THERMOECONOMICS APPLIED TO BUILDING SERVICES

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Thermoeconomics is a blend of thermodynamics with economics. The thermodynamic analysis uses the second law and the concept of exergy, the measure of usefulness of energy. Economics involves costing exergy flows in life costing techniques. The objective of thermoeconomics is to minimise a cost function, taking into account capital, maintenance and running costs. Most of these are expressed in terms of thermodynamic variables of the system. This will establish the most cost effective design parameters. This paper presents methods for applying a thermoeconomic analysis to a building services system, consisting of displacement ventilation air conditioning. The specific cost of indoor environment cooling is optimised in terms of system variables, such as water and air temperatures and mass flow rates. Firstly, certain system parameters are optimised, providing detailed exergy costing related to the plant capital and operating costs. Secondly, by analysing these, the most appropriate design modifications are implemented, and the system is again optimised. From the work presented, an improved thermal and economic design results, with a reduction in Life Cycle Cost of 21%.

INTRODUCTION

The purpose of this paper is to introduce the use of Thermoeconomics (Lozano & Valero 1993, Lozano et al 1994) to Building Services, and to do so by means of an example. The analysis involves the use of exergetic life-cycle cost equations, in order to determine the optimum system design and operating conditions for the installation (Tozer et al 1995b). Appendix A can be referred to for a brief introduction to the concept of exergy and further references are available (Tozer 1995, Tozer & James 1995). In essence, exergy quantifies the usefulness of energy and has a positive value for cooling energy, which would have a negative value of energy. Energy is transformed and not destroyed, however exergy is always destroyed in real systems and the quantity of exergy destruction provides a measure of thermodynamic irreversibility, i.e. the extent to which it has deviated from ideal conditions.

Analysis of the system was undertaken and greatly facilitated using a computer, and involving the use of an equation solving software package (EES 1994). The software is able to operate on a variety of operating systems including windows and is very easy operate. It can solve a large number of simultaneous algebraic equations, limited only by hardware memory. To solve or to minimise / maximise a set of equations it uses an iterative method of approximations, and therefore appropriate guess values of all variables have to be provided together with their upper and lower values. The software also includes a wide range of mathematical and thermodynamic functions.

The displacement ventilation system used in the analysis consists of an air cooled reciprocating chiller, a chilled water system, an air handling unit (AHU), ductwork and extract fan. The impacts of improvements on the system are analysed. These consist of replacing the thermostatic expansion valve (TEV) for an electronic expansion valve (EEV), modifying the dehumidifying cooling coil to sensible cooling only and improvements to the ductwork configuration and fan efficiencies.

AIR CONDITIONING SYSTEM

The system to be optimised consists of an air handling unit (AHU) that is supplied with chilled water by an air cooled reciprocating chiller. The AHU supplies air at 19°C to the floor plenum of a building to provide displacement ventilation. Extract fans are used to exhaust the room air. A schematic diagram of the system is shown in Figure 1.

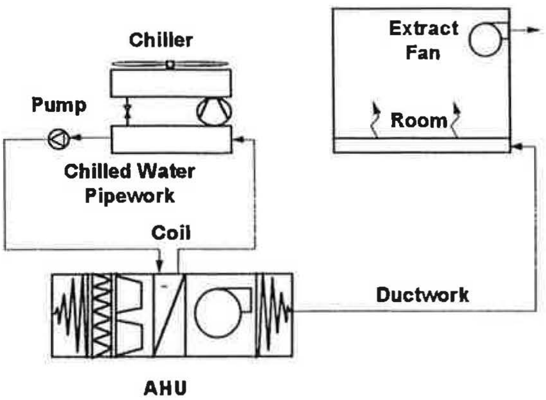

Figure 1: Displacement Ventilation Air conditioning

DESIGN CONSIDERATIONS

Standard design criteria for such a system would typically encourage relatively high leaving chilled water temperatures to maintain higher COPs in the chiller in view of a high temperature off coil condition. It would be reasonable to assume 17°C chiller entering chilled water temperature and 12°C leaving chilled water temperature and an off coil air temperature of 19°C for displacement ventilation. Due to the short running hours of cooling, pessimistic outdoor design condition were assumed at 30°C dry bulb temperature with a relative humidity of 45%.

DISPLACEMENT VENTILATION THERMOECONOMIC MODEL

A mathematical performs the same task as a purpose built physical model, but is constituted by mathematical equations. These should be correlated with experimental data, i.e. manufacturer's information and test data, and adjustments made to the equations to comply with the real situation. The number of equations provide more accuracy, although involving a more complex model. As an example a simple mathematical model of a fan's duty would consist of a quadratic or polynomial equation that fits the fan pressure-flow curve. If used in conjunction with the system pressure, the operating point is obtained by solving two equations with two variables, pressure and air flow.

The thermodynamic analysis uses the second law and availability (Moran 1989) or exergy (Wepfer 1979, Haywood 1991, Horlock 1992, Kotas 1995). Minimising the objective function will establish the most cost effective design parameters for the specific design configuration analysed. The plants modelled for this exercise were the chiller, chilled water pipework and pump, dehumidifying cooling coil, air handling unit, ductwork and extract fan. The mathematical model comprises a thermodynamic model which determines the functions regarding system variables and a thermoeconomic model which evaluates exergy and cost flows.

The breakdown of number of variables which are equal to the number of equations is:

- Thermodynamic model:		246 equations
- Chiller:	102 equations	
- Chilled water pipework / pump:	20 equations	
- Cooling coil	93 equations	
- AHU, ductwork and extract fan	31 equations	
- Thermoeconomic model:		55 equations
- Annual running costs:	6 equations	
- Capital costs:	15 equations	
- Life cycle costs:	6 equations	
- Exergy flows:	12 equations	
- Cost flows (exergy & capital costs):	16 equations	
- Total:		301 equations

The number of equations and variables would appear to be high, and for clarity reasons the chiller equations will be briefly described. A simplified refrigeration cycle can be represented by four conditions on the pressure - enthalpy diagram, but if all the components are considered together with their pressure losses and temperature changes the cycle is represented by nine conditions. Each of these requires up to five parameters including temperature, pressure, enthalpy, entropy, specific volume and quality. To model the chiller it is also necessary to consider all the heat exchangers and their subsections, i.e. the condenser has superheating, condensing and subcooling areas. Log mean temperature differences, heat exchange areas and heat transfer coefficients in terms of variable convection factors have also been considered. Compressor volumetric and isentropic efficiencies are further variables. The remaining systems were modelled with similar levels of complexity.

The chiller was modelled against published manufacturers' data (York 1994) and the design parameters were validated with their engineering department. The software package includes the thermodynamic properties of most refrigerants and the model was based on HFC 134a. For pipework and ductwork modelling, standard literature (CIBSE 1986) was referred to for Moody diagram equations. Cooling coils were modelled using the methods described in further standard literature (ASHRAE 1993).

The model philosophy considered the following:
- Electric energy was supplied by the grid and maximum demand charges were disregarded as only the cooling season is considered where these are normally not charged.
- The chiller was considered to remain constant and therefore the output varied according to the overall efficiency of the system.
- The design parameters considered as variables were both entering and leaving chilled water temperatures, pipework friction loss, number of coil rows and ductwork velocity.
- The chilled water pipework and air ductwork configuration were considered as constant and the pump and fan sizes varied in terms of their respective mass flow rates and pressure drops.
- The size of the AHU varied in terms of the air volume

COST ALLOCATION USING EXERGY

Exergy flows are determined throughout the system, starting from the exergy provided by the mains' electricity grid, and its value decreases throughout the system until the air is discharged to the outdoor ambient (exergy reference state) whereby the exergy value is zero. The costs are evaluated on a yearly basis and each plant item satisfies a cost equation based on exergy costs, i.e.: the costs entering a plant equals the costs leaving the plant.

EXergy flow cost in + Capital cost in + ENergy cost in = EXergy flow cost out [1]

In the case of the AHU the exergy flow cost in is that of the chilled water, the energy cost is the fan energy cost, the capital cost corresponds to the cost of the AHU and the exergy flow cost out is that of the cold air produced. All these costs have to be related to the same amount of time, i.e. one year.

$$c_i^b b_i eflh_i + c_i^w w_i eflh_i = c_{i+1}^b b_{i+1} eflh_{i+1} \qquad [2]$$

This equation only considers the exergy and energy flows and provides useful information on the relative efficiency of each plant comprising the overall system. The suffix i indicates entering the plant and $i+1$ indicates leaving the plant. The following equation provides useful information as it includes all costs involved (exergy, energy and capital costs).

$$c_i^{bz} b_i eflh_i + z_i crf \cdot f_{ma} + c_i^w w_i eflh_i = c_{i+1}^{bz} b_{i+1} eflh_{i+1} \qquad [3]$$

OPTIMISATION OF THE OBJECTIVE FUNCTION

Thermoeconomic optimisation consists of minimising an objective function, which is usually cost related to capital and running costs. The independent variables selected were the leaving chilled water temperature, chilled water temperature difference, pipework friction loss, number of coil rows and ductwork air velocity. In this case the chiller selected is constant and the output mass flow rate (m_a) cooled air to the building is variable. Therefore the objective function was established as the annual life cycle cost per mass flow rate of cooled air to the building:

$$llc = \frac{crf \cdot f_{ma} z + c^w \left(eflh_c w_c + poh_c (w - w_c)\right)}{m_a} n \qquad [4]$$

The first term of lcc represents the incidence of capital and maintenance cost whereas the second term represents the incidence of operating costs. A distinction has been made between the "equivalent full load hours" of the chiller compressor and the "plant on hours" of the rest of the plant including condenser fans, chilled water pump, air handling unit and extract fans.

OPTIMISATION OF DESIGN PARAMETERS

By using an equation solver software package, the set of 301 equations, and 301 unknowns was solved. The software enables functions to be minimised in terms of independent variables. In this case the yearly specific life cycle cost (lcc) was minimised in terms of the variables described. The optimum operating conditions were constrained by operating limits imposed by the chiller manufacturers. These were found to be the maximum leaving chilled water temperature of 14°C and the maximum chilled water temperature difference of 7.5K. Table 1 provides the main design parameters for the base condition, the optimised base condition and optimised improved design condition. It shows the result of the optimised variables in addition to the variable itself

Parameter	Base Design	Optimised Base Design	Optimised Improved Design	Comments
CHILLER Leaving chilled water temp °C	12	14	15.5	Variable for optimisation
Chilled water range K	5	7.5	9.5	Variable for optimisation
Compressor power kW	61.5	64.2	66.1	
Cooling output kW	201	220	233	
COP	3.27	3.42	3.53	
CHW PIPEWORK Pipe diameter mm	98	82	75	
Chilled water flow rate kg/s	10.67	7.78	6.53	
Velocity m/s	1.4	1.5	1.5	
Friction Pressure drop Pa/m	200	276	305	Variable for optimisation
Total pressure drop kPa	26	34.3	37.2	
Pump Power kW	1.8	1.1	1.1	
COIL Number of rows	3	4	8	
Sensible Heat Ratio	0.98	1	1	
Face velocity m/s	2.5	2.5	3.5	Variable of optimisation
Heat exchange area m2	452	854	1423	
AHU Filter pressure drop Pa	500	500	300	
Air flow rate kg/s	19.5	21.6	23	
Pressure Drop (ext.) Pa	1115	1045	930	
Fan motor size kW	31	32	28	
Width = Height of AHU m	2.63	2.77	2.41	
DUCTWORK Equivalent Diameter m	1.63	1.9	2	
Air velocity m/s	8	6.5	6.3	Variable for optimisation
Friction pressure drop Pa/m	0.3	0.17	0.15	
Ductwork static pressure drop Pa	282	185	85	
EXTRACT FAN Power fan motor kW	1.4	1.5	1.5	
OUTPUT % of Air flow rate	**100**	**111**	**118**	
LIFE CYCLE COST £/year-air flow rate	30686	27987	24355	
LIFE CYCLE COST %	**100**	**91**	**79**	

Table 1: Thermodynamic parameters

COST CALCULATIONS

The following Figure 2 indicates the specific exergy costing throughout the process from the chiller plant to the final product of building cooling, of the base system optimised as described previously. Two figures are given for each stage numbered 1 to 9, the c^{bz} figures which indicate the cost taking into account the exergy and plant costs, whereas the c^{b} figures indicate these costs in terms of the exergy costs alone. The term "z" refers to capital costs, "w" to electric power and for plant denominations, please refer to suffixes in symbols section.

ANALYSIS OF RESULTS

The analysis tool used was to determine the relative cost increase, between the exergy flow into and out of each plant. A careful analysis of these costs indicates that the cost of chilled water has the highest increase with respect to its incoming value. The next highest ratio is for the air handling unit. In order to improve the overall system efficiency an appropriate assessment of real achievable economic and exergy factors must be considered.

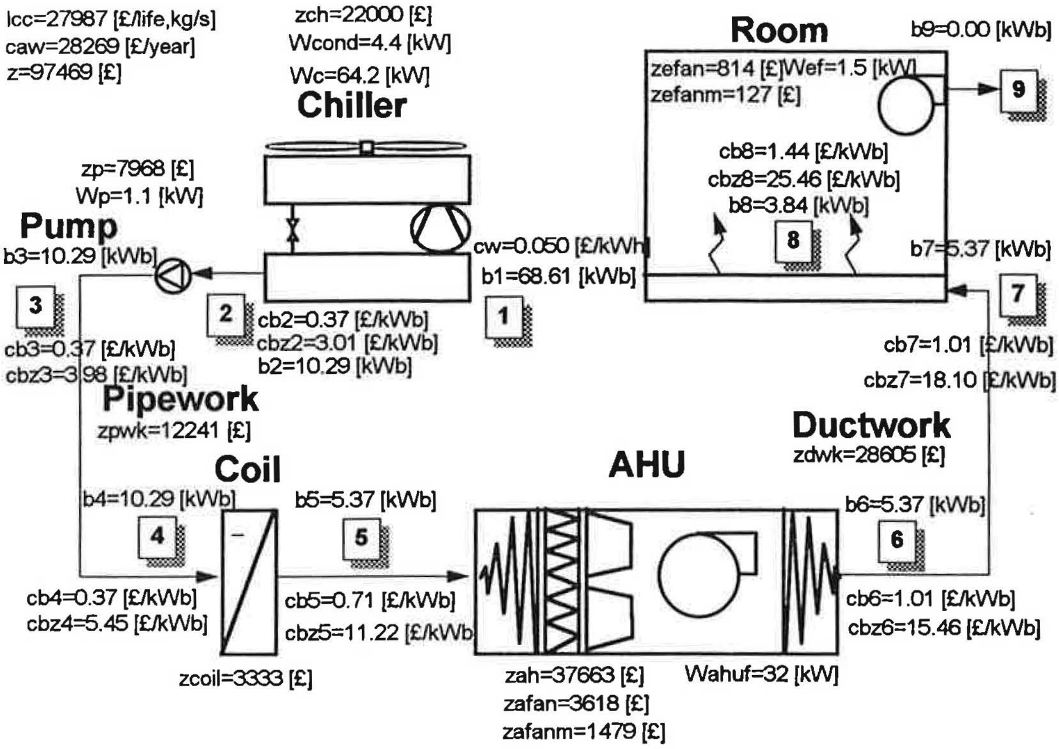

Figure 2: Diagram of optimised base design

- The optimisation of the chiller had been limited by parameters imposed by the manufacturer. The maximum leaving chilled water temperature of 14°C was limited by the thermostatic valve (TEV) that requires a minimum pressure difference between the condenser and evaporator. An electronic expansion valve (EEV) resolves this limitation. The manufacturer allowed to extend the temperature differential of 7.5K to a maximum of 10K, providing minimum flow rates were satisfied to guarantee heat transfer and that the compressor motor was not overloaded.
- By raising the chilled water temperatures the cooling coil was no longer dehumidifying, therefore the coil face velocity was allowed to float up as a variable of the optimisation process.
- In order to reduce the AHU operating costs, the specification of the filters was relaxed, thus reducing its pressure and the extent of the ductwork configuration was reduced.

Taking the above changes into account the system costs were recalculated and then optimised in terms of the variables described. The equation solver again minimised the value of *lcc* in terms of the same independent variables. The most relevant parameters are indicated in Table 2, where it can be noted that there has been a noticeable improvement in specific costs and in system performance.

Optimised Flow	Base Design Exergy kW (b)	Base Design Exergy Cost £/kWh (c^b)	Base Design Exergy and Capital Cost £/kWh (c^{bz})	Improved D. Exergy kW (b)	Improved D. Exergy Cost £/kWh (c^b)	Improved D. Exergy and Capital Cost £/kWh (c^{bz})
1	68.61	0.05	0.05	70.5	0.05	0.05
2	10.29	0.37	3.01	8.6	0.45	3.6
3	10.29	0.37	3.98	8.6	0.45	4.68
4	10.29	0.37	5.45	8.6	0.45	6.33
5	5.37	0.71	11.42	5.7	0.69	10.87
6	5.37	1.01	15.46	5.7	0.93	14.07
7	5.37	1.01	18.10	5.7	0.93	15.19
8	3.84	1.44	25.46	4.1	1.33	21.38
9	0	na	na	0	na	na

Table 2: Summary of Exergy and Flow Costs

CONCLUSIONS

A thermoeconomic analysis has been carried out on an air conditioning system which comprises a displacement ventilation system. The concepts of exergy and costs, therefore of thermoeconomics, have been applied to determine the internal costs of each process within the system. Using these, the internal costs has been a measure to evaluate on design modifications or improvements as these costs relate to both the plant capital cost and maintenance and to the fuel or energy costs.

By applying thermoeconomics to this case the cooling capacity of the overall system is elevated by 18%, whilst reducing the final specific energy cost by 25% (by comparing the initial and final figures of $c^{bz}{}_8$). Both of these are substantial improvements for the owner as the life cycle cost is reduced by 21%.

The practical application of this technique to Building Service is easily feasible, although initially laborious to develop the equations. It indicates the need to overcome limitations historically indicated by manufacturers in their catalogues for practical reasons. Also, it has the potential to model beyond the designer's original best judgement, providing a fundamental method for innovative and cost effective design.

SYMBOLS USED

b	=	Thermal exergy (kW)
c	=	Cost (£/kWh)
crf	=	Capital recovery factor
eflh	=	Equivalent full load hours (hours)
f	=	Factor
h	=	Enthalpy (kJ/kg)
llc	=	Life cycle cost per output (£/life kg/s)
m	=	Mass flow rate (kg/s)
n	=	Number of years (years)
poh	=	Plant on hours (hours)
Q	=	Thermal load (kW)
s	=	Entropy (kJ/kgK)
T	=	Absolute temperature (K)
w	=	Electric power (kW)
W	=	Work (kW)
z	=	Plant cost (£)
ϕ	=	Destroyed exergy (kW)

Suffixes

a	air
afan	ahu fan
afanm	ahu fan motor
ahu	air handling unit
ahuf	ahu fan
c	cooling / compressor
ch	chiller
cond	condenser
dwk	ductwork
efan	extract fan
efanm	extract fan motor
ma	maintenance
p	pump
pwk	pipework
s	entropy
0	reference state

Superscripts

b	exergy
z	capital cost

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APPENDIX A: INTRODUCTION TO EXERGY

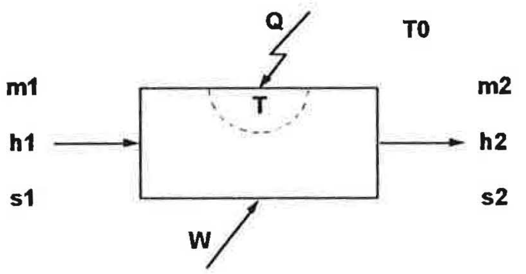

Figure A1: Thermodynamic system in stationary state

Consider a system that works in a stationary state (Lozano & Valero 1993). The balances of mass, energy and entropy are:

$$m_1 = m_2 = m \qquad \text{[A1]}$$

$$W = m(h_2 - h_1) - Q \qquad \text{[A2]}$$

$$\phi_s = m(s_2 - s_1) - \frac{Q}{T} \geq 0 \qquad \text{[A3]}$$

where ϕ_S is the generated entropy due to internal irreversibilities.

Note:

$\phi_S = 0$ implies a reversible process
$\phi_S > 0$ implies a irreversible process

Given T_0 as the ambient temperature, and combining the energy equation [2] with the entropy equation [3], and operating as [2]-T_0[3]:

$$W = m[(h_2 - T_0 s_2) - (h_1 - T_0 s_1)] - Q\left(1 - \frac{T_0}{T}\right) + T_0\phi_s \qquad \text{[A4]}$$

This equation provides the exergy balance of the system and all the terms in the equation have exergy units. The following terms can be identified, where the final term is zero for a reversible process

$m(h - T_0 s)$	exergy of flow
$Q\left(1 - \frac{T_0}{T}\right)$	exergy of heat
$T_0\phi_s$	exergy destruction

To produce a change from state 1 to 2 on the mass flow in a system that only exchanges heat with the ambient ($T = T_0$), a minimum amount of work will be required. It will be equal to the difference of exergy of flow between states 2 and 1 which is equal to $m[(h_2 - T_0 s_2) - (h_1 - T_0 s_1)]$, when the process is internally reversible ($T_0\phi_s$).

Another point of interest in refrigeration is to consider Q as the cooling capacity of a room at temperature T. The heat dissipated to the outdoor environment at T_0, is Q_0. The refrigeration plant works a cycle in a closed system. Applying the exergy equation [4] to this system:

$$W = Q_0\left(1 - \frac{T_0}{T_0}\right) - Q\left(1 - \frac{T_0}{T}\right) + T_0\phi_s \qquad \text{[A5]}$$

$$W = -Q\left(1 - \frac{T_0}{T}\right) + T_0\phi_s \qquad \text{[A6]}$$

From equation [6] it can be seen that the minimum amount of work required for refrigeration is $Q(T_0 - T)/T$ which corresponds to the Carnot reversible cycle ($T_0\phi_s$), where more work will be required for colder rooms.

3.3