Computational Model for the Optimization of Energy Consumption of non-residential buildings.

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Abstract. The intense process demand for evaluating new projects and retrofit second energy efficiency standards and sustainability parameters have attracted interest in the computer programs development to simulate accurately the buildings overall efficiency. Many simulation softwares that are often utilized in this process, including EnergyPlus and ESP-r, allows engineers and architects to analyses complex phenomenon of hourly or sub-hourly heat transfer. However, much software has restricted interfaces, in which the equations and parameters cannot be altered. This article proposes to describe the development of a computational tool, using the software *Engineering Equation Solver – EES*, to help designers minimize the consumption and costs of energy associated with single-zone HVAC systems of non-residential buildings. In addition, the optimization processes show the mutual influence of parameters on both the total cost and the thermal comfort.

Keywords: *Simulation, Optimization, Energy, Building, EES, HVAC*

1. INTRODUCTION

The intense process demand for evaluating new projects and retrofit second energy efficiency standards and sustainability parameters have attracted interest in the computer programs development to simulate accurately the buildings overall efficiency. Moreover, the energy consumption evaluation by end use of all charges and their demands are necessary information in building design and commissioning and / or retrofit. Given the large diversity of commercially available HVAC system, the intelligent building design, requires a careful the design and operating conditions analysis (AL-Homoud, 2001).

Many simulation softwares that are often utilized in this process, including EnergyPlus and ESP-r, allows engineers and architects to analyses complex phenomenon of hourly or sub-hourly heat transfer. Yet access to equations and intermediate results, these programs are wholly or partially restricted. Currently, the main difficulties of the simulation process are not computer power but on the understanding of the mathematical model of the user associated with that equipment should be analyzed (Gomes, 2005).

Today, the main difficulties of the simulation process are not computer power but on the understanding by users, the mathematical model associated with the equipment that should be simulated.

In an optimization problem has an objective function and constraints parameters, both related to decision variables (AL-Homoud, 2009).

The purpose of this study is to show the results of a computer program designed to minimize single-zone HVAC system consumption and energy costs of a nonresidential building - office. The program was developed in Software *Engineering Equation Solver - EES*. Solvers such as EES can be used to develop simulators fully adaptive and transparent since the equations can be written as a textbook.

The cooling loads for the model were calculated for design day for the climatic São Paulo conditions. The peak conditions were established according to ASHRAE Standard 90.1-2007 Appendix G for the following occurrence frequency: 99.6%, heating design temperatures and 1% dry-bulb and 1% wet-bulb cooling design temperatures.

This article demands special attention to model development and macro variables validation such as consumption and energy costs.

2. DESCRIPTION OF THE STRUCTURE COMPUTATIONAL MODELING

The computer program was developed with a set of simulation routines that work together to calculate the thermal and electrical demands (heating and cooling) in model representing a non-residential building.

Computational modeling began with a heat load fundamental theoretical concepts review, the heat transfer processes applied to buildings, simulation tools and computational methods. After the evaluation and study of theoretical concepts, three objectives were outlined:

- Develop a model in *EES - Engineering Equation Solver* to simulate a single zone HVAC system.
- Develop a model to simulate load sizing describing its static way.
- Develop model to simulate the single zone thermal performance to 8760h, describing its dynamic way.

2.1- Weather data

The weather file used by the model was structured by the International Weather for Energy Calculations (IWEC) as a result of the 1015 ASHRAE research to analyze weather data recorded by the U.S. National Climatic Data Center. "The model was simulated for the São Paulo city, southeastern Brazil (latitude: - 23.62 °, longitude: - 46.65 °; GMT: -3 h, elevation: 802 m). The weather data are entered into the model through two lookup table. An EES lookup table provides means of used tabular information in the solution of the equation. The first lookup contains hourly parameters for each of the critical elements needed during climate calculations (i.e. dry-bulb temperature, relative humidity, barometric pressure, wind speed, direct normal radiation, diffuse horizontal radiation and global radiation). The second lookup contains solar values for day depending on their position (i.e solar zenith angle).

2.2 - Building description and walls thermal models

The case study is performed for an office building located in São Paulo. The building consists of 10 rooms, one in each floor. Figure 1. shows a general plan considered for all floors of the office building. The 3 D geometric model and cooling system implemented in computer program are described in the table 1. below:

Table 1: Brief descriptions of the building and HVAC systems.

Dimensions	28 x 28 x 3.4 m		
Building type and stories	Office building, 10 stories above ground		
Walls	U_{wall} = 0.514 W/m².K; C = 14534 J/m².K (indoor insulation)		
Roof	U_{roof} = 0.318 W/m².K; C = 18170 J/m².K		
Floor	Floor U_{floor} = 0.039 W/m².K; ground coupling		
Windows	U_{window} = 1 W/m².K ; Solar Factor = 0.34 x 582 m²; South oriented		
Infiltration	0.3 ACH		
Internal gains	Occupancy	Lighting	Plug equipments
	7m^2/person from 9:00 AM to 9:00 PM	12W/m^2	20W/m^2
System	VAV-AHU - Central Cooling		

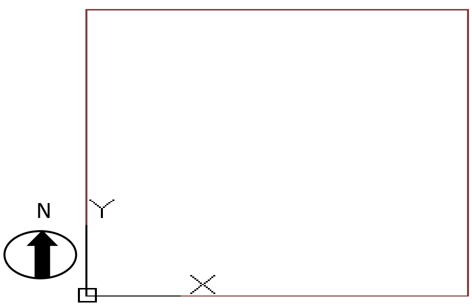

Figure 1. Single zone of the building.

In figure 1. the north orientation is noted by "N". The building facade is shows in figure 2. For defining geometry in computation model are necessary the floor area, floor-to-floor height and floor-to-ceiling height.

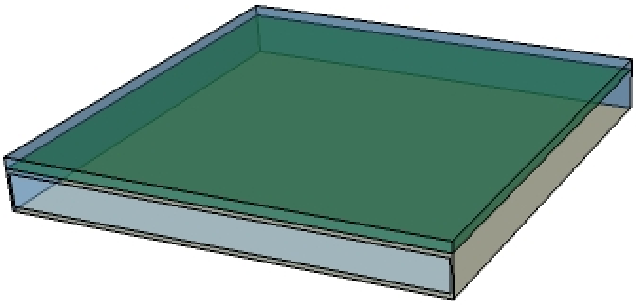

Figure 2. 3D model with the building facade

To model the surface we used the simplified model proposed for Laret (1981) in analogy to electrical circuit developed for Ngendakumana (1988).

The model includes heat transfer for infiltration - Rv, the heat exchange through the low thermal mass surfaces such as glass - R_0, and high thermal mass such as the external walls R_{21} and R_{22} with thermal storage capacity - C_2. The heat flow emitted by the surface of the wall (roof, floor, opaque frontages and windows), the enthalpy flow rate corresponding to ventilation and infiltration air and the internal sensible gains (including local heating/cooling and internal general gains) are summed, as in Eq.1, in order to compute the energy storage inside the indoor environment. This energy storage is computed by the means of a first order differential equation.

$$\frac{dU}{dt_{in}} = \dot{Q}_{roof,surf,in} + \dot{Q}_{floor,surf,in} + \dot{Q}_{opaque,front,in} + \dot{Q}_{windows} + \dot{H}_{s,vent} + \dot{H}_{s,\inf} + \dot{Q}_{s,in} \quad (1)$$

where:

U = *Energy storage*
t = *Time, in sec,*
Q = *Heat flow*
H = *Enthalpy flow*

subscripts:

in = *Indoor*
S= *Sensible*
vent= *Ventilation*
inf= *Infiltration*

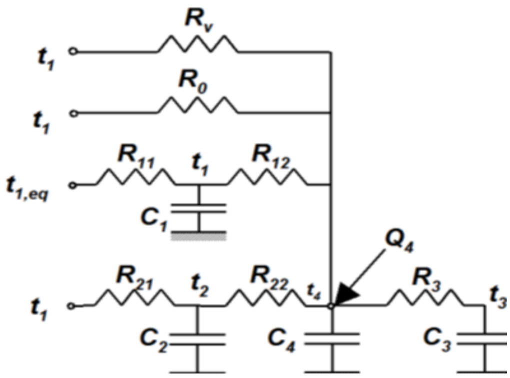

Figure 3. Graphical representation of the heat balance

2.3 - Model development

The computational program as described above includes simulation models for both the building and for HVAC system. These two modules interact with each simulation step (time-step) with a control module. The main processes and heat transfer phenomena involved in the model dynamic behavior are considered in the assessment of actual demand for cooling and heating.

In a central cooling plant, the overall system efficiency is directly influenced to both the equipment efficiency specified in the design and control operation. Remember that the main equipment in a facility such as: chillers, evaporators and condensing units, fan-coil and cooling towers are essentially heat exchangers. Many studies have been conducted to develop models to assess with precision and clarity the coils and heat exchangers in the process (Kusiak, 2009).

There are basically two models types to assess the heat exchangers: theoretical models and engineering models or empirical. The theoretical models are generally more complex and detailed and are based on heat and mass transfer relations. Moreover, these models require knowledge of a greater geometric factors number that generally are not available in the catalogs of the manufacturers.

Thus, crossing information on the total tubes length, transverse spacing between tubes, as well the tubes internal and external diameter, the exchange surface area can be determined and consequently the heat transfer flow.

The second model, used in developing this article demonstrates that it is possible to evaluate the heat exchangers and coils performance, under varying operating conditions, without knowing its geometric configuration. Applying the first law of thermodynamics for the fan coil the air temperature and flow can be determined by the equations 2. below:

$$\dot{Q} = \dot{m}_{water}\,.\Delta h_{water} = \dot{m}_{ar}.\ ff\,.\Delta T_{ar} \tag{2}$$

where:

m_{water} = *Water flow*
$m_{ar=}$ *Air flow*
ff= *Loss heat*

In the analysis presented here, the product of the heat transfer global coefficient to heat exchange area is determined for each cooling/heating surface at the design point. In this product are applied correction factors that take into account variations in the air and water flow, the thermodynamic properties and fluids transport involved and the performance curves of these parameters in manufacturer catalogs, as in Eq.3.

$$US_{coil,p} = US_{coil,d}\,.CF \tag{3}$$

where:

$US_{coil,p}$= Product of the heat transfer global to area in design point
$US_{coil,d}$= Product of the heat transfer global to area in off-design point
CF = Correction factors depending on the thermodynamic and transport properties

2.3 - HVAC system control

According to the heating and cooling demands required in a given period the coils will need more or less chilled water to meet the load. The chilled water control flowing through the coil and the cooling load modulation is affected by control valves installed at the right of the hydraulic network (McQuiston, 2005).

In all control systems there are at least three basic elements: sensor, controller and controlled device. Control actions can be classified as dual position or on-off and floating action or proportional. In the model developed in this paper operates with the coil chilled water flow proportional to the cooling load desired. The output variable in the control module is given by the following relationship, as in Eq. 4:

$$O = A + eK_p \tag{4}$$

where:

O = Output variable
A = Output variable with the system operating in nominal condition
E = Error (offset) is equal to the set point minus the measured value ($t_{setpoint}$ - *t*)
K_p =Proportional gain constant

Thus the value of the output variable will vary between the maximum value given for the parameter in the nominal condition and a number proportional to error.

2.4 - Optimization model

To establish an appropriate optimization strategy was necessary to formulate a problem to minimize power consumption and hence the energy cost including an objective function, constraints and the design variables number and type.

The module was developed to optimize the software environment *Engineering Equation Solver - EES*, which has its own language and has some routines for thermodynamic properties and mathematics functions, (Fisher, 2006).

The *Engineering Equation Solver - EES* has three optimization methods for multidimensional problems: Direct Search, Variable Metric or Genetic method. The genetic method was used in this study because it has a robust algorithm that allows the global maximum point location even in the presence of local optima. The genetic algorithm method is based on the mechanics of Darwin natural selection theory (Nabil Nassif, 2005). Since energy use and thermal comfort are the object functions in the muilt-objective genetic algorithm studied. This algorithm uses elite-preserving operator, which favors the elites of a population by giving them an opportunity to be directly carried over to next generation. After two offspring are creating using the crossover and mutation operators, they are compared with both oh theirs parents to select the two best solution among the four parent-offspring solutions.

2.4.1 - Problem Formulation

The optimization problem is to determine the setpoint values as a HVAC system control strategy. Setpoints must be optimized to minimize energy consumption for end use while maintaining building thermal comfort and unmet hours number for cooling demand within the limits set out in ASHRAE Standard 90.1-2007 Appendix G.

2.4.2 - Problem Variables

a) Zone temperature *setpoint;*
b) Pressure loss in air distribution network;
c) AHU supply air temperature *setpoint*;
d) Chilled water temperature s*etpoint*;
e) Outside air minimum flow as ASHRAE Standard 62.1-2007.
f) Comfort level as ASHRAE Standard 55-2004.

2.4.3 - Objective Function

As described above, the setpoints should be optimized to minimize energy consumption for end use while maintaining building thermal comfort and unmet hours number for demand cooling within the limits set out in ASHRAE Standard 90.1-2007 Appendix G.

The energy end use minimized includes the chilled water generation, ventilation, pumps and cooling tower, as in Eq. 5.

$$Min \sum_{i=1}^{t} (\phi_{i,cooling} + \phi_{i,fan} + \phi_{i,pump} + \phi_{i,tower}) \tag{5}$$

where:

ϕ_i =energy consumption for end use

2.4.4 – Constraints

The restrictions imposed on the optimization model are of diverse nature; design constraints, operational restrictions and regulatory restrictions.

a) The pressure loss in distribution network to ensure proper air flow in the zone. Thus the air flow rate obtained by the optimization must be equal or less than the maximum set by the following correlation, as in Eq. 6:

$$Q_{z\,Max} = Q_{z\,Max,design} * \sqrt{\frac{\Delta P}{\Delta P_{design}}} \tag{6}$$

where:

$Q_{z\,Max}$ =Optimized air flow

$Q_{z\,Max,design}$ =Maximum air flow design

ΔP =Dynamic load loss

ΔP_{design} =Maximum pressure loss design

b) Outside air minimum flow as ASHRAE Standard 62.1-2007 for office.

c) The unmet hours number for cooling demand should not exceed 300 hours of the 8760 simulated.

d) Zone uncomfortable hours percentage is within the range of this [5 – 12%].

3. DEFINITION OF INVESTIGATED CONTROL STRATEGIES

To evaluate the performance of the optimal control strategy of a typical HVAC system using the two-objective genetic algorithm, three different control strategies (A, B, and C) applied to the modified HVAC system are tested:

• Strategy A: no set point of item 2.4.2 has been optimized (*base case*)

• Strategy B: the setpoints of zone and supply air temperatures are optimized. The setpoint of zone can changes linearly within the 24°C to 27°C range and the setpoint of supply air temperature can changes linearly within the 13°C to 16°C range.

• Strategy C: the setpoints of zone and supply air temperatures and chilled water supply temperature are optimized. The chilled water supply temperature setpoint can changes linearly within 7°C to 18°C range above the optimized setpoints in strategy B.

Since the chilled water supply temperature setpoint in strategy B is constant at 6.3°C and in strategy C all setpoint proposed in item 2.4.2 has been optimized. The optimization problem is then solved using the one-objective optimization problem, respecting constraint and the thermal comfort criterion.

4. RESULTS OF STUDIED CONTROL STRATEGIES

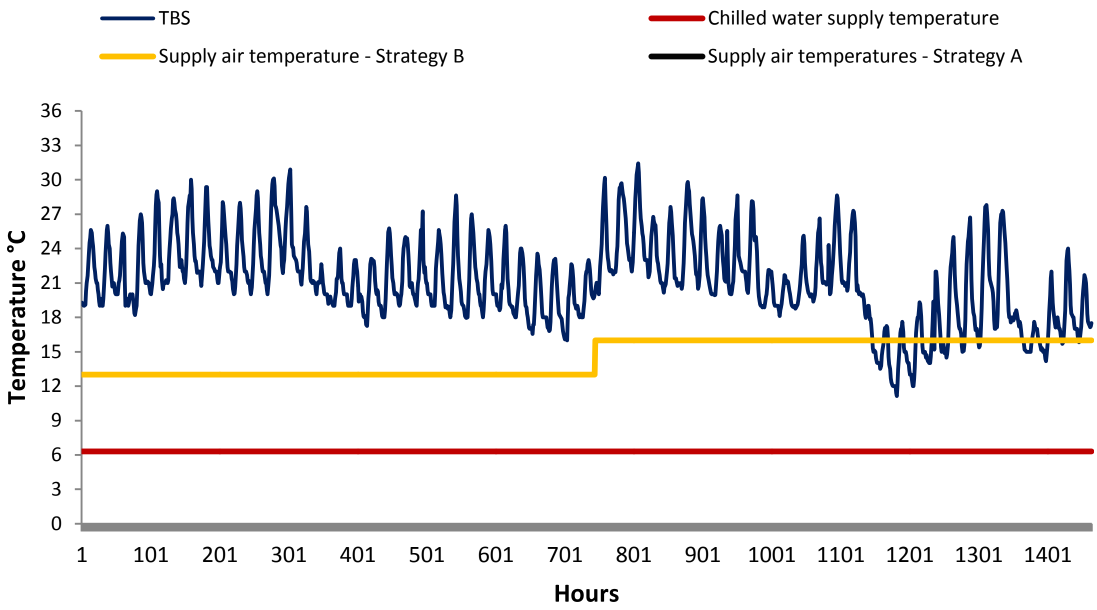

Figure 4. Optimal supply air temperature setpoint for period from march to april typical year.

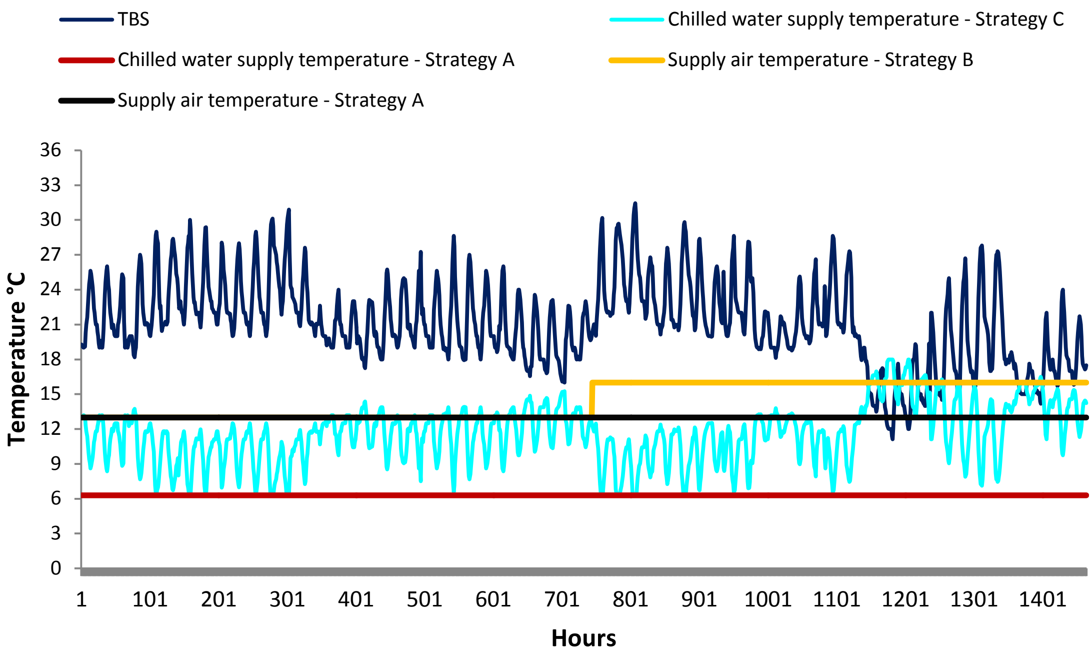

Figure 5. Optimal chiller water and supply air temperatures setpoint for period from march to april typical year.

The optimization of controller setpoints is done for strategy B and C under indoor sensible loads and outdoor conditions, as presented in Figure 4 and Figure 5, respectively. The two-objective genetic algorithm optimization program was run for 8760 h at each 15 minutes, using the genetic algorithm control parameters described above. To compare the scenarios using strategies B and C with that using strategy A, this strategy was selected as baseline. The results of the control strategies studied are illustrated in Figures 4 and 5 and table below. The figure 4 shows the optimization of the supply air temperature setpoint with the change external temperature (strategy B). The supply air temperature setpoint changes from 13°C to 16°C when the outdoor temperature decreases which reduces the 1.3% of the cooling load and 1% of the electricity. The figure 5 shows the optimization of the supply air temperature setpoint together with chiller water temperature setpoint (strategy C). The chilled water supply temperature setpoint can changes linearly within 7°C to 18°C range which reduces the 11.6% of the cooling load and 4.6% of the electricity. There was no energy savings when the setpoint of the zone could vary linearly within 24°C to 27°C for the São Paulo climate. This is because for an office load profile the heat that the zone receives at the close of the day increase the zone indoor temperature, and this heat must be removed by the HVAC system the next day. The figures 4 and 5 illustrate also the zone sensor sensitivity and AHU and chilled water sensor. It can be seen that the zone sensor (thermostat) is less sensitive to change outdoor temperature due to building thermal mass effects. It was observed that for all scenarios studied the thermal comfort level was kept within the acceptable range in 4% (350 hours).

Table 2. Time uncomfortable and reductions percentages in cooling and electricity for strategies B and C in relation A.

Parameter	**Strategy A**	**Strategy B**	**Strategy C**
Cooling [kWh]	372816.4	367923.6	329464.3
Electricity [kWh]	1350124.4	1338230.2	1288222.8
Time uncomfortable	350 (4%)	350 (4%)	350 (4%)
Reduction percentage [%]	---	1.3%	11.6%
Reduction percentage [%]	---	1%	4.6%

5. CONCLUSION

The proposed optimization process was applied to a single zone and HVAC system VAV-AHU - Central Cooling. The design of building and HVAC system is multi-parametric problem with both objectives and constrains. The objectives and constrains usually have different natures. The analyses indicate that the optimization of the control strategy with required constraints could improve the operating performance of the HVAC system. The results then show that by comparing strategy A and C, the optimization of a supervisory control strategy could save annual energy by 4.6 % while satisfying minimum zone airflow rates and zone thermal comfort. In addition, the studies show that the optimization supervisory control strategy to all controller set points, excluding zone temperature set point, performs better and provides more energy savings for the São Paulo city. The results indicate that the application of a multi-objective optimization problem could help control daily energy use or daily building thermal comfort.

6. REFERENCES

AL-Homoud, M. S. Computer-aided building energy analysis techniques. Building and Environment, [S.l.]: Elsevier, v. 36, p. 421-433, 2001.

AL-Homoud, M. S. Thermal Design Optimization of Buildings with Intermittent Occupancy, Journal of Building Physics, Vol. 33, No. 1, 2009.

Alpuche, Ma Guadalupe and Heard, Christopher – Exergy analysis of air cooling systems in buildings in hot humid climates, Elsevier, v. 25, p. 507-517, 2005.

Arenales, S. and Darezzo, A. – Calculo Numérico – Aprendizagem com apoio de software, São Paulo, Thomson Learning, 2008.

ASHRAE. 2007. 2007 ASHRAE Handbook—HVAC Applications, SI edition, chapter 40. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2007. 2007 ASHRAE Handbook—Fundamentals, SI edition, chapter 8. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Crawley, Drury B. and Hand, W. Jon – Contrasting the capabilities of building energy performance simulation programs, Elsevier, v. 43, p. 661-673, 2006.

EnergyPlus, 7/7/2010, Department of Energy, Energy Efficiency and Renewable Energy, Office of Building Technologies; www.energyplus.gov

Favrat D. and Marechal, F. and Epelly, O. – The challenge of introducing and exergy indicator in a local law on energy, Elsevier, v. 33, p. 130-136, 2008.

Fisher, Daniel E. and Taylor, Russell D. – A Modular, loop-Based approach to HVAC energy simulation and its implementation in EnergyPlus, Building Services Jornal, 2006.

Gomes, Ricardo Antonio do Espirito Santo, - Computation Model of Heat Recovery Steam Generators, Master Thesis, Faculty of Mechanical Engineering, UNICAMP, Campinas, 2005.

Kusiak, Andrew and Li, Mingyang – Cooling output optimization of and air handling unit, Elsevier, v. 33, p. 1 -9, 2009.

K. F. Fong, V. I. Hanby and T. T. Chow- HVAC system optimization for energy management by evolutionary programing: Elsevier, v. 38, p. 220-231, 2005.

Laurent Magnier and Fariborz Haghighat - Multiobjective optimization of building design using TRNSYS simulations, genetic algorithm, and Artificial Neural Network Building and Environment Volume 45, Issue 3, March 2010, Pages 739-746

McQuiston, Parker and Spitler – Heating, Ventilation and Air Conditionning – Analysis and Design – 6th ed., 2005.

Nabil Nassif, Stanistaw Kajl and Robert Sanbourin – Optimization of HVAC control system strategy using two-objective genetic algorithm, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. vol 11, n° 3, july 2005.

Neymark, J., and R. Judkoff. 2001. International Energy Agency Building Energy Simulation Test and Diagnostic Method for HVAC Equipment Models (HVAC BESTEST), National Renewable Energy Laboratory, Golden, Colorado, October 2001.