Thermoeconomic Lifecycle Energy Recovery System Optimization for Central Air-Conditioning System Using Evolutionary Technique

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RECEIVED ON 07.06.2009 ACCEPTED ON 13.08.2009

ABSTRACT

Energy efficient systems are the most desirable systems. Due to huge rise in energy prices and lack of availability of energy, the effective use of energy has become the need of time. Energy recovery both in heating systems as well as in air-conditioning systems saves a lot of energy. In this paper energy recovery system has been designed and optimized for central air-conditioning systems for various ranges. Cost function includes capital cost along with pumping and exergy destruction cost. This shows that installation of energy recovery system with a central air-conditioning has a significant amount of saved energy and payback period is within a year. PFHE (Plate Fin Heat Exchanger) is designed and optimized using evolutionary optimization. In order to verify the capabilities of the proposed method, a case study is also presented showing that significant amount of energy is recovered at a reasonable payback period. Sensitivity analysis is also done with the energy prices.

Key Words: Heat Exchanger, Evolutionary Optimization, PFHE, Payback Period.

1. INTRODUCTION

Due to global warming, the use of air-conditioning system has increased substantially. Central air-conditioning systems are employed in hospitals, factories, universities, offices, auditoriums etc. These systems take fresh hot air at regular interval of time to maintain the level of oxygen in the environment. In ventilation hot air goes into and cold air leaves air-conditioning space, which changes the temperature of the system.

Energy recovery ventilation systems provide a controlled way of ventilating a space while minimizing energy loss. This reduce the cost of heating ventilated air in the winter by transferring heat from warm inside air being exhausted to the fresh (but cold) supply air. In the summer, the inside air cools the warmer fresh air to reduce ventilating cooling costs. Off course there is a capital investment for installing these energy recovery systems but they will pay off in few years time.

Efficient energy recovery can substantially reduce the mechanical heating and cooling requirements associated with conditioning ventilation air in most of the spaces. There are some spaces (hospitals, laboratories, auditoriums etc) which typically require 100% outside air at high ventilation rates between 6 and 15 air changes per hour. Heating and cooling systems can be downsized when energy recovery is used, because energy recovery

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systems reduce peak heating and cooling requirements. Energy recovery system used for air-conditioning (centrally air-conditioning) uses a compact heat exchanger. Fluids on both sides of that heat exchanger is air, which has low specific heat, hence a large surface to volume ratio (β) is required to get maximum energy recovery. PFHE is the type of heat exchanger used for air to air (gas to gas) heat exchange.

In the literature, attempts have been made to automate and optimize the heat exchanger design process for energy recovery system but the problem is still the subject of ongoing research. The suggested approaches vary in the choice of objective function, type and number of sizing parameters and optimization method employed.

Tozer, et. al. [1] discussed how to economically optimize the HVAC system for an absorption-cogeneration variable air-volume system. Their objective function was annual life-cycle cost per unit cooling capacity, which was minimized. Summerer [2] suggested a method for estimating absorption system costs for the optimum design to maximize the COP. Zalewski, et. al. [3] solved two thermo-economic objectives for designing an evaporative fluid cooler to maximize heat transfer duty and minimize operating cost.

Fowler and Bejan [4] determined the optimum Reynolds number for external flows over several simple geometries. They used entropy generation as a minimization method to determine the optimal solution. They first showed the optimal Reynolds number scaled with the size of the device, but with ever-increasing entropy generation. They showed that the size of the device should increase to minimize losses, but the tradeoff for larger size is the cost.

Arzu, et.al. [5] did the energy analysis of lithium bromide/water absorption systems. They analyzed the classic vapor absorption system using first law and second law of thermodynamics. They showed that COP improved slightly when increasing the heat source temperature, and exergetic efficiency decreases.

Yonghan, et.al. [6] provided the experimental data that can be used in the optimal design of flat plate finned-tube heat exchanger with large fin pitch. They showed that the convective heat transfer coefficient increases with reduced fin pitch.

Hao, et. al. [7] used GA (Genetic Algorithm) combined with BP (Back Propagation) neural networks for the optimal design of PFHE. The major objectives were the minimum total weight and total annual cost for given constrained conditions.

Reneaume, et. al. [8] used the COLTECH software and optimized the length of hot and cold streams, volume or weight keeping fin spacing fixed and in second step optimized the fin spacing keeping lengths fixed. Nobrega, et.al. [9] studied the application of heat and enthalpy wheels in HVAC system to reduce the thermal load. Their results showed that heat wheels can be far less efficient than enthalpy recovery wheels, depending on the atmospheric conditions.

In the present study optimized PFHE parameters are found for maximum life cycle net saved energy. In order to contribute to a solution of the optimized PFHE design parameters, return and fresh air temperatures and humidity is obtained. To get optimized life cycle saved energy using PFHE evolutionary optimization technique is used. Evolutionary optimization uses the power of computer and given penalty function gives the best optimum results. It searches for number of possible minima/maximum and gives the best minimum or maximum value. Chances to take the local minima will reduce and hence more accurate results. The drawback of this method is that it needs lot of computer resources. With the increased power of computers, it becomes an opportunity to use these methods for such system having discrete as well as continuous parameters to optimize.

2. OPTIMIZATION METHODOLOGY

Design of optimum heat exchanger includes the following steps.

- Based on required amount of heat transfer, estimate the surface area required and other design specifications, assuming some initial values.
- Calculate the capital cost, operating cost, total cost and amount of saved energy (objective function)
- Using the optimization algorithm, to select the new set of values for the design parameters.
- Iteration of the previous steps until a minimum of the objective function is found.
- Repeat the process for different geometries (about 21 geometries) mentioned in Kays & London [10].
- Optimize the net saved energy for the geometry giving maximum net saved energy for different air-conditioning capacities.

In addition to the optimizing algorithm, heat exchanger lifecycle optimizing model for maximum energy recover includes a comprehensive thermal and hydraulic design procedure. The thermal design of heat exchangers is used to calculate an adequate surface area to handle the thermal duty for the given operating conditions whereas the hydraulic analysis determines the pressure drop of the fluids (fresh air and return air) flowing in the system, and consequently the pump or fan required energy input necessary to maintain the flow.

Design parameters are the mass flow rates for fresh and return air and their inlet temperatures return air humidity, type of geometry (mf, mr, Tfi, Tfo, Rhr, Geom). Other parameters are found using energy balance. Fixed parameters of geometry are assigned directly.

The optimization parameters whose values are iterated are fresh air (hot air) pressure drop (δPf), return (return air) air pressure drop (δPr), relative humidity of fresh air (Rhf), fresh air side length (Lft) and return air side length (Lrt). This process of optimization is shown in Fig. 1.

2.1 PFHE Design Procedure

The heat exchanger must meet the process requirements (to recover maximum energy), it must withstand the service conditions, it must be maintainable and it should cost should be consistent with the above requirements. In the design process usually effectiveness and pressure drops are specified and design is made to satisfy these specifications.

To compute the fluid bulk mean temperature and the fluid thermo-physical properties on each fluid side, determine the fluid outlet temperature from the specified heat duty.

$$q = (\dot{m}\,Cp)(T_{f,i} - T_{f,o}) = \dot{m}\,Cp(T_{r,o} - T_{r,i}) \qquad (1)$$

In most of the sizing problems of PFHE is specified then

$$T_{f,o} = T_{f,i} - \varepsilon(C_{min}/C_f)\,(T_{f,i} - T_{f,i}) \qquad (2)$$

$$T_{r,o} = T_{r,i} - \varepsilon(C_{min}/C_r)\,(T_{f,i} - T_{r,i}) \qquad (3)$$

Initially the air Cp values are taken at the given inlet temperatures of fresh (hot air) and return fluid (cold air) and then calculate the outlet temperature using Equations (2-3). Dynamic viscosity (μ), specific heat at constant pressure (Cp), thermal conductivity (k), Prandtl Number (Pr) and density (ρ) are taken at the mean fluid temperature.

ε-NTU approach is used for cross flow with fluid unmixed condition. In case both fluids are gases or both fluids are liquid, we can consider that the design is balanced (i.e. the thermal resistances are distributed approximately equally on the hot and cold sides).

$$C^* = \frac{C_{min}}{C_{max}} \tag{4}$$

$$\varepsilon = 1 - \exp[(\frac{1}{C_r})(NTU)^{0.23}\{\exp[-C^*(NTU)^{0.78}] - 1\}] \tag{5}$$

$$NTU_r = NTU_f = 2NTU \tag{6}$$

When a particular geometry is selected on each side we take some initial guess for Colburn and Fanning friction factor ratio (j/f) for initial approximation of core mass velocities (G) and Reynolds number (Re). Initial guess for μ_o is usually between 0.7-0.8. Shah [11] and calculate the Reynolds number.

Where D_h is the hydraulic diameter.

Once the Reynolds number is found values of j and f is again calculated using:

$$j = 0.06522\, Re^{-0.5403} (\frac{s}{h'})^{-0.1541} (\frac{\delta}{ls})^{0.1499} (\frac{\delta}{s})^{-0.0678} \times [1 + 5.269 \times 10^{-5} Re^{0.504} (\frac{s}{h'})^{0.504} (\frac{\delta}{ls})^{0.456} (\frac{\delta}{s})^{-1.055}]^{0.1} \tag{7}$$

$$f = 9.6243\, Re^{-0.7422} (\frac{s}{h'})^{-0.1856} (\frac{\delta}{ls})^{0.3053} (\frac{\delta}{s})^{-0.2659} \times [1 + 7.669 \times 10^{-8} Re^{4.429} (\frac{s}{h'})^{0.920} (\frac{\delta}{ls})^{3.767} (\frac{\delta}{s})^{0.236}]^{0.1} \tag{8}$$

These correlations are valid for $120 < Re < 104$.

Where s = Pf - δ, h' = b - δ b = b1

Equations (7-8) are from Shah [11].

Where, Pf, b, δ, s and h are geometry dependent properties which are given in [10].

By knowing theses properties heat transfer coefficient on hot and cold sides are calculated. After this overall fin efficiency is calculated using Equation (9).

$$\eta_o = [1 - (1 - \eta_f)\frac{A_f}{A}] \tag{9}$$

Overall heat transfer coefficient is calculated using thermal resistance (Equation (10)).

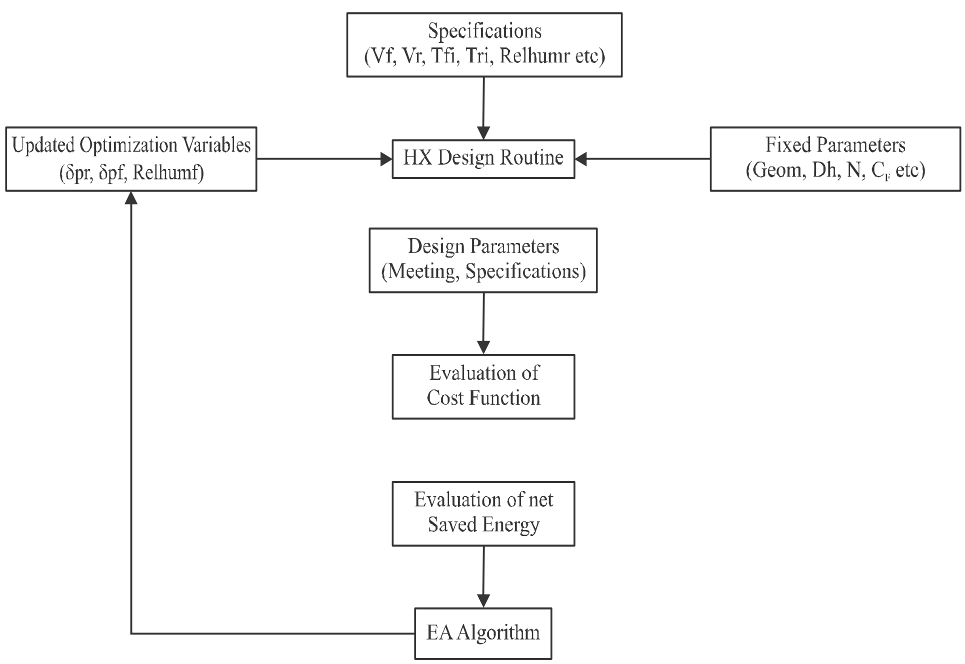

FIG. 1. PROPOSED EA OPTIMIZATION FLOW CHART

$$\frac{1}{U}=\frac{1}{(\eta_o h)_f}+\frac{1}{(\eta_o h)_r}+\frac{\alpha_f/\alpha_r}{(\eta_o h)_r}+\frac{\alpha_f/\alpha_r}{(\eta_o h)_f} \tag{10}$$

Values of α_f and α_r are defined as:

$$\alpha_f=\frac{b_f\beta_f}{b_f+b_r+2\delta_w} \tag{11}$$

$$\alpha_r=\frac{b_r\beta_r}{b_r+b_f+2\delta_w} \tag{12}$$

Where β is total heat transfer Area/Volume between plates (m^2/m^3).

The ratio of surface area on inlet air side (hot side) and return air side is equal to the ratio of α_f and α_r. Once NTU and C_{min} is known surface area can be calculated. From the specified $\dot{m}$ and computed G, the minimum free-flow area on the hot side (fresh air) is calculated using Equation (13):

$$A_{o,f}=(\frac{\dot{m}}{G})_f \tag{13}$$

The air flow length is then computed from the definition of hydraulic diameter.

$$L_f=(\frac{D_hA}{4A_o})_f \tag{14}$$

Similarly the return air (cold side) surface area and length can be calculated:

$$L_r=(\frac{D_hA}{4A_o})_r \tag{15}$$

To calculate the core frontal area on each fluid side, first cross-sectional area to frontal area (σ) is calculated, and then frontal area on both sides can be calculated using Equation (16):

$$A_{fr}=\frac{A_o}{\sigma} \tag{16}$$

After the frontal area on fresh air and return air sides are calculated length L_3 from both sides is approximately the same.

$$L_3=\frac{A_{fr}}{L_f}=\frac{A_{fr}}{L_r} \tag{17}$$

Next is to calculate the pressure drop on tube and shell side. This eventually gives the amount of power required to overcome this pressure drop. This contributes to operating cost of a heat exchanger. First determine the fluid densities at the exchanger inlet and outlet (ρ_i, ρ_o) for each fluid. The mean specific volume on each fluid side is then computed from:

$$(\frac{1}{\rho})_m=\frac{1}{2}(\frac{1}{\rho_i}+\frac{1}{\rho_o}) \tag{18}$$

For the offset strips fin geometries because of the frequent boundary layer interruptions, the flow is well mixed and is treated having the Reynolds number very large $(\mathrm{Re}\to\infty)$. Friction coefficient is corrected at the mean temperature.

Finally the pressure drop is calculated using following Equation (19) [11]:

$$\Delta P=\frac{G^2}{2g_c\rho_i}[(1-\sigma^2+Kc)+2(\frac{\rho_i}{\rho_o}-1)+ f\frac{L}{r_h}\rho_i(\frac{1}{\rho})_m-(1-\sigma^2-Ke)\frac{\rho_i}{\rho_o}] \tag{19}$$

2.2 Objective Function Calculation

Heat exchanger lifecycle cost is divided in two parts, capital cost and the operating cost. The capital cost is the sum of material cost and construction or manufacturing cost. Operating cost is the sum of pumping cost and energy destruction cost. Once the total operating cost of the heat exchanger for its entire life has been formulated the present cost estimation due to inflation rate must be included.

$$C_{TOT} = C_{capcost} + C_{opD} \tag{20}$$

Different methods are used for capital cost. In this paper capital invest $C_{capcost}$ is computed as a function of the exchanger surface area adopting Vatavuk's correlation [12].

$$C_{capcost} = 231(Af + Ar)^{0.639} \tag{21}$$

Heat exchanger operating cost incorporates pumping cost for fresh air (hot air) and return air (cold air) and the exergy destruction cost due to the irreversibility. Since the energy used for operation is electricity, the operating cost is related to the electricity cost in kWh (kilowatt hour).

Total pumping power is the sum of the pumping power required for return and fresh air. Pumping power is calculated:

$$P_P = \frac{\dot{m}\,\Delta P}{\rho\eta} \tag{22}$$

Exergy destruction cost is associated with irreversibilities in the system. It includes energy change due to temperature difference and energy change due to pressure drop. Energy change for the fresh air (hot air) coming in and the return air (cold air) leaving the a/c space is calculated using the following Equation (23-25):

$$excf = \dot{m}f[(h_{fin} - h_{fout}) - T_o(s_{fin} - s_{fout})] \tag{23}$$

$$excr = \dot{m}r[(h_{rin} - h_{rout}) - T_o(s_{rin} - s_{rout})] \tag{24}$$

$$\dot{exd} = -excf - excr \tag{25}$$

Total value of pumping power and energy destruction cost required in kWh is calculated by multiplying with number of hours of use with the cost of 1kWh of electricity.

Fresh air entering the system when passed through PFHE (energy recovery system) will reduce its temperature and eventually the compressor work which is the saving of energy and eventually the value of saved energy. This saved energy is calculated as:

$$W_C = \frac{\dot{q}_f}{COP} \tag{26}$$

$$\dot{q}_f = \dot{m}f \times C_{Pf} \times \Delta T \tag{27}$$

Equations (26-27) give the power saved in kW. Once it is multiplied with the hours of operation and number of days of operation during a year, it gives the energy saving due to energy recovery system. Amount of air exchanged depends upon the tonnage and the application of air-conditioning system. Values used for case study are for a hospital and Carrier standards [13] are used.

$$W_{CY} = \frac{m_f \times \dot{C}_{Pf} \times \Delta T}{COP} \times N_H \times N_D \tag{28}$$

Total operating cost for the life of heat exchanger (pumping and exergy destruction cost) including inflation effect.

$$OPC = \frac{((CP_{PY} + \dot{Exdcy}) \times C_F)(\frac{1}{(I+1)^N} - 1)}{\frac{1}{I+1}} \tag{29}$$

Similarly cost of energy recovered over the life span of air-conditioner is found using the same approach.

$$TER = \frac{((Wcy) \times C_E)(\frac{1}{(I+1)^N} - 1)}{\frac{1}{I+1}} \tag{30}$$

Where I is the inflation rate and N is the life of heat exchanger in years.

Hence, the total LCC (Life Cycle Cost) of energy recovery system is:

$$LCC = C_{capcost} + OPC \tag{31}$$

2.3 Optimization Problem Description

The optimization procedure intends to find out the maximum value of life cycle saved energy, which presents the objective function, subjected to the following constrains.

Objective Function:

Maximize:

TER (Total Energy Recovered)

Subject to:

(a) Different geometries with different fin pitch Kays and London [10]

(b) $\dot{m}_r$ = the specified value of return air

(c) Tfin = $Tfin_{min} < Tfin < Tfin_{max}$

(d) Tfout = fresh air specified value

(e) Trin = return air specified value

(f) Rel-Humdity of Fresh air = $Relhumf_{min} < Reslhumf < Relhumf_{max}$

(g) $\Delta P_H = \Delta P_C = \Delta P_{min} < \Delta P < \Delta P_{max}$

2.4 Evolutionary Optimization

Evolutionary optimization procedures derive inspiration from the Darwinian Theory of evolution. The principle idea as opposed to classical optimization methods is that, instead of using the analytical model of a function f(x) and the corresponding gradients for guiding the search along suitable directions a stochastic procedure is used. This procedure is initiated by randomly generating a set of possible solutions. Each solution refers to as an individual and the set itself has referred to as the population. For reasonable performance of the optimization procedure, it is necessary that these individuals be scattered through the entire solution space [14].

2.4.1 Initialization

The population is initialized by generating individuals randomly such that they lie within specified constraints. Each individual consists of numerical values that represent the fresh air inlet temperature (hot fluid inlet temperature), fresh air relative humidity (hot fluid relative humidity) and fresh and return air pressure drops. The number of parent and offspring individuals in each generation is fixed at five and thirty-five respectively. Each simulation run of the evolutionary optimization procedure consists of hundred generations.

2.4.2 Fitness Function

The objective of the optimization procedure is to determine ΔPH, ΔPC, and fresh air relative humidity such that for given operating conditions net energy recovered for the lifecycle of air-conditioning system is maximized. The fitness function is therefore, defined simply as:

Maximize:

f = NER (Net Energy Recovered)

Subject to:

$$Tfin \in [Tfin \min \quad Tfout \max]\, K$$
$$\delta PH \in [\delta PH \min \quad \delta PH \max]\, kPa$$
$$\delta PC \in [\delta PC \min \quad \delta PC \max]\, kPa$$
$$relhum\_hot \in [relhum_{\min} \quad relhum_{\max}]\,\%$$

Here, maximum lifecycle energy recovered value is determined by the given limits of fresh air (hot) inlet temperature, fresh and return air pressure drops and fresh air relative humidity conditions.

The evolutionary optimization method of choice for this investigation is Evolutionary Strategies. Fig. 2 shows the flow chart of this method.

Optimization is carried on personal computer using the evolutionary technique by using program written in scientific computing environment MATLAB®. Each generation was made of 35 individuals. The maximum number of generation was set to 500. However, in the test convergence was always obtained within about 50 generations.

4. RESULTS

A computer simulation program has been developed in MATLAB® to do the optimization using evolutionary technique shown in Fig. 2.

4.1 Case Study Analysis

Data used for analysis and optimization for energy recovery system is of a moderate size hospital with four operation theaters, twenty private rooms shared by two patients, wards, a laboratory and other (mess, kitchen etc).

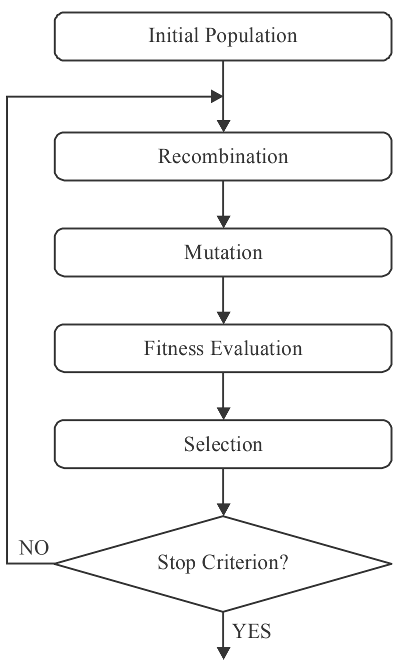

FIG. 2. EVOLUTIONARY STRATEGIES FLOW CHART

Fresh air requirement for each is taken using Carrier hand book [12]. For such amount of load air-conditioner load required is around 100 tons.

Fresh air requirement for such a capacity is found to be 3.16 m^3/s, which is approximately 0.0316 m^3/s per ton of air-conditioning. Energy is recovered from the return air (cold air) such that fresh air (hot air) coming in is at low temperature and hence reduces the compressor work. Plate-fin cross flow, with both fluid unmixed heat exchanger is used for energy recovery system. Temperature and thermal properties for the case are shown in Table 1.

Different standard geometries named (A-U) [10] are used for fresh and return air sides. On both sides same type of geometries are used. Twenty one different geometries are used to find which geometry gives the minimum cost and maximum energy recovered.

For the optimization variables fresh air pressure drop, return air pressure drop, fresh air humidity, return air length, fresh air length (dPf, dPr, Relhum_fresh, Lrt, Lft) the following lower and upper bounds were imposed.

$0.5kPa < \delta Pf < 1.5kPa$

$0.5kPa < \delta Pr < 1.5kPa$

35% < Relhum_fresh < 85%

$0.8m < Lrt < 1.5m$

$0.8m < Lft < 1.5m$

Discounted operating costs were computed with N=10 years, annual discount rate i=10%, energy cost C_{fuel}=0.12 \$/kWh, daily working hours Hd=18 and number of days in a year the system is used are Dy=180.

TABLE 1. INPUT DATA FOR THE CASE STUDY

Case Data	Volume Flow Rate (m^3/s)	T Input (ºC)	Relative Humidity (%)	ρ (kg/m^3)	Cp (kJ/kg K)	μ (Pa s)	k (W/m K)
Fresh Air Data: Moist Air	3.16	33.9	35-85	Mean Temperature	Mean Temperature	Mean Temperature	Mean Temperature
Return Air Data: Moist Air	35.31	23.9	50	Mean Temperature	Mean Temperature	Mean Temperature	Mean Temperature

For different types of fins (A-U) configuration used on both sides keeping other parameters like fresh air inlet temperature, return air temperature, fresh and return air humilities and pressure drops are constant, value of energy recovered and cost associated with it is evaluated. It is found that geometry-H gives the maximum value of energy recovered over the whole life of energy recovery system. In calculating the value of energy recovered effect of inflation with time is also included. It is evident from Fig. 3 that maximum value of energy recovered for 100 ton air-conditioning is of geometry-H. Fig. 4 shows that for different air-conditioning capacities geometry-H is still the best choice as far as net energy recovered value is concerned.

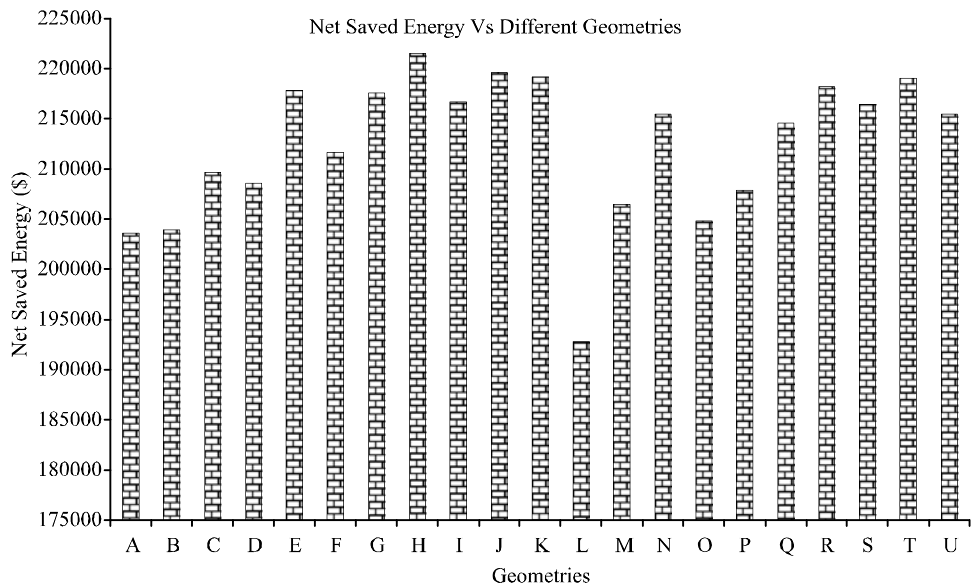

FIG. 3. NET SAVED ENERGY VALUE FOR DIFFERENTGEOMETRIES OF A 100 TON AIR CONDITIONING SYSTEM

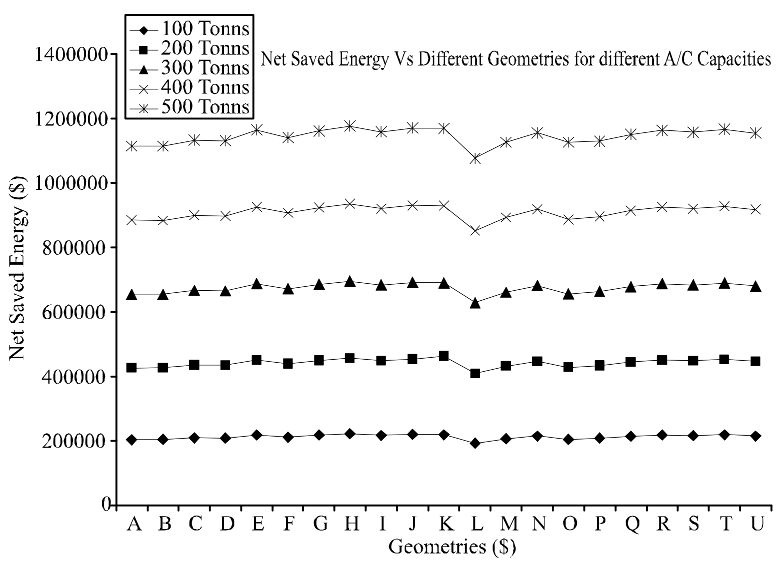

FIG. 4. NET SAVED ENERGY VALUE FOR DIFFERENT GEOMETRIES AT DIFFERENT AIR-CONDITIONING CAPACITIES

Further investigation on cost of energy recovery systems and amount of energy recovered value shows that for the longer period of time net energy recovered is large compared to system capital and running cost (pumping cost and energy destruction cost).

Effect of inlet temperature and humidity on net saved energy is also studied.

Net recovered energy increases as fresh air inlet temperature increases which is shown in Fig. 5. It is due to the fact that higher temperature gradient recovered more energy. Hence energy recovery system becomes more effective in summer.

Relative humidity is the percentage of water vapors present in air compared with the saturated water vapors present in air at atmospheric temperatures. Fig. 6 shows that net amount of energy recovered increases as the relative humidity increases. It is due to the removal of latent heat which is quiet high. As humidity increases more latent heat is removed and hence larger amount of energy is recovered.

Payback period is one of the important factors for the recovery of the investment. This is one of the measures of time in which the investment is recovered. Analysis shows that payback period is within months (less than a year) and it is decreasing with the increasing capacity because of the economy of scale.

CRR (Capital Recovery Rate) is the payment rate per dollar of investment and the interest. It is found to be 6.14% per year which is lower than the interest rate of 10% and quiet feasible investment.

The CCE (Cost of Conserved Energy) is the better measure of cost effectiveness than the length of payback period because it can be directly compared to local energy costs. Its value is calculated for different capacities of A/C and it is in the range of 0.55-0.25 ¢kWh which is much less than the residential electricity at 12 ¢/kWh. Fig. 7 shows clearly that CCE is decreasing with the increase in A/C capacity due to economy of scale.

For the sake of sensitivity analysis, trials were also made by parametrically changing electricity cost in the total cost function in order to assess the sensitivity of EA to variation in the economic parameters. The effect of ±50% variation of electricity prices with respect to nominal value was examined. The results are shown in Table 2. These results

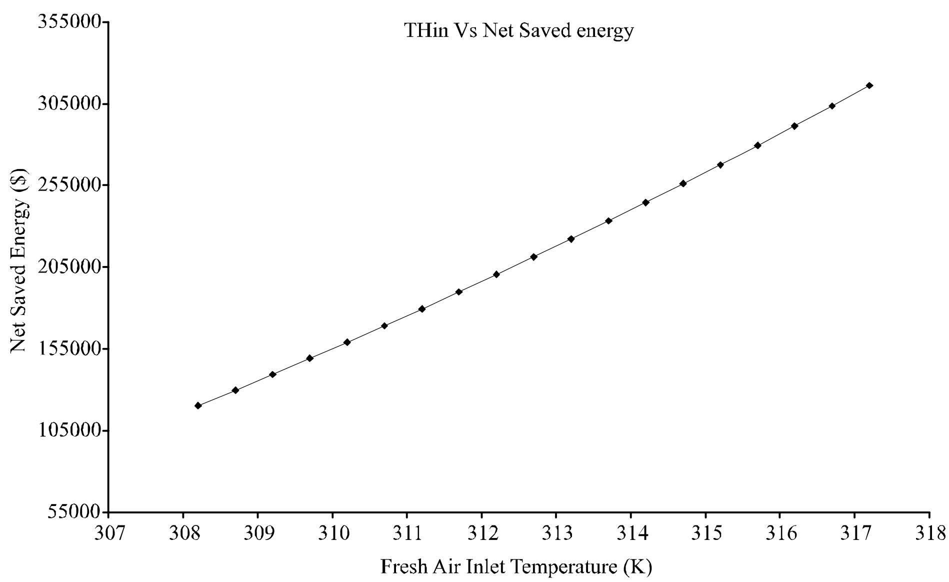

FIG. 5. NET ENERGY RECOVERED VALUE FOR DIFFERENT FRESH AIR INLET TEMPERATURES OF 100 TON OF AIR-CONDITIONING

describe that the EA responds correctly by trying to increase the net energy savings with less payback period with the increase of fuel cost and making opposite when CF (Cost of Fuel) decreases. Values shown in Table 2, when CF increased by 50% pumping cost increased by 48% and discounted operating cost increased by 50%, but the net saved energy (saved energy value minus the total operating cost) increased by 57% and the payback period reduced to only 7 months. When CF was decreased by 50%, pumping cost was decreased by 50% total operating cost was also decreased by 50% and net saved energy decreased by 57%. CCE has decreased from 0.84-0.58 ¢/kWh when cost of energy increased from 12-18¢/kWh.

4.2 Application of Evolutionary Optimization

For the purpose of this study, the return air (cold fluid) temperature is taken as 298.2K, its humidity is taken as 50%

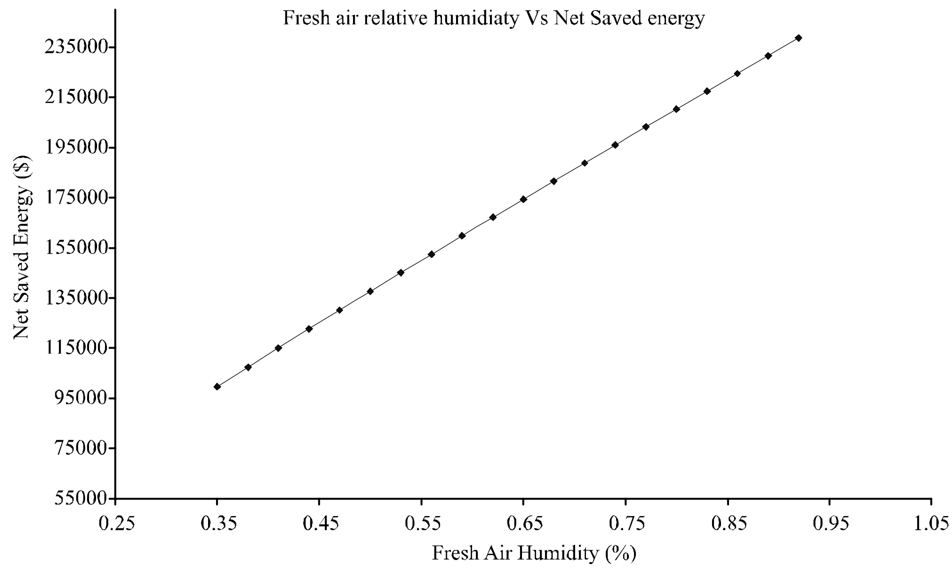

FIG. 6. COMPARISON OF NET RECOVERED ENERGY FOR DIFFERENT FRESH AIR RELATIVE HUMIDITY

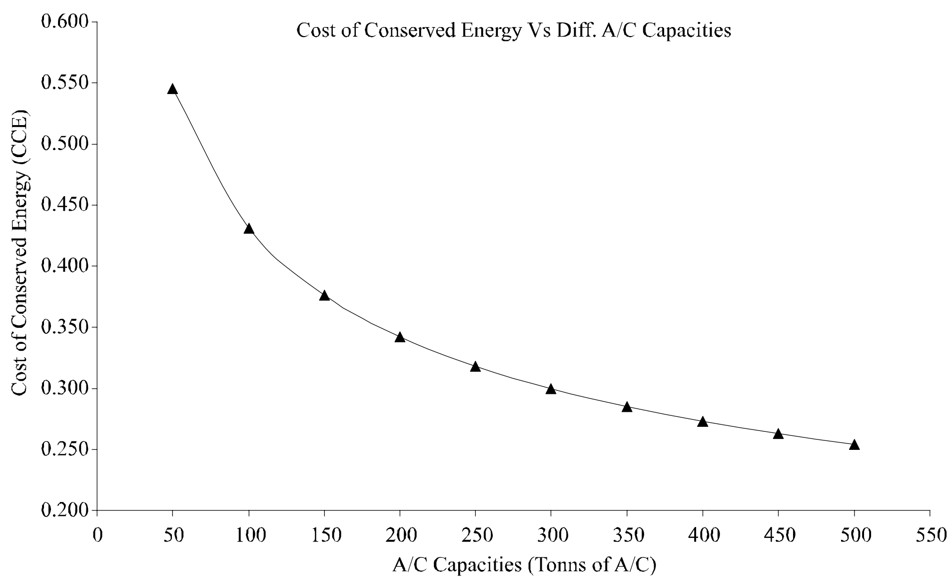

FIG. 7. COST OF CONSERVED ENERGY AT DIFFERENT CAPACITIES

(the desired maintained humidity). Fresh air (hot fluid) inlet temperature is varied from 35-45°C (308.2-318.2K) humidity is varied from 35-85%. Violation of constraints has been avoided by adding penalty terms. Previously mentioned upper and lower bounds are used.

With the given constraints and using the evolutionary optimization we achieve the convergence for different parameters.

4.2.1 Fitness Evaluation and Selection

The offspring individuals thus generated has been evaluated using the analytical model [13] and best five of these have been chosen as parents for the next generation.

Fig. 8 illustrates the convergence of the best fitness values with respect to the generations. It has been seen clearly that as the generations pass, the best fitness values of the individuals have been minimized. Figs. 9-11 illustrate the convergence of the individuals (dPf, dPr, rel_humidityf) to the optimal value. Each individual has been indicated by a cross. As illustrated in Figs. 9-11, the individuals have been scattered through out the entire design space for the first five generations, after which they start to converge and reach the optimal value.

The optimal values of dPf, dPr, rel-humidity determined using the evolutionary procedure are 0.5kPa,0.5kPa, and 85% respectively. These values are calculated for the 100

TABLE 2. SENSITIVITY TO THE ELECTRICITY PRICE

	C_{fuel}: 0.06 $/kWh	C_{fuel}: 0.12 $/kWh	C_{fuel}: 0.18 $/kWh
PCY ($)	761	1523	2284
TPC ($)	5146	10291	15437
Exdc/y (4)	404	808	1212
Texdc ($)	2730	54591	8188
TOPC ($)	7875	15750	23625
NetsavEnergy ($)	95213	221506	347839
Capcost ($)	31101	31101	31101
Payback Period (Months)	11	10	7
CCE (¢/kWh)	1.62	0.84	0.58

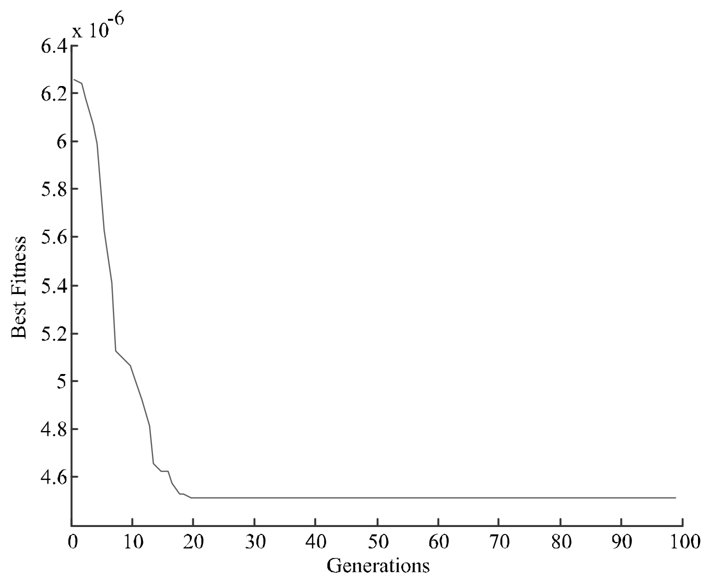

FIG. 8. CONVERGENCE OF FITNESS WITH RESPECT TO GENERATIONS

ton of air-conditioning load with volume flow rate 3.16 m^3/s. Maximum net energy recovered value is \$221,525 or 2215 \$/Ton.

Optimization has been carried out for different air-conditioning capacities (50-500 Tons) and compared per ton energy recovered for different capacities. Fig. 12 shows per ton energy recovered at different air-conditioning capacities at optimum parameters. It is clear that per ton net energy recovered is increasing with the increase in A/C capacity. It is due to the economy of the scale.

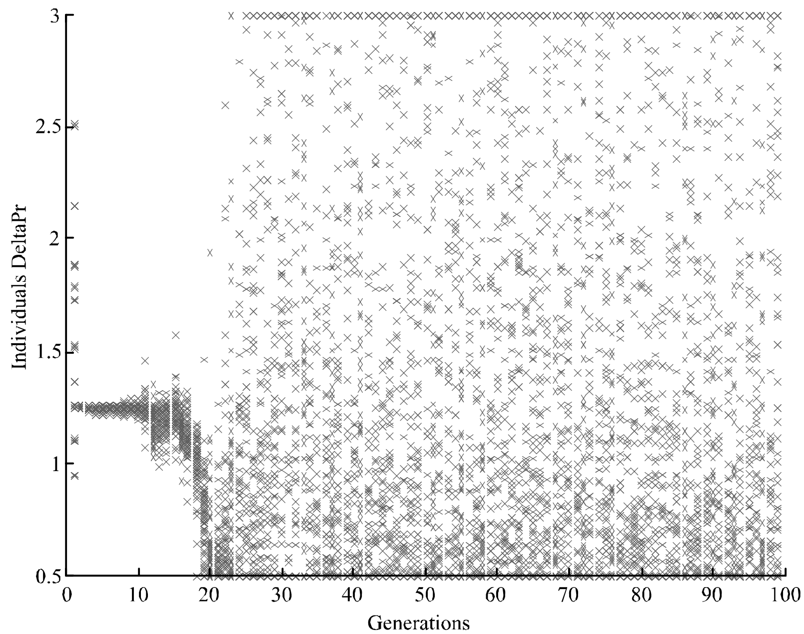

FIG. 9. CONVERGENCE OF POPULATIONS WITH RESPECT TO GENERATIONS OF ΔP_R

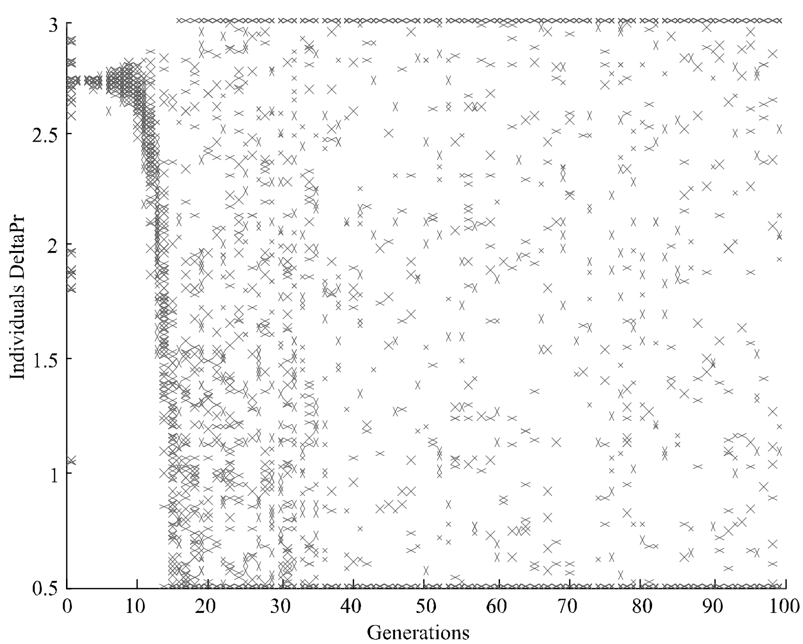

FIG.10. CONVERGENCE OF POPULATIONS WITH RESPECT TO GENERATIONS OF ΔP_F

4.3 Discussion

In the analyzed case, the cost of energy recovery system is not very large and results shows that it can be recovered in a year time. Although there was a capital investment of the system but as compared to the amount of energy recovered it is not significant. Net value of energy recovered per ton of air-conditioning is high for higher air-conditioning capacity, which is according to the economy of scale.

The cost of CRR is 6.4% for an interest rate of 10% and life cycle of 10 years for an energy recovery system. This is quite below the interest rate and therefore, it is feasible. CCE is varied from 0.55-0.25 ¢/kWh, which is much less than the electricity rate of 12 ¢/kWh. Therefore,

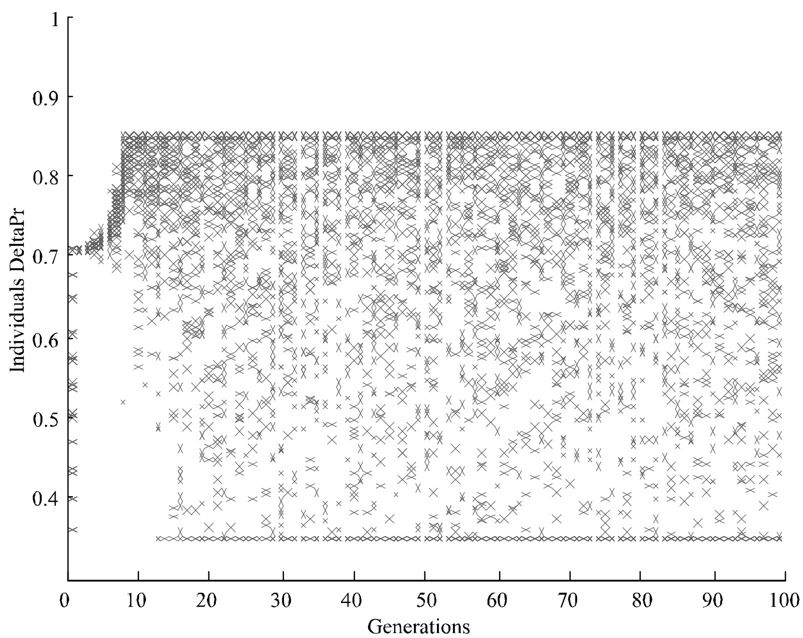

FIG. 11. CONVERGENCE OF POPULATIONS WITH RESPECT TO GENERATIONS OF FRESH AIR RELATIVE HUMIDITY

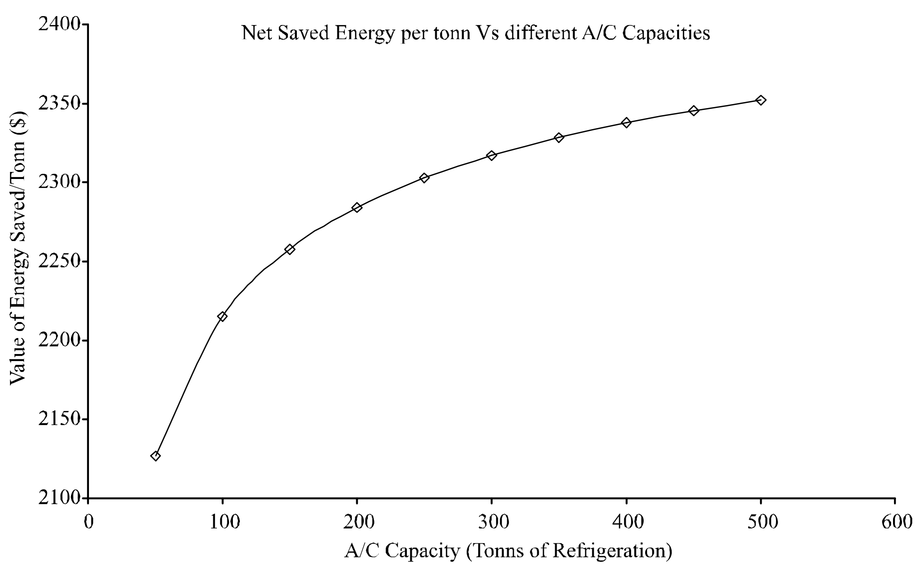

FIG. 12. NET ENERGY RECOVERED PER TON OF AIR-CONDITIONING FOR DIFFERENT AIR-CONDITIONING CAPACITIES AT OPTIMUM CONDITIONS

energy recovery system for central air-conditioning is a viable choice to save energy by reducing the compressor work.

Majority of the literature approaches to PFHE. Optimization is based on the assumption of simplified cost correlation as the one adopted in this study, owing to a difficulty of a detailed characterization of the manufacturing process. As a future work more detailed study is required to have manufacturing process effect on the PFHE cost, in order to have more realistic cost estimation for energy recovery system.

5. CONCLUSION

In this paper, a solution method of cross flow (both fluid unmixed) PFHE design optimization problem for energy recovery system in a central air-conditioning system is proposed based on the utilization of evolutionary algorithm. By doing the analysis on test case, cost of capital recovery is only 0.85 ¢/kWh which is quiet less to the cost of energy (12 ¢/kWh). Payback period is within the range of one year which is quiet feasible. Hence energy recovery system is economically viable. It also shows that energy recovery system is more viable for larger air-conditioning system compared to small air-conditioning systems. This shows the improvement potential of the proposed method for energy recovery system. Furthermore, the evolutionary algorithm allows for rapid solution of the design problem and enables to examine number of alternative solutions of good quality, and hence giving the designer more degree of freedom in the final choice compared to the traditional methods. For future work, study is required for detail mechanical design and the plate-fin heat exchanger manufacturing process to significantly improve the estimation capability of the capital investment to obtain a more realistic PFHE design and cost estimation to check the economic viability of energy recovery system.

6. NOMENCLATURE

A_f	Fresh air side surface area (m^2)
A_f	Return air side surface area (m^2)
b	Plate sapcing
C_{Pf}	Fresh air specific heat (kW/kg K)
C_{Pr}	Return air specific heat (kW/kg K)
C_{TOT}	Total Cost ($)
COP	Coefficient of performance
$C_{capcost}$	Capital investment ($)
C_F	Cost of energy ($/kWh)
CP_{PY}	Annual pumping cost ($)
LCC	Life Cycle Cost ($)
D_h	Hydraulic diameter
excf	Exergy change of fresh air (kW)
excr	Exergy change of return air (kW)
exd	Exergy destruction (kW)
f	Friction coefficient
G	Core mass velocity
h_r	Convective coefficient return air side (W/m^2K)
h_f	Convective coefficient fresh air side (W/m^2K)
I	Annual discount rate (%)
j	Colburn factor
k	Thermal conductivity of air
k_w	Thermal conductivity of metal
L3	Length of heat exchanger
Lrt	Return air side length of heat exchanger
Lft	Fresh air side length of heat exchanger
m_r	Return air mass flow rate (kg/s)
m_f	Fresh air mass flow rate (kg/s)
N	Life of heat exchanger in years
N_H	Annual operating time (h/year)
NTU	Net transfer units
OPC	Total operating cost
Pr_f	Prandtl number (fresh air side)
Pr_r	Prandtl number (return air side)
P_P	Pumping power (kW)
PPr	Return air pumping power (kW)
PPf	Fresh air pumping power (kW)
Pf	Fin pitch (m)

Re_s	Reynolds number (shellside)
Re_t	Reynolds number (tubeside)
Relhumf	Fresh air relative humidity (%)
Relhumr	Return air relative humidity (%)
T_{Fin}	Fresh air inlet temperature (K)
T_{Fout}	Fresh air outlet temperature (K)
T_{rin}	Return air inlet temperature (K)
T_{rout}	Return air outlet temperature (K)
T_{Fin}	Fresh air inlet temperature (K)
TER	Total Energy Recovered
U	Overall heat transfer coefficient (W/m^2K)
W_c	Compressor work
Wcy	Compressor work per year

GREEK SYMBOLS

δPr	Return air presure drop (kPa)
δP_f	Fresh air pressure drop (kPa)
η	Overal pumping efficiency
μ_r	Viscosity of return air (Pa.s)
μ_f	Viscosity of fresh aire (Pa.S)
σ	Corss sectional area of frontal side (m^2)

ACKNOWLEDGEMENT

Authors are grateful to Professor Dr. Ali El-Ghalban, Munayaf University, Egypt, for giving untiring help in completing this work.

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