ABSORPTION SYSTEM DIVERSIFICATION INFLUENCE ON TEWI

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Modern refrigeration technology including absorption refrigeration has coexisted with the more standard type of vapour compression systems ever since the beginning of refrigeration. Absorption refrigeration mainly requires a heat source to drive its cooling cycle. Only in the USA and Japan is absorption refrigeration widespread in the market, a market tendency of the last decades. Concerns regarding the Global Warming Potential and the more recently defined Thermal Equivalent Warming Impact (TEWI) is an opportunity to further explore the applications of absorption refrigeration. TEWI considers both the direct refrigerant effect and the primary energy impact on the equivalent carbon dioxide (CO_2) emissions. This paper explores these applications over a range of annual operating times, and therefore makes recommendations on absorption system diversification to reduce the TEWI.

List of symbols

B	Thermal exergy (kW)	c_p	Specific heat (kJ/kgK)
COP	Coefficient of performance	CO2	Equivalent emissions (kg CO_2/kWh)
eflh	Equivalent full load hours (hours)	E	Energy (kWh or GJ)
f	Factor	F	Exergy of Fuel (kW)
h	Enthalpy (kJ/kg)	HDR	Heat Dissipation Ratio
I	Indirect emissions (ton CO_2 / life)	k	Specific exergy consumption (kW/kW)
k*	Specific exergetic cost (£/£)	l	Refrigerant leakage and purge (%)
L	Leakage & release (ton CO_2/life)	m	Mass flow rate (kg/s)
n	Number of years (years)	N	Number of stages
poh	Plant on hours (hours)	p	Pressure (kPa)
P	Exergy of Product (kW)	Q	Heat (kW)
R	Recovery losses (ton CO_2 / life)	s	Entropy (kJ/kgK)
s	Refrigerant release (%)	t	Temperature (°C)
T	Absolute temperature (K)	W	Work or Electric Power (kW)
α	Refrigerant recovery factor	α	Thermodynamic ratio
Δ	Difference	ε	Utilisation factor
η	Efficiency	ϕ	Destroyed exergy (kW)

Subscripts

burn	burner	ch	chiller
chp	combined heat & power / cogeneration	comp	compressor
ct	cooling tower	cw	condenser water
e	evaporator	eflh	equivalent full load hours
g	generator (of absorption chiller)	g	grid (electric)
hw	hot water	m	motor
ng	natural gas	p	pump
poh	plant on hours	q	thermal
r	refrigerant	s	entropy
w	electricity	0	reference state

Superscripts

b	exergy	e	energy

1. Introduction

Any change in refrigeration technology by means of introducing new refrigerants or by adopting new techniques must be carefully balanced to reduce the overall environmental impact (Ure 1995, 1996a, 1996b). This has recently been classified as Total Equivalent Warming Impact (TEWI), in which the influence upon the Greenhouse Effect referred to as Global Warming Potential (GWP), can be judged for the operation of individual refrigeration plants (BRA 1996). TEWI calculations are based on two categories, namely Direct GWP and Indirect GWP. The Direct GWP is produced mainly by all halogenated refrigerants, including HFCs such a HFC 134a which is used in this paper as the base case with centrifugal chillers. The Indirect GWP is attributed to the indirect CO_2 emission due to the energy usage for the duration of its useful life. The amount of CO_2 emissions heavily depends on the type of electricity generation and power plant technology which varies from one country to another (Beggs 1994).

Refrigeration with absorption chillers is possible at evaporating temperatures ranging from 10°C to as low as -50°C with a variety of cycles and fluids. Absorption systems, which are heat driven, can both cool and heat for human comfort and process applications. Practically any type of heat source can be used to drive the absorption cycles, but this paper will only concentrate on the options mostly applied to the market, which are direct fired on natural gas, and indirectly heat driven via cogeneration systems (Tozer and James 1995). In addition to this the absorption chillers considered use the binary mixture water and lithium bromide where water is the refrigerant. By doing so, the direct refrigerant impact on TEWI is eliminated from these absorption options. Single and double stage units are used whereby the coefficient of performance (COP) is related to the number of stages and is defined as the ratio of cooling output with respect to heating input.

To allocate TEWI values to electricity and heat produced by cogeneration systems, it has been necessary to refer to thermoeconomic techniques (Lozano & Valero 1993). The thermodynamic component of such an analysis can conveniently be based on second law availability (Moran 1989) or exergy (Kotas 1995) concept, which is a measure of the usefulness of energy.

The results obtained by applying the theory to select absorption chillers based on TEWI calculations are presented. For completeness the appendices give a brief introduction to the concept of exergy and TEWI allocations.

2. Allocation of variables using exergy

Several theories have been described (Lozano et al 1994) regarding how the energy costs are to be allocated to the thermal and electric energy produced by the same Cogeneration plant, which consumes a fuel, i.e. natural gas. The main concept is that variables such as costs or TEWI values should be allocated in accordance with the usefulness of the energy produced by a cogeneration plant. Exergy is the thermodynamic variable that best quantifies the usefulness of energy.

2.1 Exergy

An introduction to the theoretical concept of exergy is detailed in Appendix A. However, exergy is not a wide spread concept and a couple of practical examples are given to provide an indicative guide of its use in practical situations. The usefulness of thermal energy depends strongly on its temperature. Depending on the fluid, temperature will determine its pressure.

For example, if only temperature is considered, the same amount of energy at different temperatures can be used in different ways. At 2000°C it can drive power stations with high efficiencies (0.45), at 800°C the efficiency is 0.15 at 400°C the efficiency drops to below 0.1.

With lower temperature applications, heat sources at 80°C can be used for direct or radiant heating. At 30°C heating is limited to fabric radiant heating and at 20°C no heating is possible because it is equal to the reference temperature. However, in Antarctica, which has a lower reference temperature, a heat source at 20°C could by useful.

The same concept applies to absorption chillers where the following approximate values are given (Tozer 1995, Tozer & James 1995b). Single stage chillers have a COP of 0.7 and require a heat source of 100°C. Double stage chillers have a COP of 1.1 but require a higher temperature heat source of 160°C. Triple stage chiller when they become available on the market with have COPs around 1.6 but will require higher temperature heat sources around 200°C or more. Again the concept of exergy is applied whereby more cooling can be achieved with more stages and higher temperature heat sources.

2.2 TEWI Allocation

The TEWI allocation was based on the specific exergy (note, not energy) costs of both thermal and electric energies being identical. This necessarily implies that the TEWI allocations of thermal energy will be lower than electric energy. The inverse of the exergetic efficiency (η) is the specific exergy consumption; the ratio of the exergy of the thermal and electric products (P) with respect to the exergy of the natural gas fuel (F) (Lozano & Valero 1993).

$$\eta^{b} = \frac{P}{F} \qquad [1]$$

$$k = \frac{F}{P} = \frac{1}{\eta^{p}} \quad [2]$$

Appendix B gives details of the TEWI allocation calculations using Exergy. The TEWI allocations for electric and thermal energy produced by a cogeneration plant are detailed below, in terms of the exergy efficiency, the Carnot factor and the emissions of natural gas supplying the cogeneration plant.

$$f_{carnot} = 1 - \frac{T_0}{T} \quad [3]$$

$$CO2_{wchp} = \frac{CO2_{ng}}{\eta^{p}} \quad [4]$$

$$CO2_{qchp} = \frac{CO2_{ng}\, f_{carnot}}{\eta^{p}} \quad [5]$$

3. Plant Design

The spreadsheet detailed in Table 1 provides information on different types of absorption chillers in order to determine in subsequent tables their TEWI based on exergy efficiency allocated to each case. The objective is to determine the chiller with the lowest TEWI based on only the chiller itself. The following chillers are considered (Tozer et al 1995):

Type of Chiller	Symbol
a) Single stage absorption chiller, hot water driven @ 85°C	A1,w85
b) Single stage absorption chiller, hot water driven @ 100°C	A1,w100
c) Single stage absorption chiller, hot water driven @ 115°C	A1,w115
d) Double stage absorption chiller, direct fired driven	A2,df
e) Double stage absorption chiller, steam driven	A2,st
f) Double stage absorption chiller, exhaust gas driven	A2,hr
g) Centrifugal compressor chiller, electric driven	Centrif

Table 1 provides design details of all the related plant items that vary in terms of the type of chiller, i.e.: hot water pumps, condenser water pumps and cooling tower.

The Heat Dissipation Ratio (HDR) is the ratio of condenser and absorber heat (if applicable) with respect to the evaporator heat (Tozer 1991, 1992). It is important because of the energy requirements to dissipate this heat through dry coolers of cooling towers. In the case of direct fired chillers the efficiency of the burner has also to be taken into account.

$$HDR = 1 + \frac{\eta_{burn}}{COP} \quad [6]$$

The Carnot factor indicates the quality of energy and is used for exergy costing, equation [3]. The main energy input to the chillers depends on the chiller capacity (evaporator heat) and COP. For absorber chillers the main heat input is to the generator (Q_e), whereas for centrifugal chillers, this is to the compressor.

$$Q_{input} = \frac{Q_e}{COP} \quad [7]$$

The exergy input results from applying the Carnot factor to the energy input.

$$B_{input} = f_{carnot} Q_{input} = \left(1 - \frac{T_0}{T}\right) Q_{input} \quad [8]$$

The mass flow rate of condenser water is calculated in terms of the chiller capacity, Heat Dissipation Ratio and condenser water differential.

$$m_{cw} = \frac{Q_e HDR}{c_p \Delta t_{cw}} \qquad [9]$$

The condenser water pump power is dependent on the flow rate, pressure drop and pump and motor efficiencies.

$$W_{cw} = \frac{m_{cw} \Delta p_{cw}}{\eta_p \eta_m} \qquad [10]$$

The cooling towers were selected for each case although a specific power requirement per heat dissipated could have been assumed. The total electric power of all plant is:

$$W = W_{ch} + W_{hw} + W_{cw} + W_{ct} \qquad [11]$$

For energy costs it is important to determine the hours of operation of each plant. For compressor power of centrifugal chillers and generator input to absorber chillers, Equivalent Full Load Hours (EFLH) were used. For all other electric equipment Plant On Hours (POH) were used. Therefore it was necessary to determine the power corresponding to both of these running hours.

DESIGN	Symbol	a)	b)	c)	d)	e)	f)	g)
Chiller Type		Absorp	Absorp	Absorp	Absorp	Absorp	Absorp	Centrif
Capacity kW	Q_e	2100	2100	2100	2100	2100	2100	2100
Stages	N	1	1	1	2	2	2	-
Energy Source		hw85	hw100	hw115	direct fired	steam	heat recov	electric
Temp. of heat source °C	t_{input}	85	100	115	1500	150	190	N/A.
Coefficient of performance	COP	0.695	0.7	0.705	1.05	1.2	1.1	5.5
Burner efficiency	η_{burn}				0.88			
Heat dissipation ratio	HDR	2.44	2.43	2.42	1.84	1.83	1.91	1.18
Reference temperature °C	t_0	25	25	25	25	25	25	25
Carnot factor	f_{carnot}	0.17	0.20	0.23	0.83	0.30	0.36	1
Energy input kW	Q_{input}	3021	3000	2978	2000	1750	1909	381
Absorption Th energy kW	Q_g	3021	3000	2978	2000	1750	1909	
Exergy input kW	B_{input}	506	603	691	1664	517	680	382
Auxiliary chiller power kW	W_{ch}	10	9	8	15	11	11	5
Centrifugal compressor kW	W_{comp}							382
Hot water pump kW	W_{hw}	2	2	2		1		
Cond water temp. diff K	Δt_{cw}	7	7	7	8.3	8.3	5.5	5.5
Condenser water flow l/s	m_{cw}	174	174	173	111	110	174	107
Cond water press drop kPa	Δp_{cw}	200	200	200	200	200	200	200
Cond water pump effic.	η_p	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Cond. wtr pump motor eff.	η_m	0.95	0.95	0.95	0.95	0.95	0.95	0.95
Cond wtr pump power kW	W_{cw}	52	52	52	33	33	52	32
Cooling tower fan kW	W_{ct}	60	60	60	40	38	60	38
ELECT POWER kW	W	124	123	122	88	83	123	457
Electric power @ eflh kW	W_{eflh}	0	0	0	0	0	0	382

Electric power @ poh kW	W_{poh}	124	123	122	88	83	123	75

Table 1: Plant Design Details

A preliminary analysis of this table indicates the following:

- The exergy input to single and double stage absorption chillers is approximately equal (excluding direct fired). In theory both single and double stage chillers are equivalent, i.e. single stage chillers have lower COPs but require lower grade energy than double stage chillers (Tozer 1995).
- Heat Dissipation Ratio: In practical applications this is the predominant factor in energy usage, where double stage absorption chiller benefits from lower Heat Dissipation Ratios than single stage chillers (Tozer & James 1995).

4. Energy

In Table 2 the energy is evaluated for a range of Equivalent Full Load Hours (*eflh*) and Plant On (running) Hours (*poh*). It was assumed that the plant on hours was 50% higher than the equivalent full load hours.

The table indicates the type Cogeneration system, i.e. engine or turbine, and therefore specific Cogeneration system costs are indicated in terms of the electric output capacity. The equivalent full load hours of Cogeneration annual operation is assumed at 5000 hours. The electric and thermal energies together with the total exergetic performances are indicated for each case.

ENERGY	Symbol	a)	b)	c)	d)	e)	f)	g)
Equivalent full load hours	eflh	1000	1000	1000	1000	1000	1000	1000
Plant on hours	poh	1500	1500	1500	1500	1500	1500	1500
Number of years	n	10	10	10	10	10	10	10
Cogeneration system		engine	engine	engine	none	turbine	turbine	none
Cogeneration elec. perform.	η_w	0.31	0.31	0.31		0.25	0.25	
Cogeneration exergy perf.	η^b	0.415	0.42	0.425		0.55	0.55	
Electric energy kWh	E_e	186588	184757	182930	132453	124823	184795	494786
Thermal energy MWh	Eq	3022	3000	2979	2000	1750	1909	

Table 2: Energy Calculations

The electric energy is based on both *eflh* and *poh* plants.

$$E_w = eflh \cdot W_{eflh} + poh \cdot W_{poh} \qquad [12]$$

Thermal energy is derived based on the equivalent full load hours of heat required to run the absorption chillers.

$$E_q = eflh \cdot Q_g \qquad [13]$$

5. TEWI Calculations

Table 3 provides the detailed calculations of TEWI for each case. The Cogeneration electric and thermal TEWI allocations are based on the electric and thermal performance amongst others. Refer to Appendix B for further details.

Thermal and electric energy TEWI values are allocated to each case depending on whether it is supplied via a Cogeneration system or through the electric or natural gas grids. The Carbon Dioxide (CO_2) emissions for natural gas and grid electricity are 0.18 and 0.53 kg CO_2/kWh respectively. (BRA 1996).

TEWI	Symbol	a)	b)	c)	d)	e)	f)	g)
Global Warming Potential	GWP							1300
Refrigerant charge	m_r							450
Annual leakage %	l1							2
Annual purge %	l2							0.5
Annual service release %	s1							0.25
Annual accidental release %	s2							0
Leakage CO_2 ton/life	L							161
Refrigerant recovery factor	α							0.95
Recovery losses ton/life	R							29
CO2 emissions/kWhe grid	Ceg				0.53			0.53
CO2 emissions/kWhe cogen	Cechp	0.43	0.43	0.42		0.33	0.33	
Electric Energy kWh	Ee	186588	184757	182930	132453	124823	184795	494786
CO2 emiss / kWh nat gas	Cng	0.18	0.18	0.18	0.18	0.18	0.18	0.18
CO2 emiss/kWh Cogen heat	Cqchp	0.07	0.09	0.1		0.1	0.12	
Indirect ton CO_2 / life	I	3005	3376	3700	4302	2100	2831	2622
TEWI ton CO_2 / life	TEWI	3005	3376	3700	4302	2100	2831	2812

Table 3: TEWI Calculations

The TEWI values were established for various Equivalent Full Load Hours and the results are represented in Figure 1.

Figure 1 indicates the following:

- Double stage steam driven absorption chiller. This option resulted in the best overall TEWI values for all EFLH and approximately 25% lower than those of the equivalent centrifugal chiller. This unit has an excellent COP due to that it uses the steam condensate heat to improve efficiency.

- Double stage heat recovery driven absorption chiller. Above 900 EFLH it resulted in marginally lower TEWI values. Due to the high initial costs of these units, it can not be recommended as a viable proposal (Tozer et al 1995).

- Single stage hot water driven absorption chillers. Within this group the best results were obtained for low hot water temperatures, i.e. 85°C as opposed to 100°C and 115°C. This is mainly due to the influence of temperature on the exergy allocation of TEWI values. This option is the most favourable in economic terms (Tozer & James 1996), but from the TEWI standpoint can only be recommended for low running hour applications below 500 EFLH.

Alternatively this option can be used for diversity arrangements of vapour compression and Cogeneration absorption chillers, where the later is used for the peak loads only.

- Double stage direct fired chillers: The TEWI values are over 50% higher than those of centrifugal chillers for most applications, i.e. EFLH higher than 500 hours. At 125 EFLH it does intersect the centrifugal TEWI line. Despite the environmental disadvantages in certain instances this option has proved to be economically viable (Tozer 1994). However, this option cannot be recommended on a TEWI basis unless the EFLH are below 100 hours as in a diversity arrangement of vapour compression and absorption chillers.

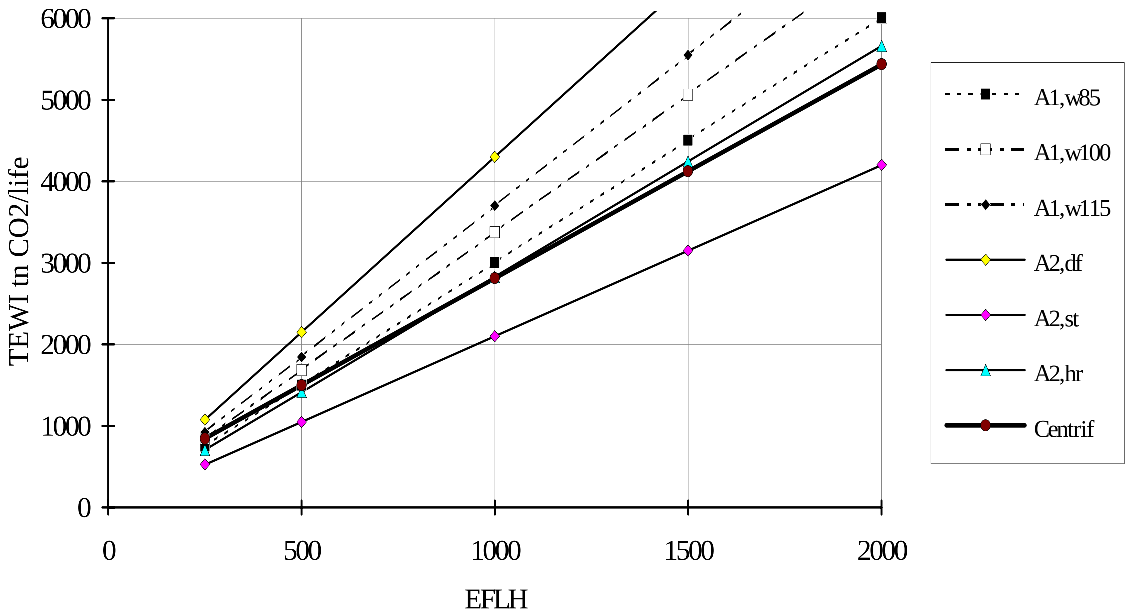

Figure 1: TEWI of Absorption Chillers

6. TEWI Calculations for Multistage Direct Fired Absorption Chillers

Due to the outstanding high values of TEWI for the direct fired double stage absorption chillers, a series of speculative calculations were carried out in order to predict the benefits to the environment of the direct fired, triple and quadruple stage absorption chillers. The following Table 4 summarises the basic theoretical and practical efficiency parameters.

STAGES	2	3	4
Thermodynamic ratio α	0.7	0.72	0.74
Burner Efficiency η_{burn}	0.88	0.90	0.92
Heat Dissipation Ratio HDR	1.84	1.62	1.50
Temperature of Generator t_g	150°C	190°C	230°
COP	1.05	1.45	1.83

Table 4: Parameters of Absorption Multistage Chillers

For theoretical cycles the thermodynamic ratio α is defined as the ratio of absolute evaporator and absorber temperatures, and also defines the ideal COP for single and multistage absorption cycles, where N is the number of stages (Tozer 1992).

$$\alpha = \frac{T_e}{T_a} = \frac{T_c}{T_g} \qquad [14]$$

$$COP = \alpha + \alpha^2 + \alpha^3 + \cdots + \alpha^N \qquad [15]$$

In this instance it was assumed that the practical thermodynamic ratio would increase as irreversibilities within the cycles are removed. However, they are still at a reasonable distance from the ideal value of approximately 0.9.

Based on the above the TEWI calculations were carried out in a similar manner to that described previously and the results are indicated in Figure 2.

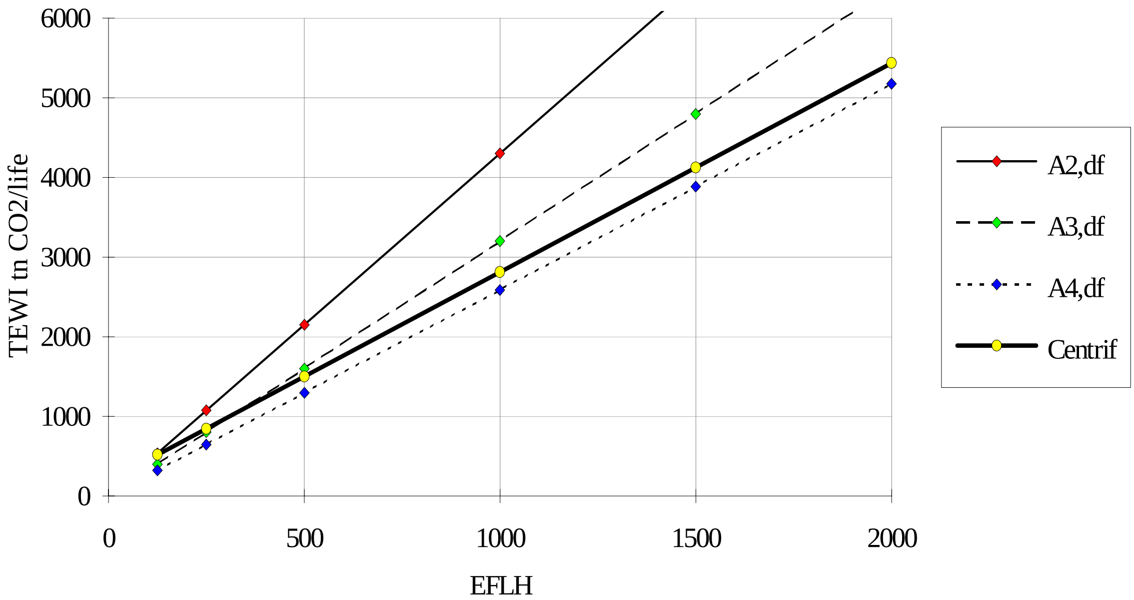

Figure 2: TEWI of Multistage Direct Fired Absorption Chillers

From the figure it can be seen that only the four stage direct fired absorption chiller can compete on a TEWI basis with centrifugal chillers. Three stage chillers, like double stage chillers can only be recommended for low EFLH applications, such as for short peak loads. The assumptions are that the ratio of Carbon Dioxide emissions of gas combustion and of electricity production remain the same, although they will both improve with time.

If these results are compared with those of other authors (Oak Ridge Laboratory 1996), it can be seen that the results are similar, although this paper indicates lower TEWI values for direct fired absorption chillers. The reason for this is likely to be due to the lower UK national electric production efficiency.

7. Conclusions

A Total Equivalent Warming Impact (TEWI) analysis has been carried out on the entire range of single and double stage absorption chillers to determine the most beneficial options. These included the direct fired absorption chillers and the indirect driven absorption chillers from cogeneration systems. These were based on exergy allocated values.

The best selections were found to be double stage steam driven absorption chillers for all applications.

The next best options are single stage hot water driven absorption chillers for short running hour applications and diversification where the absorption chillers are used only for the peak loads.

Within the same context double stage direct fired absorption chillers were found to be feasible for a very narrow and limited band of applications. Therefore a speculative calculation was carried out for multistage direct fired absorption chillers. This proved that, based on current technology, only the quadruple stage direct fired absorption chiller could compete on a TEWI basis with centrifugal chillers for a wide range of operating hours.

A clear indication on the absorption system diversification influence on TEWI has been provided together with guidelines to make this influence positive to the environment. These calculations are based on annual average CO2 emissions related to the electricity generation. Further work is required to ascertain the impact of daily or seasonal variations which have the potential to influence results significantly. A positive impact to the environment is mainly achieved by avoiding direct fired absorption chillers and using indirect driven absorption chillers used within integrated energy systems such as cogeneration.

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Appendix A: Introduction to Exergy

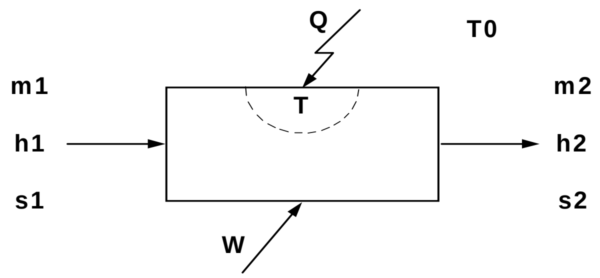

Figure A1: Thermodynamic system in a stationary state

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

$$m_1 = m_2 = m \quad [A1]$$

$$W = m(h_2 - h_1) - Q \quad [A2]$$

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

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 [A2] with the entropy equation [A3], and operating as [A2]-T_0[A3]:

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

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

$m\left(h - T_0 s\right)$ exergy of flow

$Q\left(1 - \frac{T_0}{T}\right)$ exergy of heat

$T_0 \phi_s$ exergy destruction

By analysing the exergy of heat the Carnot factor is derived, which is the ratio of thermal exergy to thermal energy:

$$f_{carnot} = 1 - \frac{T_0}{T} \quad [A5]$$

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 $m\left[\left(h_2 - T_0 s_2\right) - \left(h_1 - T_0 s_1\right)\right]$, when the process is internally reversible ($T_0 \phi_s = 0$).

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 [A4] to this system:

$$W = Q_0\left(1 - \frac{T_0}{T_0}\right) - Q\left(1 - \frac{T_0}{T}\right) + T_0 \phi_s \quad [A6]$$

$$W = -Q\left(1 - \frac{T_0}{T}\right) + T_0 \phi_s \quad [A7]$$

From equation [A7] it can be seen that the minimum amount of work required for refrigeration is $Q\left(T_0 - T\right)/T$ which corresponds to the Carnot reversible cycle, where more work will be required for colder rooms, in addition to the extra work due to irreversibilities ($T_0 \phi_s$).

Appendix B: TEWI Allocations to Cogeneration electrical and thermal energy

Unless otherwise specified the energy cost allocation considers that the specific exergy costs of both thermal and electric energies are identical. The same assumption is made with regard to allocating TEWI values to thermal and electric energy.

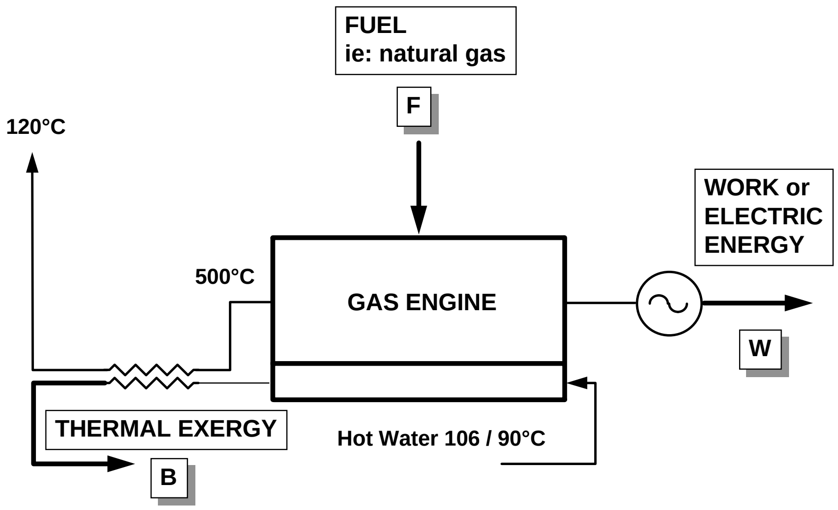

Figure B1: TEWI Allocation Schematic

The exergetic efficiency is the ratio of product and fuel exergies, which can be expressed in terms of the work produced and the quality of thermal energy produced.

$$\eta^b = \frac{P}{F} = \frac{W+B}{F} = \frac{W+Q\cdot f_{carnot}}{F} = \frac{W+Q(1-\frac{T_0}{T})}{F} \qquad \text{[B1]}$$

Also

$$\eta^b = \eta^b_w + \eta^b_q = \eta^e_w + \eta^e_q \cdot f_{carnot} \qquad \text{[B2]}$$

The inverse of the exergetic efficiency (η) is the specific exergy consumption (k), the ratio of the exergy of the thermal and electric products (P) with respect to the exergy of the natural gas fuel (F).

$$k = \frac{F}{P} = \frac{1}{\eta^b} \qquad \text{[B3]}$$

The specific exergetic cost (k*) considers the fuel cost:

$$k* = c_{ng}k \qquad \text{[B4]}$$

Therefore the electricity and thermal exergy costs supplied by the cogeneration plant are both equal to the specific exergetic cost (k*):

$$c^b_w = c^b_q = k* \qquad \text{[B5]}$$

By using the same concept to allocate TEWI values:

$$CO2^b_w = CO2^b_q = \frac{CO2_{ng}}{\eta^b} \qquad \text{[B6]}$$

To express these costs in terms of energy values, the electric exergy will remain unchanged because in this case energy is equal to exergy. In the case of thermal energy, the magnitude of energy will be the Carnot factor times larger than the thermal exergy, and therefore its TEWI value will be reduced by the same ratio:

$$CO2_w^e = \frac{CO2_{ng}}{\eta^b}$$ [B7]

$$CO2_q^e = \frac{CO2_{ng}\, f_{carnot}}{\eta^b}$$ [B8]

If a certain amount of the heat produced by the cogeneration plant is dumped, this will have an increasing effect on the thermal and electric energy costs. For this purpose the utilisation factor has been defined to vary between 0 and 1:

$\varepsilon_q = 0$ if all the heat is dumped

$\varepsilon_q = 1$ if all the heat is utilised (no heat dumped)

The exergetic efficiency is reduced by introducing the utilisation factor:

$$\eta^b = \eta_w^b + \eta_q^b = \eta_w^e + \varepsilon_q \eta_q^e \cdot f_{carnot}$$ [B9]

The reduction of exergetic efficiency increases the TEWI values which are therefore:

$$CO2_w^e = \frac{CO2_{ng}}{\eta_w + \varepsilon_q \eta_q^e f_{carnot}}$$ [B10]

$$CO2_q^e = \frac{CO2_{ng}\, f_{carnot}}{\eta_w + \varepsilon_q \eta_q^e f_{carnot}}$$ [B11]

If all the heat is dumped the following equation results which clearly produces higher TEWI values than equation [B10] when no heat is dumped.

$$CO2_w^e = \frac{CO2_{ng}}{\eta_w}$$ [B12]

<u>Example</u>

Given a gas engine cogeneration plant with the following details:

Electric energy efficiency	$\eta_w = 0.3$
Thermal energy efficiency	$\eta_q^e = 0.45$
Thermal energy temperatures:	90°C to 105°C
Full thermal energy utilisation	$\varepsilon_q = 1$
Reference temperature	t_0 = 28°C
CO2 emissions of natural gas	CO2=0.18 kg CO2/kWh natural gas

$$f_{carnot} = 1 - \frac{T_0}{T} = 1 - \frac{273.15 + 28}{273.15 + (105 + 90)/2} = 0.188$$

Therefore the TEWI of electric energy is:

$$CO2_w^e = \frac{CO2_{ng}}{\eta_w + \varepsilon_q \eta_q^e f_{carnot}}$$

$$CO2_w^e = \frac{CO2_{ng}}{\eta_w + \varepsilon_q \eta_q^e f_{carnot}}$$

$$CO2_w = \frac{0.18}{0.3 + 1 \cdot 0.45 \cdot 0.188} = 0.468 \text{ kg CO2 / kWh}$$

The TEWI of thermal energy is:

$$CO2_q^e = CO2_w \cdot f_{carnot} = 0.468 \cdot 0.188 = 0.084 \text{ kg CO2 / kWh}$$

When these values are compared to those of natural gas (0.18) and the electric grid (0.53), the TEWI advantages of Cogeneration become obvious.