# Trigeneration using Absorption Chiller Technology

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Cogeneration and trigeneration are matters of great current interest. Recently a number of norms and legislative measures have been introduced which are apt for promoting extensive use of these technologies and consequently significant application developments are expected. The topics examined in this report concern small cogeneration plants, that is to say those covering electrical power generation up to 1 MW, which use for that purpose internal combustion engines. In particular it considers the complications which arise when adopting, alongside the cogeneration engines, absorption chillers, being equipment items which permit one to convert the heat recovered from the cogeneration system into cooling energy suitable for various uses such as industrial processes and space air-conditioning. The chillers employed for this purpose are characterised by a cooling capacity of up to 176 kW, are of modular design and compatible one to another, which allows the installation of several units together to meet different cooling capacities. To energise absorption chillers one can use hot water at relatively low temperature values, between 70 °C and 95 °C.
They produce chilled water at an outlet temperature of 7 °C, with a conversion efficiency of 70%.
Some specific suggestions aimed at obtaining the maximum energy output performance from the combined system are formulated hereunder.
It is also worthwhile to point out that this plant-system technology, otherwise known as trigeneration, has already, for some time, become amply and gainfully widespread in northern European countries, which are particularly susceptible to all proposals regarding the efficient use of energy.

## The Concept of CHP and CHCP

The term cogeneration refers to the combined production of heat and power (CHP), obtained by the consumption of primary energy. This primary energy may be gas or diesel fuel used in an internal combustion engine to drive an electricity generator. A great proportion of the heat lost by the engine in producing power is recovered and is itself employed for other purposes.

In the past, it was common to use on-site electric power as emergency, stand-by facilities, in case of failure of the utility supply from the grid, or as the only means of electrical power in those cases where no supply was available from an utility. In such situations no great importance was placed on the overall output efficiency of the system: figure 1 illustrates the efficiency which is characteristic of an electricity generation plant which involves no heat recovery from the generating engine.

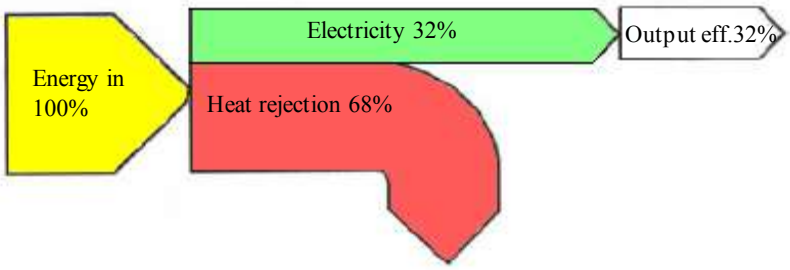



*Figure 1 - Efficiency of traditional electric power generation without recovery of any of the waste heat from the engine*

Present day costs of primary energy, the costs of purchasing electricity and the overall efficiencies which can be obtained, if heat recovery from a generator unit is implemented, have completely changed the whole installation concept adopted: businesses today are opting for installation solutions targeted at drastically reducing costs. Engine efficiency obtainable, converting shaft power to electrical power, is in the order of 32%, which means that the jacket cooling water and gaseous exhaust products of the engine normally reject from the system almost 70% of the potential energy embodied in the primary energy fuel. Typically also, the efficiency of heat recovery from the engine waste heat can be in the order of 80%.
The efficiency of CHP systems can be expressed as follows:

$$Te = \frac{Ep + (Pe - Ep)\,er}{Pe}$$

where:

Te = Total efficiency
Ep = Electric power generated
Pe = Primary energy employed
er = Efficiency of the heat recovery system adopted

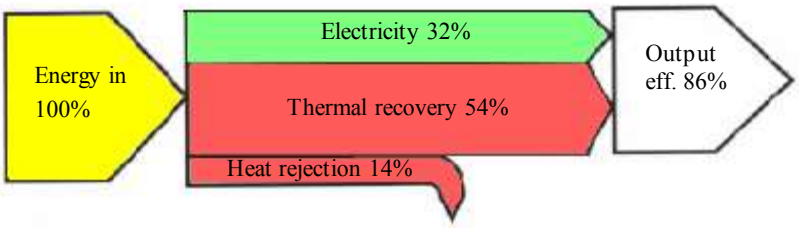



*Figure 2 - Efficiency of a CHP unit in which the waste heat from the engine is recovered for heating purposes or as sanitary hot water*

The various values in question are put into perspective by figure 2. The foregoing clearly demonstrates that instead of utilising a fuel to produce electricity, at an efficiency of 32%, the recovery and use of the waste heat realises, an increase of 54% to an efficiency value of 86 %
However a continuous need for all of the recoverable waste heat of the engine must be present throughout the periods that electricity is generated. Should only a proportion of this heat be required, the overall

efficiency of the CHP would be penalised. Figure 3 illustrates the declining efficiency of the system as a function of the lack of utilisation of the heat available.

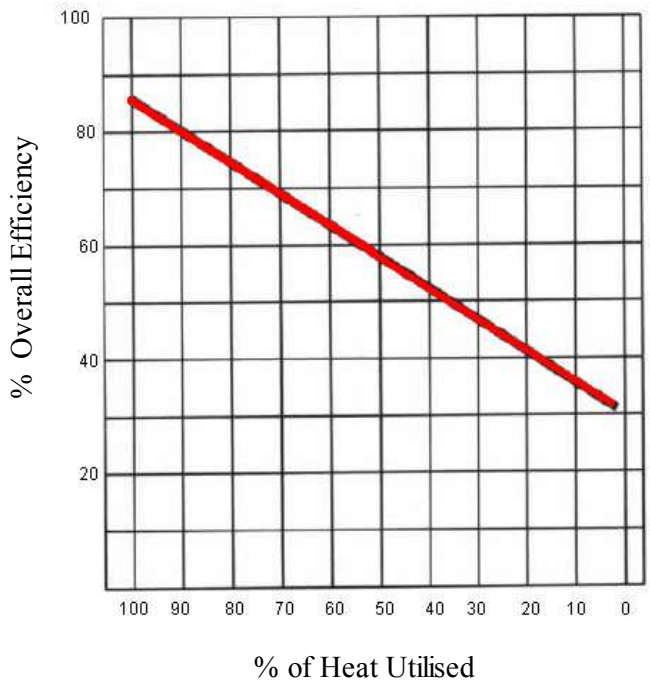



*Figure 3 – Overall efficiency of a CHP unit as a function of the amount of waste heat recovered and utilised.*

Since in order for a cogeneration system to be viable, taken for granted the generation of electricity, there must be a simultaneous requirement for the heat available: one could also say that, given a convenient usage for the heat, the electricity generated must be suitably employed.
The critical factor for the cogeneration system is thus the conjunction of the heat and electricity requirements.

The possibility of transferring to the grid (utility network) excess electricity available, i.e. that not directly utilisable, does immediately resolve the problem of the simultaneous "in phase" usage requirement. In other words in such a case the CHP plant is always completely and continuously employed at full load. Obviously the excess electricity transferred to the grid must be adequately compensated for in monetary terms ("buy back" on feed in).

It is highly probable that the cogenerator uses all the electricity produced at his own facility, particularly if industrial applications are involved. Many applications of this type have an immediate use for the recovered waste heat in an associated process.

The use of the recovered heat in commercial buildings (office complexes, shopping centres, hotels, etc.) must on the other hand be considered differently. Whilst the winter months probably provide good opportunities for the utilisation of the heat for comfort space heating and for sanitary hot water, the same cannot be said of the summer months. The great majority of the recovered heat available would otherwise be totally wasted if it were not possible to profitably utilise it in an absorption chiller. In this case one should speak not of cogeneration but rather trigeneration or CHCP = combined heat, cool & power.

The absorption chiller units currently available on the market use as the working fluid an aqueous solution of lithium bromide. They have a characteristic COP (coefficient of performance)of 0.7, operating as a single effect absorption cycle, producing chilled water at 7°C, when fed with hot water from the CHP plant at about 90°C.

Allowing for such a COP value and assuming for example that all the heat recovered from the engine is used in the absorption chiller, figure 4 shows the overall system efficiency, which will be in the order of 70%.

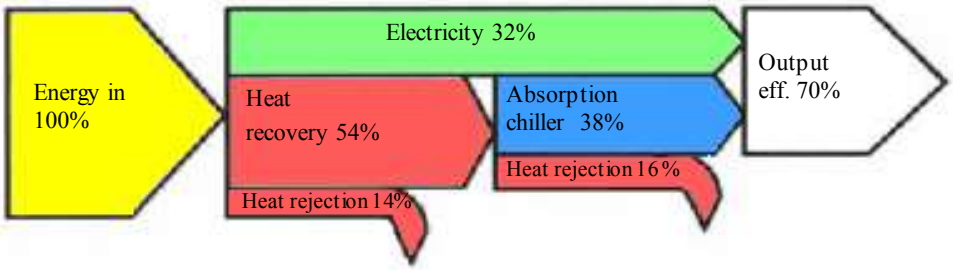



*Figure 4 - Trigeneration / CHCP system with the heat recovered used for the production of chilled water*

The use of the heat recovered for cooling along with space heating and hot water production does substantially improve the economics of the system.

Figure 5 illustrates how the system efficiency increases if a fraction of the heat unused for heating and hot water purposes is employed to provide cooling.

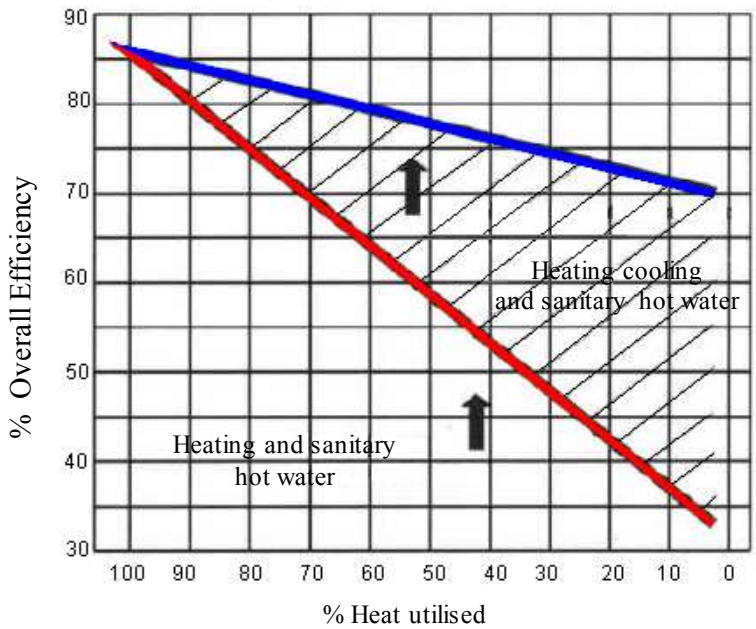



*Figure 5 – The graphs refer to a CHCP system in which all the heat available, unused for heating or hot water production, is employed for provide cooling.*

In practice it is however most unlikely that all of the heat recovered from the cogeneration engine can be completely utilised. Since the heat available can be defined by:

Hr = (Pe – Ep) er

where:

Hr = Usable heat recovered
Pe = Primary energy used
Ep = Electrical power generated
er = Efficiency of the heat recovery system adopted

It therefore follows that the overall system efficiency can be expressed by the following equation:

$$Te = \frac{Ep + (Hr \cdot qh) + (Hr \cdot qc) \cdot Ae}{Pe}$$

where:

Te = Total efficiency
qh = Percentage of heat utilised for space heating
qc = Percentage of heat utilised for cooling
Ae = COP of the absorption process

The formula indicated above should be used to calculate, albeit approximately, the overall efficiency and the economic convenience of the installation, taking account of the various energy forms involved.

## Absorption Chillers

The absorption chillers considered in this report are those of lower capacity, using a water and lithium bromide solution as their working fluid and hot water as the driving force.
The said units have been designed to use low-temperature heat and find application typically in industrial processes and in small-capacity CHP systems.
The absorption cycle requires hot water in the 70°C to 95°C temperature range. The chilled water produced leaves the evaporator at 7°C, in other words, the effective temperature for technological cooling and air-conditioning processes. The heat is rejected from the circuit as the water circulates in the absorber and condenser heat exchangers. The standard series of absorption chillers comprises five units with rated cooling capacities of 17.6 kW, 35 kW, 70 kW, 105 kW and 176 kW respectively (corresponding to 5, 10, 20, 30 & 50 RT where 1 Refrigerant Ton or "RT" equates to 3,5 kW of refrigeration). Since they are modular, mutually compatible appliances, they can be installed in combinations of two or more units to provide capacities other than those listed above. Table 1 lists the technical engineering specifications of the absorptions chillers examined here. The utilisation of the heat recovered in a CHP plant proves itself to be a natural application of this equipment.
The performance characteristics of each single unit are illustrated by the related characteristic curves.
Figure 6 below carries those of the unit with a nominal cooling capacity of 105 kW, which will serve us in illustrating the example shown below.
The curves contemplate cooling water inlet temperatures of 27 °C, 29.5 °C, 31 °C and 32 °C respectively.
The following considerations are valid for these, and indeed for the whole product range of these units.
At a given, fixed temperature of chilled water production, taken as 7°C, the chilling capacity generated is greatly affected by the temperatures of the cooling (i.e. heat rejection) water and of the hot feed water. Indeed, higher capacities will be achieved by lowering the temperature of the cooling water or raising that of the heat medium.
The temperature of the cooling water, which cannot in any case be lower than 24°C if working solution crystallization is to be avoided, is dictated, where evaporative cooling towers are used, by the ambient wet bulb temperature at the place of installation.
The heat medium temperature cannot exceed 95°C in the units under consideration for reasons of operating safety, the appliances' top design temperature being 100°C. It can, however, descend to very low levels, albeit suffering a steep downturn in capacity. Lastly, figure 6 again, shows the de-rating factor as a function of reduced hot water flow rate, all the other characteristic operating parameters remaining equal.

The values obtained from the graphs shown can be used for hypotheses of different operating conditions of the units, although the results thereby obtained must be considered to be only indicative.

| **Model** | | | | **WFC SC 5** | **WFC SC 10** | **WFC SC 20** | **WFC SC 30** | **WFC SC 50** |
|---|---|---|---|---|---|---|---|---|
| Cooling capacity | | | kW | 17.6 | 35.2 | 70.3 | 105.6 | 175.8 |
| Chilled water | Temperature | Inlet | °C | 12.5 | | | | |
| | | Outlet | °C | 7 | | | | |
| | Evaporator pressure drop | | kPa | 52.6 | 56.1 | 65.8 | 70.1 | 40.2 |
| | Maximum operating pressure | | kPa | 588 | | | | |
| | Flow rate | | l/s | 0.77 | 1.53 | 3.06 | 4.58 | 7.64 |
| | Water retention volume | | l | 8 | 17 | 47 | 73 | 120 |
| Cooling water | Heat rejection | | kW | 42.7 | 85.4 | 170.8 | 256.2 | 427 |
| | Temperature | Inlet | °C | 31 | | | | |
| | | Outlet | °C | 35 | | | | |
| | Absorber/condenser pressure drop | | kPa | 38.3 | 85.3 | 45.3 | 46.4 | 41.2 |
| | Coil fouling factor | | $m^2hK/kW$ | 0.086 | | | | |
| | Maximum operating pressure | | kPa | 588 | | | | |
| | Flow rate | | l/s | 2.55 | 5.1 | 10.2 | 15.3 | 25.5 |
| | Water retention volume | | l | 37 | 66 | 125 | 194 | 335 |
| Heat medium | Heat input | | kW | 25.1 | 50.2 | 100.4 | 150.6 | 251 |
| | Temperature | Inlet | °C | 88 | | | | |
| | | Outlet | °C | 83 | | | | |
| | | Range limits | °C | Minimum 70 – Maximum 95 | | | | |
| | Generator pressure drop | | kPa | 95.8 | 90.4 | 46.4 | 60.4 | 85.2 |
| | Maximum operating pressure | | kPa | 588 | | | | |
| | Flow rate | | l/s | 1.2 | 2.4 | 4.8 | 7.2 | 12 |
| | Water retention volume | | l | 10 | 21 | 54 | 84 | 170 |
| Electrical supply | Power supply | | | 220V 1-phase 50Hz | 400V 3-phase 50Hz | | | |
| | Power | | W | 48 | 210 | 260 | 310 | 590 |
| | Absorbed current | | A | 0.22 | 0.43 | 0.92 | 1.25 | 2.6 |
| Control | Chilling | | | ON - OFF | | | | |
| Dimensions | Width | | mm | 594 | 760 | 1,060 | 1,380 | 1,785 |
| | Depth | | mm | 744 | 970 | 1,300 | 1,545 | 2,060 |
| | Height (including mounting plate / levelling bolt) | | mm | 1,786 | 1,983 | 2,116 | 2,130 | 2,223 |
| Weight | Dry | | kg | 365 | 500 | 930 | 1,450 | 2,100 |
| | Operating | | kg | 420 | 604 | 1,156 | 1,801 | 2,725 |
| Noise | Sound pressure level dB(A) at 1 m | | | 46 | 46 | 49 | 46 | 57 |
| Piping diameter (A) | Chilled water | | mm | 32 | 40 | 50 | 50 | 80 |
| | Cooling water | | mm | 40 | 50 | 50 | 65 | 80 |
| | Heat medium | | mm | 40 | 40 | 50 | 65 | 80 |
| Enclosure | | | | Waterproof, suitable for outdoor installation, fitted with galvanised steel panels painted in an aluminium finish. | | | | |

*Table 1 –Technical specifications of the chillers under consideration*

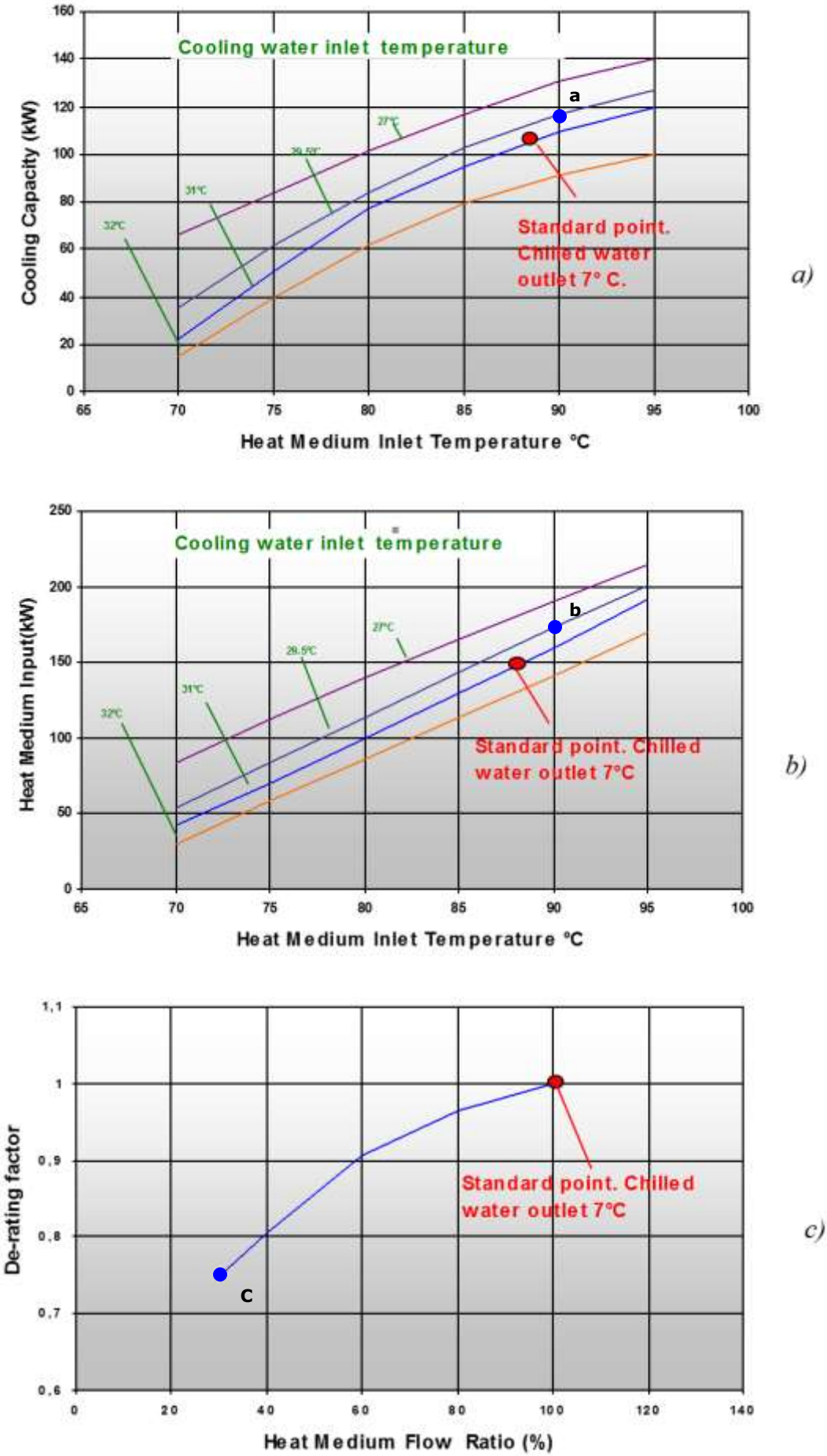
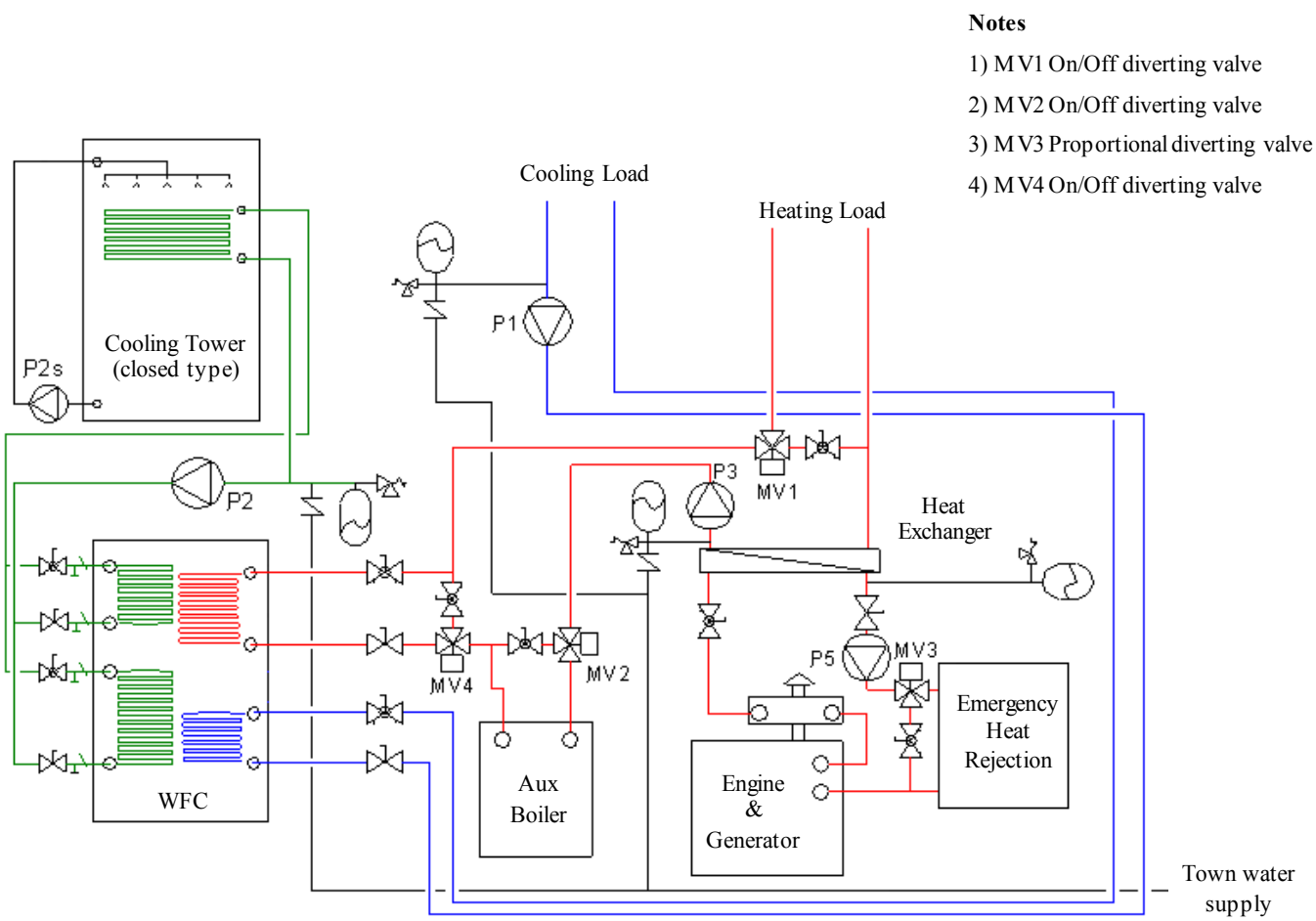



*Figure 6 - Characteristic performance curves for absorption chiller of 105 kW nominal cooling capacity*

## CHCP System Layout Schematic

Figure 7 shows the layout scheme of a CHCP installation which uses an internal combustion engine and an absorption chiller with a 4-pipe distribution system.
A characteristic feature of the plant are the two circuits, of which the primary serves the engine / alternator, fitted with an air-blast cooler for emergency heat dissipation, whereas the secondary circuit manages the absorption chiller and related cooling tower. The two circuits are interconnected by a heat exchanger.
When the engine starts-up, the pump P5 is also activated. The engine cooling water flows from the heat exchanger, through the mixing valve MV3 to the engine and then returns to the heat exchanger.
Whenever the system does not require heat and the coolant exceeds the pre-established temperature limit (approximately 90°C) the valve MV3 will divert a part of it to the emergency heat dissipation unit. This ensures that the maximum return temperature dictated by the engine manufacturer is in fact respected.

*Figure 7 - Schematic layout of a CHCP installation with the heat rejection effected by a cooling tower*




When the available hot water temperature in the heat exchanger is at least 70 °C, and chilled water is required, pumps P3, P2 and P1 serving the absorption chiller's hydraulic circuits are activated.
The hot feed water flows out of the heat exchanger through the diverting valves MV2 and MV4 to the absorption chiller and then returns from there to the heat exchanger. The chilled water driven by pump P1 flows to the various user equipment items such as the air handling units, fan-coils or other units.

If there is a simultaneous requirement for heat then the diverting valve MV1 is activated with the aim of directing hot water, coming from the absorption chiller, to the user point requiring heat.

In those cases in which there is only a heat requirement for heating purposes, valve MV4 will shut-off the absorption chiller from the rest of the circuit.

Whenever the engine is put out of service for maintenance or other reasons, valve MV2 will assure the diversion of all the heat transfer fluid towards the auxiliary boiler, in order to satisfy in any case the (heat) needs of the user equipment. The former must, obviously, be capable of supplying the hot water at the design temperatures and in the right quantities.

In the layout schematic a closed-circuit cooling tower is fore**s**een. Since the water quality and treatment do always require particular attention, closed-circuit cooling towers offer a suitable solution for safeguarding the absorption chiller.

## The Control System

The equipment operation control system is set out below, with reference to the layout shown in Figure 7. The technical specifications table lists the ON-OFF control functions. In ON mode, a thermostat (WT1) positioned on the chilled water delivery pipe controls the hot feed water pump as a function of the temperature of the chilled water.
The standard control temperatures are as follows:

- hot water pump activated when chilled water outlet (supply) temperature rises to 10.5°C;
- pump switched off when chilled water outlet (supply) temperature falls to 6.5°C.

*Figure 8. - Trigeneration plant with absorption chiller used for process cooling at A. Gandola & C - Ponte San Marco, Brescia, a chocolate producer. 200 kW electrical power generation, 280 kW useful heat production, 105 kW cooling capacity.*

The aforementioned temperatures (set-points) can be altered, although the temperature differential between them cannot be changed.
This feature means that the units can be step controlled by setting the individual chillers' set-points differently, in cascade, in those systems comprising a combination of several units: the systems' use can thus be regulated according to the load variations.

For chilling, the unit uses a chilled water pump P1, cooling water pump P2 and hot feed water pump P3. In normal operation, hot water pump P3 and cooling water pump P2 are controlled by thermostat (WT1), which is positioned on the chilled water outlet. As a result, if the temperature of the chilled water drops

to 6.5°C or less, pumps P2 and P3 are switched off, whereas the chilled water pump P1 is kept running, thus handling the variable demand made on the air-conditioning system.

If the temperature of the chilled water conveyed by pump P1 rises to 10.5°C, pumps P2 and P3 turn themselves automatically back on, thus restoring the chiller to full running mode. It may thus be said that the unit is actually never turned off by the thermostat (WT1), but is merely cut off from the hot water supply, adjusting to the cooling load required by the system.
When the absorption chiller is turned OFF, showing STOP on the CPU control panel, pump P1 also stops running. This state obviously consists in the unit being completely out of action.

| **Model** | | | | **PMS 6/65** | **ICT 3-63** | **ICT 4-54** | **ICT 4-66** | **ICT 4-59** |
|---|---|---|---|---|---|---|---|---|
| Capacity | | | kW | 42.7 | 85.4 (51,2) | 171 | 256 | 428 |
| Water | Temperature | Inlet | °C | 35 | | | | |
| | | Outlet | °C | 31 | | | | |
| | Flow rate | | l/s | 2.55 | 5.12 | 10.2 | 15.3 | 25.5 |
| | Pressure drop | | kPa | 4 | 2. | 17.9 | 17.9 | 22.1 |
| | Wet bulb | | °C | 26 | 26 | 26 | 26 | 26 |
| | Water evaporated max | | l/m | 1.11 | 2.12 | 4.5 | 6.7 | 11.2 |
| Air | Fans | | n° | 1 | 1 | 1 | 1 | 2 |
| | Consumption | | kW | 0.55 | 0.7 (0.15) | 1.5 | 2. | 2 x 1.5 |
| | Air flow (max) | | m3/s | 1.19 | 2.32 | 4.3 | 6.3 | 9.3 |
| Power supply | Electricity | | | 400V 3-phase 50Hz | | | | |
| Motor | | | | 4-pole | 4/8-pole | 4-pole | 4-pole | 4-pole |
| Dimensions | Width | | mm | 800 | 914 | 1216 | 1826 | 2731 |
| | Depth | | mm | 800 | 921 | 1226 | 1226 | 1226 |
| | Height | | mm | 2110 | 1880 | 2312 (2414) | 2617 (2719) | 2616 |
| Weight | Dry | | kg | 75 | 235 | 320 (365) | 575 (620) | 853 |
| | Operating | | kg | 180 | 400 | 685 (730) | 1085 (1130) | 1592 |
| Noise | Sound pressure level dB(A) at 3 meters in free field, measured at top | | | 55.3 at 10 m | 69 (62)* | 76 (60)** | 79 (68)** | 84 (72)** at 1.5 m |
| Piping diameter | Inlet | | mm | 50 | 80 | 100 | 100 | 100 |
| | Outlet | | mm | 50 | 80 | 100 | 100 | 100 |
| | Make-up | | mm | 20 | 25 | 25 | 25 | 25 |
| | Overflow | | mm | 15 | 50 | 50 | 50 | 50 |
| | Drain | | mm | 20 | 50 | 50 | 50 | 50 |
| Finishes | Heat exchange surface / fill pack | | | PVC | | | | |
| | Water distribution ramp | | | PVC | | | | |
| | Water spraying nozzles | | | PP | ABS | ABS | ABS | ABS |
| | Droplet separators / drift eliminators | | | PVC | | | | |
| | Basin | | | Fibreglass-reinforced plastic | Galvanised steel Z725 | Galvanised steel Z725 | Galvanised steel Z725 | Galvanised steel Z725 |
| | Facing panels | | | Fibreglass reinforced plastic | Galvanised steel Z725 | Galvanised steel Z725 | Galvanised steel Z725 | Galvanised steel Z725 |
| | Axial fan | | | Galvanised steel | Aluminium | | | |
| NB. | *Values at low fan speed for unit ICT 3-63 stated in brackets<br>**Values for adoption of low-noise fan for units ICT 4-54, ICT 4-66 and ICT 4-59 stated in brackets | | | | | | | |

*Table 2. Technical/engineering specifications of the evaporative cooling towers taken into consideration*

## Heat Rejection

As mentioned previously, in order for the absorption chiller to run properly, a quantity of heat – the sum of the heat used to energise the unit itself and the heat absorbed during chilling – has to be rejected to the surroundings.
In the case in point, the WFC-SC 30 unit, the heat to be rejected will be 256.6 kW, being the sum of the heat medium's 150.6 kW and the 105 kW absorbed during chilling.
Evaporative cooling towers are usually used to reject the heat generated in absorption chillers, which use a mixture of water and lithium bromide as their working fluid. They are adopted chiefly on account of the low heat rejection temperatures required by the chiller (31-35°C).
The operating principle of an evaporative cooling tower is based on the use of the latent heat of evaporation of water. A finely divided stream of water is brought into contact with a current of air in the tower, and a small quantity of water is absorbed into the air current by evaporation, drawing its latent heat of evaporation off the remaining water.

*Figure 9. Società Metropolitana Acque, Torino (Turin Metropolitan Water Company) - Absorption chillers operating on hot water, derived from a CHP plant, of 140 kW overall cooling capacity: they are employed to cool the biogas produced in the digesters.*

The water leaving the cooling tower will be slightly less in quantity, but considerably colder than the initial entering water. The heat removed, as latent heat, will be discharged into the environment in the form of water vapour carried by the outgoing current of air, whose humidity will thus be higher than that of the intake air, normally to the point of saturation.

The heat removed from the water depends not on the dry-bulb temperature of the incoming air, but only on its wet-bulb temperature. This is important because, for a relative humidity lower than 100%, the wet-bulb temperature is lower than the dry-bulb temperature (for example, at 32°C dry bulb and 52% relative humidity, the wet-bulb temperature is only 24°C), and considerably lower temperatures than those obtainable using a dry air-blast system can be reached in the cooling processes served.

The feature specific to evaporative cooling towers is that cooling is achieved at the expense of a modest water consumption (a few percent of the circulating flow), but with lower energy consumption than is the case with equivalent air cooling.

Water consumption in an evaporative cooling tower is the sum of the water evaporated, the water entrained as drops of water in the air leaving the tower (usually in the region of 0.002% of the circulating water) and the bleed-off water, which varies according to the quality of the water used and ranges in quantity from 30 to 100% of that amount of water consumed in evaporation.
As far as bleed-off (deconcentration) is concerned, it must be borne in mind that the continual evaporation of the water gradually raises the concentration of salts and other impurities, including those absorbed from the surrounding air, in the remaining water. If it is not kept in check, the concentration of salts and other impurities dissolved in the re-circulated water rises very rapidly, causing scaling, deposits and corrosion, all prejudicial to the units' proper operation and service life. Purging / bleed-off is required to limit those concentrations, meaning constant drawing-off performed via a small valve, preferably positioned on the pipe conveying the water into the tower.

Table 2 carries the technical/engineering specifications of the evaporative cooling towers to be used in conjunction with the absorption chillers taken into consideration in this paper. By way of an idea of the figures concerned, for the WFC-SC 30 unit cited as our example, the maximum water consumption figures are as follows: evaporation: 402 l/h; entrainment: 1.1 l/h; bleed: 402 l/h (maximum figure), thus making an overall maximum of 805 l/h.

## Compatibility of the units

The various considerations set out hitherto assume that the trigeneration plant has been well designed for its specific application and that the various equipment items adopted are all perfectly compatible; in particular, that the suitability of the temperature intervals and of the flow rates of the hot water used have been checked for the absorption chillers.
The installation must be meticulously designed to ensure substantial recovery of the available heat; this is particularly necessary considering that most of it is used in the absorption chiller, whose heat supply and performance are closely bound up with it. Miscalculating the flow rates and temperature compatibilities can make the system partially inefficient.

Frequently, heat recovery from an engine is effected by means of circulating water with flowrates such that a temperature range of 20°C is obtained, with a supply value of 90 °C and the return at 70 °C.

If one assumes, for instance that water in the primary circuit, which is that of the engine, is available at a maximum temperature of 90°C, then the same temperature is, in theory, to be found as well in the secondary circuit which feeds the absorption chiller. If the chiller were in a position to exploit the whole temperature range envisaged for the primary circuit, the heat capacity recovered from the engine would be fully exploited.
It must, however, be borne in mind that absorption chillers usually cannot operate with such a wide temperature differential (range) and are substantially penalised when run at input temperatures lower than their rated temperatures (see figure 6). In practice, this translates into partial use of the thermal capacity available on the primary circuit.
The unexploited heat can, of course, be used elsewhere or rejected via the emergency air-blast cooler. Nevertheless, the heat available might have been deemed fully convertible in the design of the absorption chiller and, as mentioned above, if the heat recovered from the engine is not largely in excess of that required by the absorption chiller, a considerable reduction in cooling capacity will result.

The maximum capacity obtainable is in any case achieved when the flow rate in the secondary circuit is the same as that in the primary circuit.
The example shown below will better clarify what has been asserted above.

## Performance Evaluation Example

The values considered in the example can be identified on the characteristic performance curves shown in figure 6 from the following symbols:

- ● nominal or rated operating values.
- ■ values which are different from those of rating, contemplated in the example.

The following formulae need to be recalled.

Heat balance of the unit

$Pd = Pa + Pf$ (1) (kW)

where:
Pd = heat rejected/dissipated into the surrounding air (kW)
Pa = heat supplied (kW)
Pf = cooling produced, equivalent to the heat taken-up from the rooms and spaces to be air-conditioned (kW)

Heat exchange in a hydraulic circuit.

$P = C\ \Delta T\ Q$ (2)

where:
P = heat exchange capacity (kW)
C = specific heat of water, which is 4.187 (kWs/°C l)
ΔT = water temperature differential (range) (°C)
Q = volumetric water flow rate (l/s)

One considers an engine which, operating in a CHP arrangement, produces at full load 180 kW of heat, with a supply water temperature of 90°C and 70°C return temperature.
From equation (2) the water flowrate of the related circuit will thus be:

$$Q = \frac{P}{\Delta T \times C} = \frac{180\ kW}{20°C \times 4{,}187\ kW\ s\ /\ °C\ l} = 2{,}1\ l/s$$

For the production of chilled water it is envisaged to use absorption chillers of the series being considered.

For this purpose the adoption of an absorption chiller of 105 kW cooling capacity (WFC-SC 30) is evaluated: its performance characteristics are those illustrated by the curves reproduced in figure 6.

The supposition is made that the absorption chiller will be fed with hot water obtained from the engine, adopting a flowrate equal to that of the engine. This being 2,1 l/s turns out to be 29% of the nominal, rated value for the absorption chiller considered, fixed at 7,2 l/s, thereby corresponding to a penalisation (derate) factor on the output capacity of 0,73 (value identifiable as the ● point **c** in the graph of figure 6).

Having fixed at 90°C the supply hot water temperature to the absorption chiller, hypothesizing an inlet temperature of 29,5°C to its condensing circuit, one will obtain from the characteristic performance curves respectively:

| | | |
|---|---|---|
| Pf cooling output produced | = | 118,6 kW (point ● **a**) x 0,73 = 86,6 kW |
| Pa heat input supplied | = | 174,9 kW (point ● **b)** x 0,73 = 127,7 kW |

The temperature range of the water in the supply circuit is given by equation (2) to be:

$$\Delta T = \frac{P}{Q \times C} = \frac{127{,}7 \text{ kW}}{2{,}1 \text{ l/sec} \times 4{,}187 \text{ kW s / °C l}} = 14{,}5 \text{ °C}$$

Operating in this manner, only 127,7 kW of the 180 kW available, approx. 70% , will be gainfully utilised in the refrigeration process. Consequently the remaining 30% must be otherwise employed or simply rejected to the surrounding air.

If one were to wish to use this remaining heat to produce further cooling a second absorption chiller must be employed: naturally it will be of a lower capacity than the first unit whose operation was described above, for example of 70 kW nominal capacity (WFC-SC 20), to be installed in series with the other chiller.

So this second unit will be fed with hot water at 75,5 °C , being:

Ta = 90 °C - 14,5 °C = 75,5 °C

The flowrate of the hot water fed to the unit will obviously be the same, Q = 2,1 l/sec.

Proceeding as above, by means of the performance curves of the unit under consideration, or using a specifically-designed computer programme, one will obtain respectively (see performance data sheet shown below):

| | | |
|---|---|---|
| Pf cooling output produced | = | 44,3 kW x 0,83 = 36,8 kW |
| Pa heat input supplied | = | 57,1 kW x 0,83 = 47,4 kW |

The corresponding temperature range (difference) of the hot water supplied to the unit will be, from equation (2), equal to 5,5°C.
This range added to the previous 14,5 °C will give the total of 20 °C required.

The total heat capacity actually utilised will amount to 175,1 kW of the available 180kW and will provide 123,4 kW of cooling effect, with an overall COP equating to 0,7.

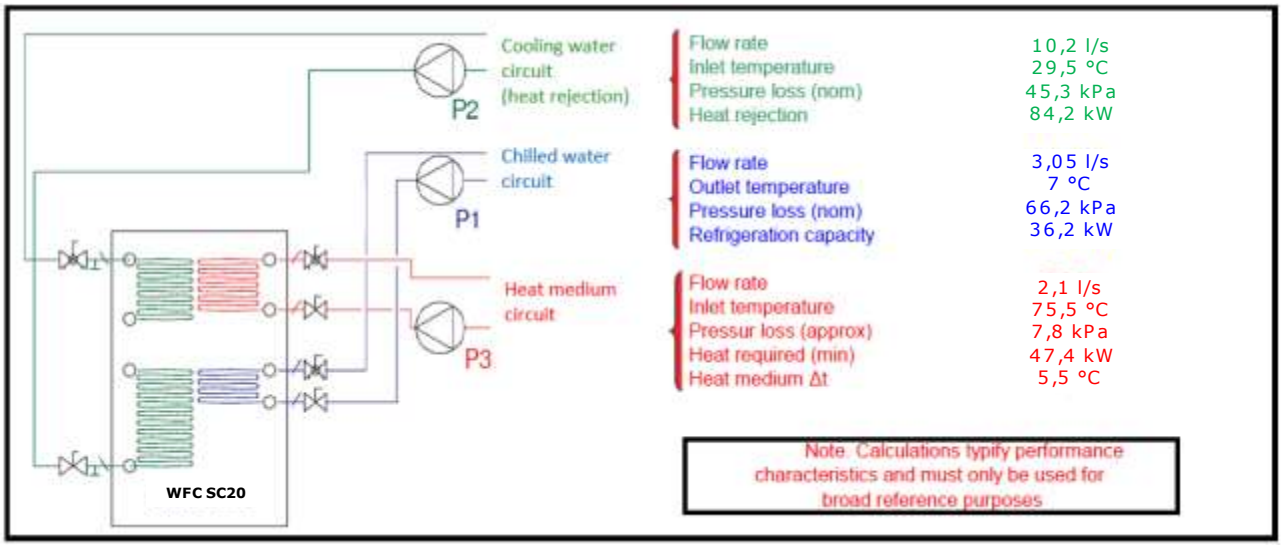
Obviously, alongside these purely technical evaluations, the economic convenience of the complete usage of the available heat energy must also be taken into account, given the low rates of return on the investments required for adding a second absorption chiller.

It should be once again be emphasized that the values for operation at conditions differing from the rated values must be derived from the characteristic performance curves or alternatively from the performance diagrams of the type shown below, that can be requested from the following e-mail address: maya@maya-airconditioning.com.
The values thus obtained are to be considered merely indicative and hence, used as such.

## WFC SC 20 Performance Diagram

for

| | |
|---|---|
| Hot water inlet temperature | 75,5°C |
| Hot water flow rate | 2,1 l/s |
| Cooling water inlet temperature | 29,5°C |



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