refrigerant circuitry optimization was proposed. The application of the genetic algorithm over pre-filtered domain resulted in a more efficient and faster optimization algorithm, which allowed the simultaneous manipulation of condenser and evaporator coil circuitries, towards the overall cycle optimization. The method was applied to a typical air conditioning system, for which a COP enhancement of up to 8.34% was found. For this particular case, refrigerant charge was reduced by 28.4%, showing that this can be an efficient tool in the design of air conditioning systems with lower environmental impact, from the direct (refrigerant charge) and indirect (energy consumption) emissions point of view.

NOMENCLATURE

COP	coefficient of performance	(-)
$\dot{Q}$	refrigeration capacity	(kW)

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