ENVIRONMENTAL IMPACT OF SORPTION SYSTEMS

R. Tozer

Waterman Gore M&E Consulting Engineers, Versailles Court, 3 Paris Garden, London SE1 8ND, UK
South Bank University, Southwark Campus, 103 Borough Road, London SE1 0AA, UK

ABSTRACT

The environmental impact of sorption systems is related to the way electric energy is produced in each country, and to the system configuration for which sorption systems are designed. For this it is important to understand the theoretical limitations of absorption cycles, as there is a close relationship with those of real systems. The environmental impact of electricity production is itself a challenge, when such issues such as nuclear waste, and other contaminants are considered, in contrast to renewable fuels and hydro-electricity. In addition to this, the impact over the useful life of a sorption system needs to be considered in terms of the future trends of reducing the environmental impact of power production. These issues are discussed and proposals are made regarding life cycle environmental impact of sorption systems. In addition to this, there are different types of sorption technology (absorption, adsorption, chillers, heat pumps, heat transformers and hybrid systems), each of which has different energy ratios for cooling or heating energies, and therefore affect and impact the environment in different ways. A single-effect direct-fired chiller has a vastly different impact to an absorption heat transformer, or a CHP steam driven absorption chiller. These would be very different for countries which have predominantly nuclear, or hydroelectric or carbon fuelled electric generation. This paper addresses the theoretical and practical issues considering the market drive of existing and future technologies.

KEYWORDS

Absorption refrigeration, multi-effect, Carnot cycles, combined heat and power (CHP), cogeneration, environment, exergy, primary energy, tri-generation..

1.0 INTRODUCTION

Absorption cooling is many times claimed as an environmentally friendly alternative to mechanical compression systems. In many cases this is true, but special care should be taken as to how these are applied. Direct-fired absorption chillers offer environmental benefits in only some particular cases[1]. When applied to heat recovery systems there are in most cases environmental benefits, but care should be taken to avoid generalisations[2].

2.0 ABSORPTON REFRIGERATION

In order to appreciate the impact of multi-effect absorption chillers, which are usually of the direct-fired type, it is important to understand their theoretical limitations. It will be seen later that there is a correlation between the performances of real and theoretical ideal cycles, and therefore it is useful to understand these limitations.

2.1 Thermodynamics of absorption cycles

The ideal absorption cycle consists of a driving cycle and of a cooling cycle as described below[1,3,4,5,6,7].

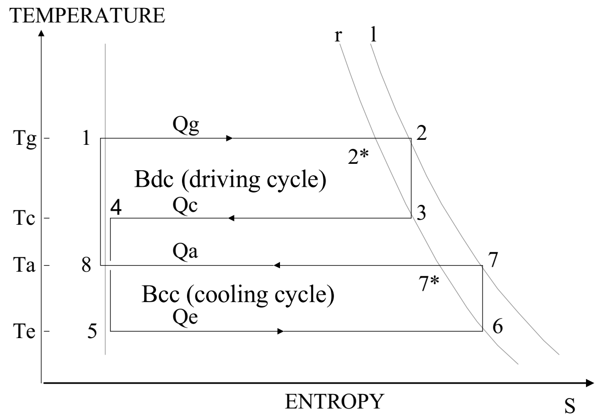

Figure 1: Ideal Absorption Cycle

The main heat transfer processes occur in the generator, condenser, absorber and evaporator.

The Coefficient of Performance (COP) is equal to the product of the cooling cycle efficiency and the cooling cycle COP:

$$COP = \eta_{dc} COP_{cc} \quad [1]$$

If expressed in terms of Carnot efficiencies:

$$COP = \frac{T_g - T_c}{T_g} \frac{T_e}{T_a - T_e} \quad [2]$$

As the same mass flow rate of refrigerant circulates through both cycles, these are linked by the following absolute temperature condition:

$$T_e T_g = T_a T_c \qquad [3]$$

Substituting the temperature relationship equation into the COP equation and simplifying gives:

$$COP = \frac{T_e}{T_a} = \frac{T_c}{T_g} = \alpha \qquad [4]$$

For double effect cycles the COP is:

$$COP = \frac{T_e}{T_a} + \left(\frac{T_e}{T_a}\right)^2 = \alpha + \alpha^2 \qquad [5]$$

For triple effect cycles the COP is:

$$COP = \frac{T_e}{T_a} + \left(\frac{T_e}{T_a}\right)^2 + \left(\frac{T_e}{T_a}\right)^3 = \alpha + \alpha^2 + \alpha^3 \qquad [6]$$

For N number of effects the COP becomes a geometrical series:

$$COP = \alpha + \alpha^2 + \alpha^3 + \cdots + \alpha^N = \alpha \frac{1-\alpha^N}{1-\alpha} \qquad [7]$$

For infinite number of effects the COP simplifies to the COP of the reverse Carnot cycle used for mechanical compression systems:

$$COP = \frac{\alpha}{1-\alpha} = \frac{T_e}{T_a - T_e} \qquad [8]$$

The equations for ideal cycles can be used to estimate real COP's based on the real single effect COP = 0.7. The table below summarises the ideal and real COP's for standard conditions, which indicates a good correlation. The Table is based on Te and Ta equal to 4°C and 40°C respectively. It also indicates the reduced benefit of increasing the number of effects.

Effect	COP ideal	COP real	Tg ideal	Tg real
1	0.88	0.7	81	100
2	1.67	1.2	127	150
3	2.36	1.5	179	210
4	2.98	(1.8)	237	(280)
∞	7.7	N.A.	∞	N.A.

Table 1: Comparison of real and ideal cycles

As the number of effects increases, increasing the COP, so does the requirement for higher temperature driving energy sources. For ideal cycles the relationship of temperatures is given by the following equation where N is the number of effects:

$$T_g T_e^N = T_c T_a^N \qquad [9]$$

For infinite effects the heat source temperature becomes infinitely high, with an exergy value equivalent to that of work required for mechanical compression systems.

2.2 Multi-effect cycles

It is clear from the above that absorption systems are equivalent to mechanical compression systems if the quality of the energy supplied (heat or work) is taken into account. Given that combustion temperatures are limited for direct-fired absorption chillers and the additional costs of increasing the number of effects, for the ever decreasing increment of COP, high multi-effect absorption cycles cannot be recommended for direct fired applications.

3.0 PRIMARY ENERGY RATIOS

Absorption cooling systems offer the possibility of environmental benefits, but not always. Global Warming Potential (GWP) and Total Equivalent Warming Impact (TEWI) relate to primary energy consumption. A popular measure is also the Primary Energy Ratio (PER), defined as the ratio of cooling energy (kW) to primary energy (kW).

Simple calculations on primary energy ratios of absorption chillers were carried out as part of the Annex 24 Country reports[8].

The following Table gives performance data for a selection of absorption machines for the UK country report[8]. The table has been sorted in order of primary energy ratio (PER). It should be noted that mechanical compression systems PER's would range from approximately 0.95 to 1.9 (based on COPs of 2.5 to 5). The allocation of primary energy to CHP heat supplied is based on the thermoeconomic cost theory (using exergy). However, note that the PER calculations in this case exclude the additional electric energy required for cooling tower fans and condenser water pumps. It should be considered in absorption machine comparisons and is addressed later in this paper.

PER	COP	Condenser water [°C]	Chilled water [°C]	Heat source & temp. [°C]

0.5	0.5	O.D. air ≅ 25°C	12-7°C	Natural gas direct fired
0.55	0.68	29-35°C	11-6°C	Boiler steam 1 bar
0.55	0.68	≅ 29-35°C	≅ 12-7°C	HW Boiler MPHW
0.95	1.2	28-33°C	12-6°C	Steam Boiler 8 bar
1	1	≅ 30-35°C	12-7°C	Natural gas direct fired
1.35	0.72	29-35°C	11-5°C	HW CHP 130-108°C
1.4	0.65	30-35°C	12-7°C	HW CHP 90-80°C
1.5	1.2	29-35°C	12-7°C	Steam CHP 7 bar

Table 2: Absorption Machines Performance Data

The case study reported by the UK for the Annex 24 referred to a double-effect steam-driven absorption chiller used with a gas turbine CHP system, applied to a pharmaceutical application[9]. In this case the additional electric energy required for the cooling tower fans and condenser water pumps was considered. The PER saving calculation was based only on the chillers and excluded the benefits of the CHP scheme. Although the PER reduction was around 3-4%, the PER benefits of the overall CHP scheme was much more significant. In this respect, it is important to ensure that the technical solution proposed is beneficial when studied in isolation.

Special care should also be taken with future trends and the impact of improving technology on the COPs of absorption and electric chillers, and the efficiencies of CHP systems and power stations. For example, over the last years the ammonia-water direct fired chillers have improved their COPs from 0.5 to around 0.63 with the introduction of GAX technology. On the other hand the efficiency of national electric supply has increased from 32% (1996) to 34% (1998). Although there is limited scope for improvements with conventional electric chillers and hot water or steam driven absorption chillers, there is scope for improvement for direct fired absorption chillers (triple effect, GAX variants). However, significant improvements can be expected from the national electric grid were modern power stations can have efficiencies in excess of 50%.

Cost effectiveness is related to increased system efficiency, and one way of achieving this is to apply absorption chillers to waste heat applications such as CHP systems. However the calculations to allocate primary energy of the CHP products requires the use of thermo-economic techniques[10,11,12]. This allocates costs or primary energy in proportion to the exergy (usefulness of energy) of its products, heat and power. Exergy is equal to energy for electric energy, however for thermal energy, exergy is calculated as the product of energy and the Carnot factor (1-To/T), where To is the reference temperature, say 298K (25°C) and T is the temperature of the heat source (K).

If similar qualities of chilled water are being analysed (say at around 7°C), the following PERs can be derived[13]. The suffix *b* refers to exergy, *source* refers to the source of energy (electric, or natural gas), *provider* refers to if energy is supplied from the national grid or CHP, and *user* refers to the final plant that uses the energy (chiller, pump, fan, etc). Although COP's refer to the main ratio of the main source of energy to a chiller in term of the cooling capacity, other COP's have been introduced to consider other sources of energy required. For example the suffix *hdr* (heat dissipation ratio = Q cooling tower / Q chiller = 1 + 1 / COP) refers to the energy required by the cooling towers and condenser water circuits[5]. See Appendix A.

Mechanical compression chiller (electric)

$$PER_m = \frac{Q_c}{F_{ps}} = \frac{P_{ps}}{F_{ps}} \frac{Q_c}{W_c + W_{hdr}} \qquad [10]$$

In this case the product from the power station is equal to the work of the compressor and heat dissipation plant.

$$PER_m = \frac{\eta_{ps}^{w}}{\frac{1}{COP_m^{w}} + \frac{1}{COP_{hdr}}} \qquad [11]$$

Direct-fired absorption chiller

$$PER_{abo}^{df} = \frac{Q_c}{F_{ps}} + \frac{Q_c}{F_{ng}} = \frac{P_{ps}}{F_{ps}} \frac{Q_c}{W_{abo} + W_{hdr}} + \frac{P_{ng}}{F_{ng}} \frac{Q_c}{F_c} \qquad [12]$$

In this case the product from the power station is equal to the work of the absorption chiller internal pumps and heat dissipation plant. Also the natural gas product (net at site destination) is what is gas energy consumed by the absorption chiller.

$$PER_{abo}^{df} = \frac{\eta_{ps}^{w}}{\frac{1}{COP_{abo}^{w}} + \frac{1}{COP_{hdr}^{w}}} + \frac{\eta_{ng}^{q}}{\frac{1}{COP_{abo}^{q}}} \qquad [13]$$

Absorption chiller applied to CHP

$$PER_{abo}^{chp} = \frac{P_{chp}}{F_{chp}} \frac{Q_c}{B_{abo}^{q} + W_{abo} + W_{hdr}} \qquad [14]$$

In this case the product (in terms of exergy) from the CHP plant is equal to the exergy of thermal energy required to drive the absorption chiller, plus work of the absorption chiller internal pumps and heat dissipation plant.

$$PER_{abo}^{chp} = \frac{\eta_{chp}^{w} + \eta_{chp}^{q} f_{carnot}}{\frac{f_{carnot}}{COP_{abo}^{q}} + \frac{1}{COP_{abo}^{w}} + \frac{1}{COP_{hdr}^{w}}} = \frac{\eta_{chp}^{b}}{\frac{1}{COP_{abo}^{equivalent}}} \qquad [15]$$

Analysing the previous PER cases, it is possible to derive these into the following generalised equation:

$$PER = \sum_{source} \frac{\eta_{provider}^{b}}{\sum \frac{1}{\eta_{user}^{b}}} \qquad [16]$$

4.0 INTEGRATED SYSTEMS' EFFICIENCY

When considering an absorption chiller with a CHP system, it is necessary to analyse the entire system[15]. However, each of these should be optimised before integrating into one system.

4.1 Sorption chillers

There is a general and instinctive objective to increase the COP of the absorption cycle, which does not necessarily mean that the thermodynamic efficiency improves[14]. The COP can be improved by increasing the number of effects, but these also require higher grade energy to drive them. In terms of exergy efficiency double effect machines have lower efficiencies than single effect machines.

An alternative way to improve the thermodynamic and exergy efficiency of an absorption machine is reduce its internal irreversibilities, which will in turn reduce its driving heat source temperature, say from 90°C to 70°C.

Further development of new generation absorption chillers optimised for operation in cogeneration plants is needed. In order to achieve this, the classical design of the absorption chillers should be reviewed very critically (i.e. by a detailed analysis of the irreversible phenomena in a chiller).

This would provide the chillers optimised to make best use of the heat transfer area and give higher COP's using very low water return temperatures (say 70°C). Theses temperatures would open up a range of new energy sources on engines such as: the jacket cooler, oil cooler, inter-coolers and recovery boiler[13].

By their design nature adsorption chiller avoid many irreversibilities absorption systems have (heat exchange in desorber and adsorber), although they have others that absorption do not have (inertia of desorption beds). The net result so far is that although adsorption systems have lower COPs than absorption systems, they can be driven by lower heat source temperatures (say 70°C) than most absorption chillers, with very small exceptions[1].

4.2 Combined Heat and Power (CHP)

When describing the efficiency of CHP systems, there is a tendency in the market to add up the work and heat outputs to quote overall high-energy efficiencies of say 80%. This is quite misleading, because with this comparison a steam turbine with 10% electric efficiency and 70 % thermal efficiency, is just as efficient as a gas engine with 32% electric efficiency and 58% thermal efficiency. This is obviously far from being a fair comparison.

If the different energy qualities involved in the work and heat produced are considered the results would be best expressed in terms of exergy efficiency. If so the efficiency of the steam turbine would probably be 60-70% of the gas engine CHP system.

This is a concept very well developed by the science of Thermo-economics, which uses exergy as true measure of usefulness of energy. Exergy (B) can be simplistically defined as Energy (Q) times the Carnot factor (1-To/T), where To is the reference temperature, say 298K (25°C), and T is the temperature of the heat source[10,11,12].

$$B = Q\left(1 - \frac{T_0}{T}\right) \qquad [17]$$

The same basic concept was addressed by the UK CHP Quality Assurance program[16]. The principles proposed by the CHP QA address the concept of different values for thermal and electric energy. For each type of CHP plant a weighting factor "X" and "Y" is allocated to the annual electric and thermal energy efficiency respectively. The electric (or thermal) efficiency is the ratio of annual electric (or thermal) energy produced, to the fuel energy consumed by the CHP plant. The sum of these weighted efficiencies has to be over 100 to achieve the Quality Assurance standard required.

The following table illustrates the Carnot factors for different heat source temperatures and also the ratio of Y/X factors from the CHP QA document. The ratio of Y/X indicates by how much less valuable is thermal energy compared to electric energy, and should ideally be comparable to the Carnot factor.

Heat Source	Carnot factor	CHP QA Ratio of	**Comments**

Temp. °C	(1-To/T)	Y / X	
60	0.11		
100	0.20	0.65 *	1 effect absorption chiller COP=0.68
170	0.33	0.57 **	2 effect absorption chiller COP=1.20
500	0.61		

Table 3: Carnot factors and CHP QA ratios

Note: Values for Y and X from CHP QA program[17]
Reciprocating engine Y/X = 130/200 =0.65
Gas turbine 1-10 MWe Y/X = 125/200=0.57

5.0 FUTURE TRENDS

The work of the IEA HPP Annex 24 has been concluded with the publication of the final report "Ab-sorption machines for heating and cooling in future energy systems"[1,2,17].

Most of the sorption market at this time comprises direct-fired absorption chillers, or hot water or steam absorption chillers indirectly driven by direct-fired boilers. In addition, the report covers absorption (reversible) heat pumps, absorption heat transformers, compression-absorption heat pumps, and adsorption chillers and heat pumps. Adsorption systems together with desiccant systems are also addressed.

An analysis of the market factors shows that the market pull favours sorption technologies in different ways. Direct-fired absorption chillers are installed in areas where there is lack of mains electricity, or restrictions on using it to power electric-driven mechanical compression chillers. A chart has been compiled (See Figure 2) to illustrate the different market pull factors. It shows, in decreasing order of market pull [indicated by horizontal arrows on the right hand side of the Figure], the following technologies:

Direct-fired or boiler-driven absorption chillers [indicated as "Chiller (DF)"]; and
Absorption chillers driven by waste heat, heat recovery or combined heat and power (CHP) systems [indicated as "Chiller (HR)"].

Far less prominent are:
Absorption heat pumps [indicated as "AHP (DF)"], competing with boilers and with electric heat pumps, then
Absorption heat transformers [indicated as "AHT (HR)"]; and finally
Adsorption chillers [not indicated in the Figure].

The Figure shows that environmental benefit [indicated downwards on the vertical axis] is inversely proportional to the market pull [indicated upwards on the vertical axis] and market share of sorption technologies. Although the lack of knowledge of sorption technology by technicians, engineers, and professionals is an important barrier, the main market barrier is considered to be the relatively high first costs of sorption plant.

In practice, different technologies are found to be most suitable for different countries, mainly depending on their energy infrastructure and particularly on how the country's electricity is produced. An energy infrastructure based mainly on coal-fired generation should favour sorption or other non-electric systems over compression systems. On the other side of the spectrum are countries that have a more renewable energy basis, which should favour electric-driven systems. The history of generation shows that energy production generally improves over time, which affects the benefits sorption technology can offer.

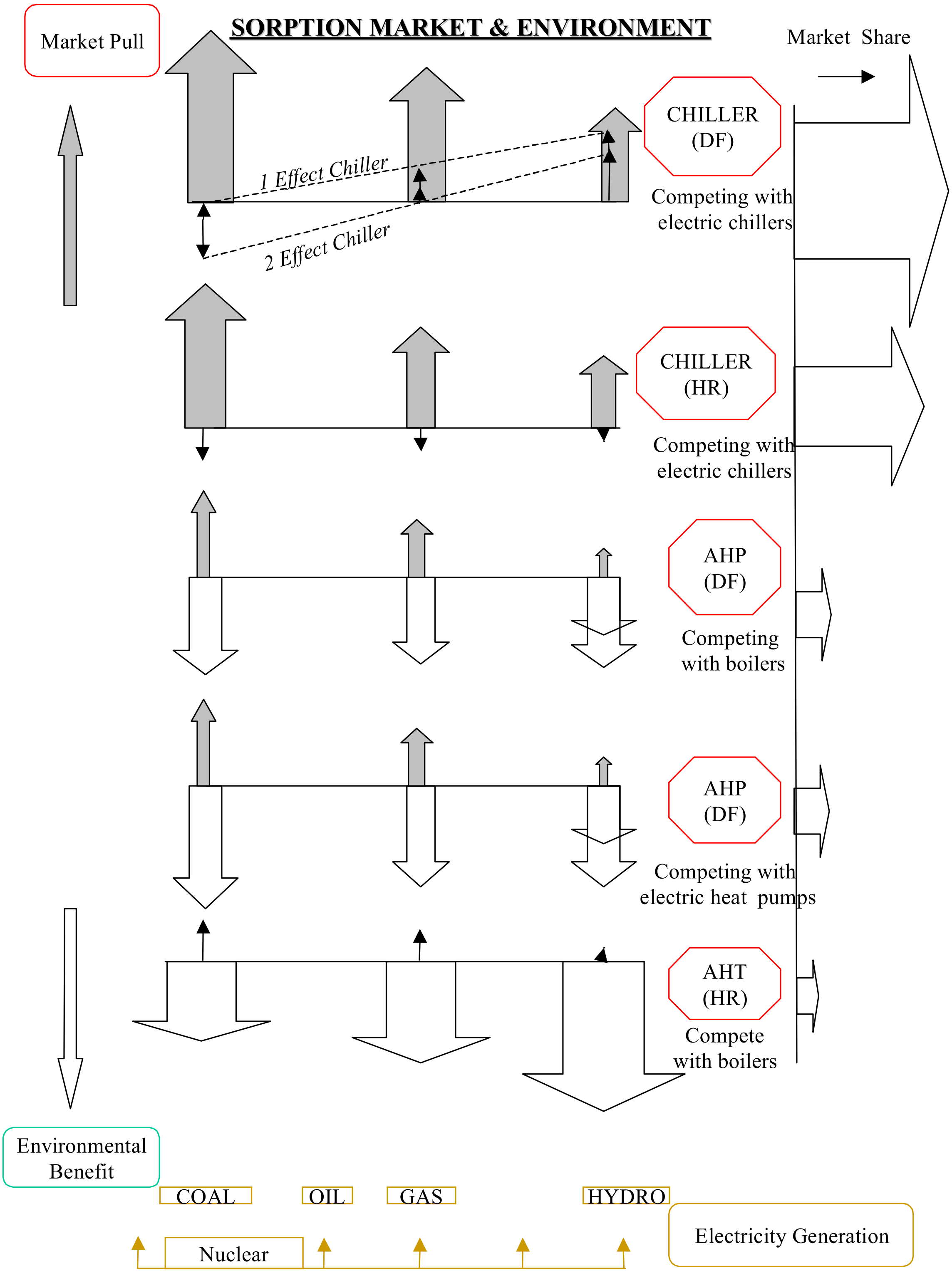

Figure 2: Sorption market and environment.*

* The area of the arrows (current market share, market pull, environmental benefit) indicate the relative magnitude of the variables. The length and width of the arrows does not provide any additional information.

6.0 CONCLUSIONS

Application of sorption technology cannot always be claimed as the best choice for the environment[1,2,17].

Encouragement should be given to absorption and adsorption chillers using waste heat, heat recovery or applied heat;
Sorption chillers applied to CHP systems have an existing market pull, and benefit the environment. However, the overall efficiency has to be relatively high with respect to each nation's power production, throughout the life of the system;
Absorption heat pumps (including reversible heat pumps) will be available on the market in the short term, and due to their environmental benefits should be strongly supported, unless electric heat pumps are more beneficial, as occurs in countries with a high proportion of hydroelectricity and even gas energy; and
Absorption heat transformers and compression-absorption heat pumps offer excellent environmental benefits for industry. However, the latter are likely to have more market pull in the medium or long term;
Direct-fired chillers should be phased out where they do not prove to be environmentally beneficial.

Technically, it is suggested that the main emphasis of future work could be on sorption applications of waste heat / heat recovery and process heat, but other factors need equal or greater attention. They include the study of the effect of existing taxes and economic incentives, and formulation of recommendation for such measures; system capital cost reduction studies; the closer study of environmental benefits and new potential markets.

Given that the environmental benefit of sorption technology is inversely proportional to its market share, policies should be set up to tax those less environmentally friendly to assist in financing those where more benefits to the environment are achieved.

NOTATION

B Thermal exergy (kW)
COP Coefficient of performance
eflh Equivalent full load hours (hours)
F Fuel (kW)
HDR Heat dissipation ratio
l Constant concentration line (%)
N Number of effects
P Produce (kW)
PER Primary Energy Ratio (-)
Q Heat or Cooling (kW)
r saturated liquid line
S Entropy (kJ/K)
T Absolute temperature (K)
W Electric power / power (kW)
α Thermodynamic ratio
η Performance

Subscripts
a absorber
abo absorption chiller
c cooling, condenser
carnot Carnot factor
cc cooling cycle
chp combined heat & power
dc driving cycle
e evaporator
g generator
hdr heat dissipation ratio
m mechanical chiller
ps power station
ng natural gas
0 reference state

Superscripts
b Exergy
chp Combined heat and power
df Direct fired (chiller)
q Thermal energy
w Work, electric

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APPENDIX A

Heat Dissipation Ratio

The Heat Dissipation Ratio (*HDR*) is the ratio of heat dissipated from the cooling tower (condenser and absorber heat) with respect to the chilled water (evaporator heat) load. This relation is necessary to analyse the cooling water requirements for different types of absorption and mechanical compression (centrifugal / screw / reciprocating) chillers.

$$HDR = \frac{Q_c + Q_a}{Q_e} \qquad [A1]$$

The heat flows crossing the absorption system boundaries (evaporator, absorber, condenser and generator) are:

$$Q_g + Q_e = Q_a + Q_c \qquad [A2]$$

This equation neglects the energy of the absorption chiller refrigerant and solution pumps. It is a very useful tool to verify data supplied by manufacturers.

The Coefficient of Performance (COP) is defined by:

$$COP = \frac{Q_e}{Q_g} \qquad [A3]$$

Therefore the Absorption Cycle *HDR* is defined as:

$$HDR = 1 + \frac{1}{COP} \qquad [A4]$$

If a similar analysis is been done for the mechanical compressor (vapour compression) cycle, the same *HDR* equation is derived.

From this simple relationship it can be seen that high *COPs* as with mechanical compression systems have the benefit of lower Heat Dissipation Ratios, therefore of less cooling water requirement. This is an additional advantage for the double effect unit with respect to the single effect absorption cycle. This in turn, means smaller condenser water pumps, domestic water pumps, condenser water pipework and cooling towers together with less water loss due to drift and evaporation and therefore lower chemical requirements.

Both the capital costs and operating costs of the Heat Dissipation Ratios need to be taken into account when considering the economics of absorption systems.