# SELF ADAPTIVE REFRIGERANT FLOW LOW TEMPERATURE DRIVEN DUAL LIFT ABSORPTION CYCLE

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## ABSTRACT

A dual lift water ammonia low temperature driven absorption cycle has been developed in which the refrigerant flow self adapts to the evaporator and absorber loads. A 3.4 kW air cooled chiller unit has been fully simulated and a prototype built and tested. The unit was run with hot water at inlet temperatures from 70°C to 95°C and with air temperature ranging from 25°C to 40°C . On the entire range of testing conditions the chiller delivers cooling capacity and runs smooth in transient and stable in steady state. In the current prototype setup ongoing test, at 35°C outdoor ambient temperature, 7°C outlet chilled water, 92°C driving hot water the chiller prototype has a cooling capacity of 3.4 kW with a COP of 0.29 and 360 W power consumption for the fan and the solution pump. At 32°C outdoor ambient temperature, 7°C outlet chilled water, 79°C driving hot water the prototype has a cooling capacity of 2.5 kW with a COP of 0.28 and 285 W power consumption for the fan and the solution pump.

## 1. INTRODUCTION

Most multistage absorption cycle (Ziegler, Kahn, Summerer, Alefeld, 1993), and in particular the vapor exchange cycle family (Richter, 1962), (Erickson and Tang, 1996) have either two vapor generators or a liquid refrigerant split dividing the refrigerant flow in two different streams. Of the two refrigerant streams coming from the two generators or the two branch of the split, one gives the cooling effect flowing through the evaporator, the other refrigerant stream is used to allow the cycle to operate at the given conditions, otherwise not possible. Both the two generators and the split cycles require precise controls of the two refrigerant loops to keep the system efficient and stable at a any given condition (Erickson and Tang, 1996), (Arivazhagan, Saravanan and Reganarayanan, 2005), (Kim and Infante Ferreira, 2005).

Specifically in the half effect double/single lift, the refrigerant split determines the amount of refrigerant for the evaporator and the amount of refrigerant to cool the low pressure absorber *RCA*, and if the split ratio is not exactly what required by the cycle at each given condition some efficiency will be lost. Also stability of the cycle could be an issue due to retro-action of the capacity of the evaporator/absorber causing oscillation of the pressures and therefore unstable flow in the entire cycle. In transient conditions the situation is even more critical as the optimum split ratio possibly varies continuously.

The typical purpose of the above cycles is their coupling with low temperature driving heat like for example engines heat rejection or solar cooling. The rejection heat from these applications has typically extremely variable behavior so the stability and load/input adaptability of a coupled cooling cycle is an essential step to their viability in real applications.

This problem has been addressed in this paper in which a novel two stage absorption cycle based on the dual lift half effect cycle is presented.

# 2. SELF ADAPTIVE REFRIGERANT FLOW LOW TEMPERATURE DRIVEN DUAL LIFT ABSORPTION CYCLE

## 2.1 Cycle Description

The self adapting refrigerant flow cycle can be derived for example from the so called single pump half effect absorption cycle (Kim and Infante Ferreira, 2005), or from the two pump series flow half effect cycle (Erickson and Tang, 1996), (Mostofizadeh, Butz, 1996), or any possible half effect cycle with a refrigerant flow split.

The following cycle (half effect dual lift series pumps) is taken as reference state of the art absorption cycle for the present development.

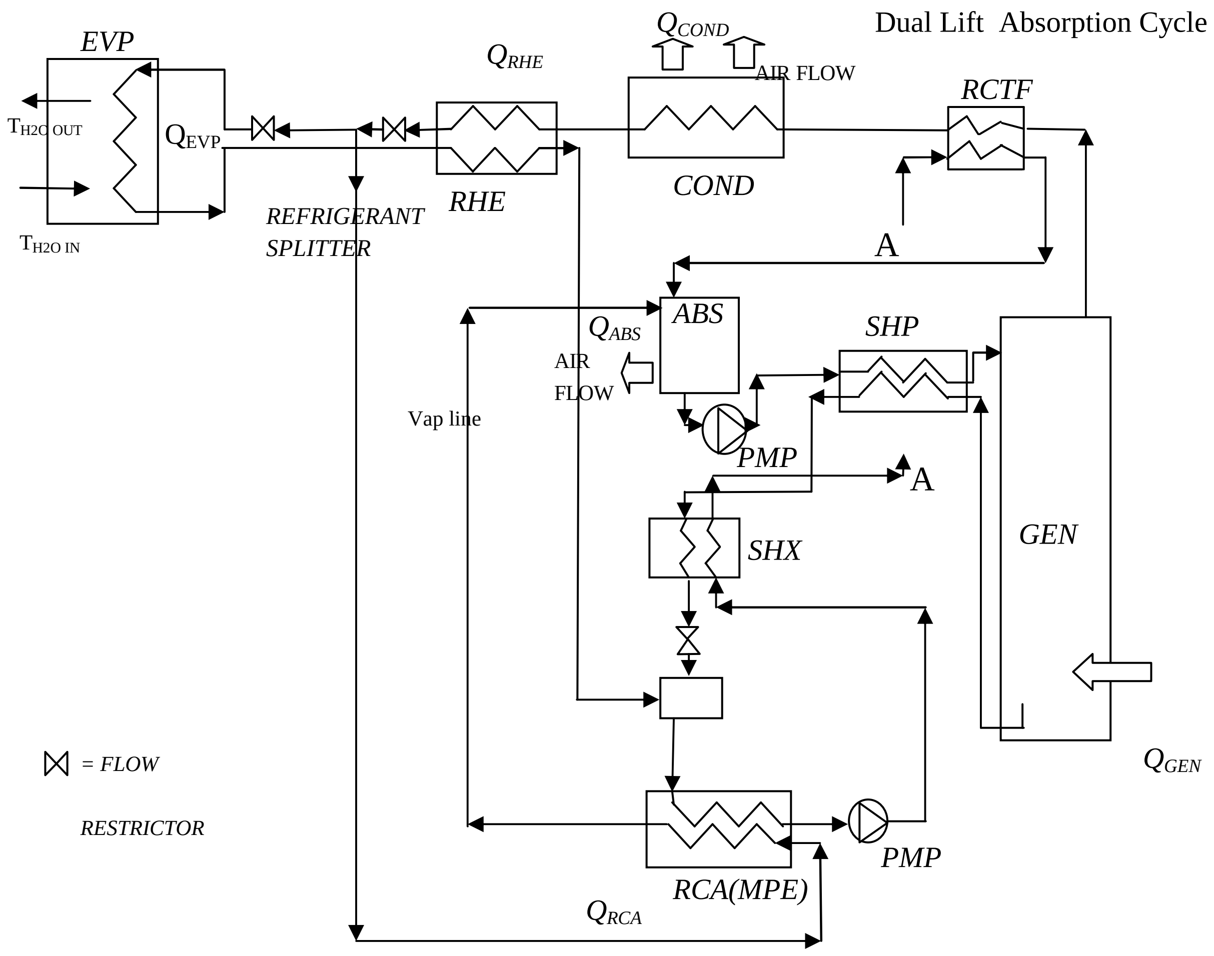


Figure 1 Dual Lift Half Effect Series Pumps

As anticipated in the introduction section the liquid refrigerant flow is divided according to the splitter geometry, to the refrigerant flow rate/pressure at the splitter inlet and at the two branch outlets and fluid viscosities at given pressure and temperature conditions. The splitter can be designed for an optimum ratio (which is approx. 42% refrigerant to the evaporator loop and 58% to the refrigerant cooled absorber, with typical variation within 20% in most working condition) at a given set of conditions like for example the generator input $Q_{GEN}$, the high, medium and low pressure, and evaporator $Q_{EVP}$. At conditions different than design the splitter will eventually divide the flow in a way so that the load of the evaporator and low pressure absorber will not be matched (even a constant split ratio could not be attained with different fluid dynamic conditions). Efficiency will be lost cause if there is extra refrigerant flow to the evaporator, the low pressure absorber, which is refrigerant cooled (from the other branch), will not be able to absorb the low pressure vapor and the low pressure will rise and the evaporator will flood reducing its capacity. If extra refrigerant will be supplied to the low pressure absorber, cooling effect will be consequently lost, and the solution out of the low pressure absorber will be needless subcooled. If we also consider a transient condition, for example determined by high pressure variation due to a generator input variation or ambient temperature variation, the optimum split ratio could possibly vary continuously, and even at start up fluctuating loads could give rise to unstable operation.

This situation could be fixed in a consistent way with the following cycle design (Guerra, 2011):

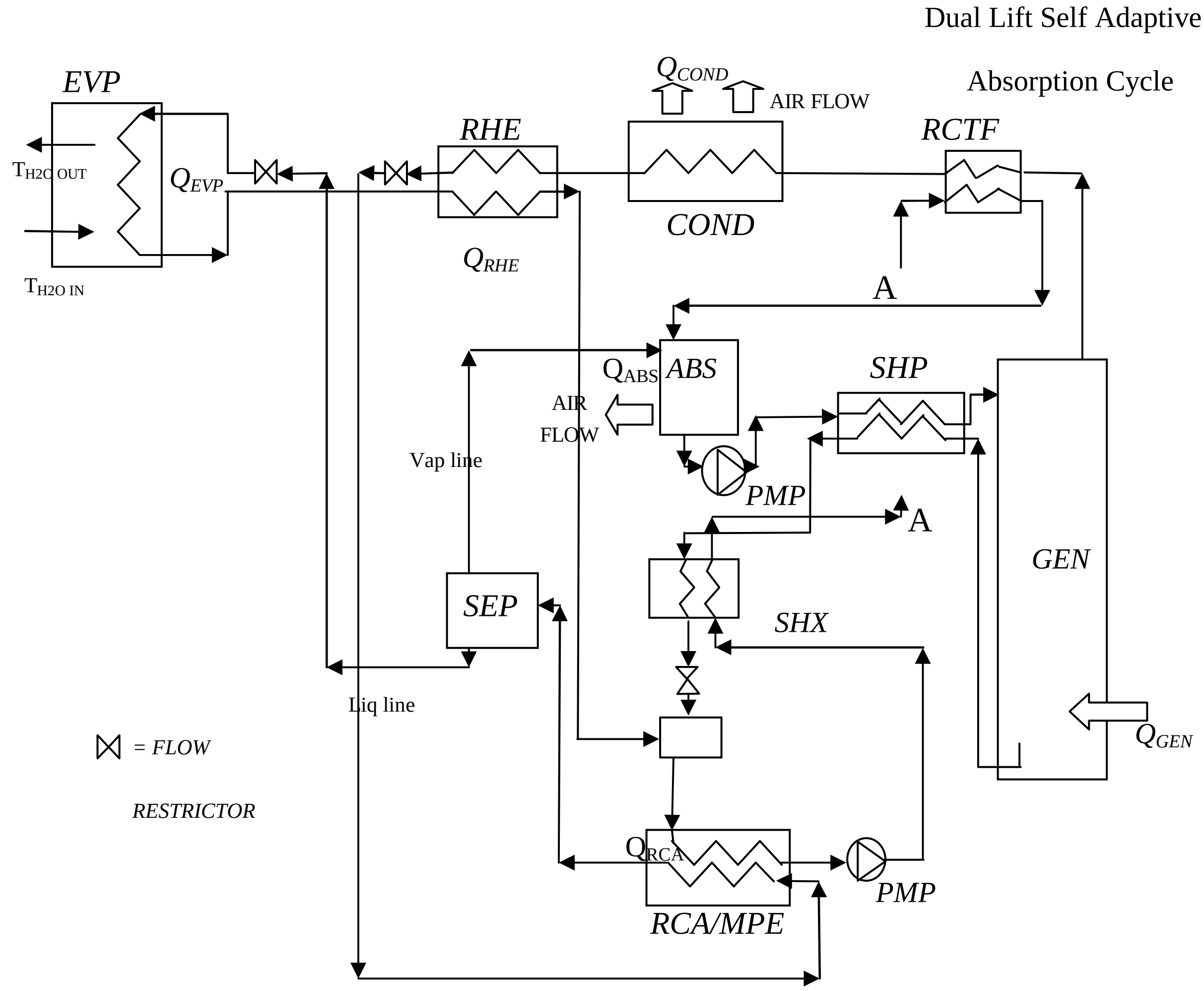


Figure 2 Self Adaptive Dual Lift Half Effect Series Pumps

The entire refrigerant flow coming from the condenser is brought to the intermediate pressure and passed through the low pressure refrigerant cooled absorber *RCA*. In the refrigerant cooled absorber, medium pressure evaporator side, the amount of refrigerant evaporated is basically only and exactly what needed to let the solution on the other exchanger side to absorb the vapor. The two phase liquid and vapor refrigerant flow leaving the refrigerant cooled absorber is routed to a liquid vapor separator *SEP* where the liquid stream is directed to the evaporator loop and the vapor is mixed with the solution in the medium pressure air cooled absorber.

The splitting of the two refrigerant streams is therefore obtained after the entire load of the low pressure absorber has been matched and it is easily done by separating a vapor phase from a liquid phase. At any given condition the match between the absorber load and the evaporator load is thus self adapted by this cycle design; if the refrigerant flow is not completely evaporated in the refrigerant cooled absorber there will be liquid refrigerant provided to the evaporator (minus flash losses), and a cooling effect assured by the fact that the low pressure absorber load is exactly satisfied; and the heavier the condition for the cycle (narrow thermal lift) the higher the fraction of refrigerant evaporated (consumed) for the absorber, the lower the cooling effect, the lower the COP and vice versa. So this cycle is expected to provide a stable cooling effect in a wide range of conditions.

**2.2 Cycle Simulation**

The simulation has been carried out for an air cooled water ammonia cycle as represented in figure 2. Given cooling capacity {$Q_{EVP}$}, cooling water temperatures and water flow rate {$T^c_{H2Oin}$,, $T^c_{H2Oout}$,, $M^c_{H2O}$ }, hot driving water temperatures and hot water flow rate { $T^h_{H2Oin}$,, $T^h_{H2Oout}$,, $M^h_{H2O}$ }, outdoor ambient temperature {*Tamb*}, typical heat exchanger temperature approach plus mass and energy conservation a steady state balance has been done determining pressure, flow rates and exchanger loads for each component of the cycle of figure 2. Water ammonia properties are those of (Ibrahim and Klein, 1993) and the software developed has been run on Wolfram's Mathematica. Here below several parameters of the cycle for different driving hot water temperature and ambient condition are shown.

*Sratio*= refrigerant to the evaporator/ refrigerant out of the condenser = equivalent split ratio

Table 1. Result of steady state simulations at *12→7°C* chilled water temp, *80→75°C* driving hot water temp

| Tamb [°C] | $P_{GEN}$ [$bar_{abs}$] | Pm [$bar_{abs}$] | $P_{EVP}$ [$bar_{abs}$] | $Q_{EVP}$ [kW] | $Q_{RCA}$ [kW] | $Q_{ABS}$ [kW] | $Q_{COND}$ [kW] | $Q_{GEN}$ [kW] | Mlret kg $h^{-1}$ | COP | Sratio |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 23 | 13.24 | 8.39 | 4.92 | 3.50 | 4.31 | 5.26 | 8.33 | 10.09 | 53.75 | .347 | 0.457 |
| 28 | 15.08 | 8.97 | 4.92 | 3.50 | 4.45 | 5.51 | 8.39 | 10.41 | 73.10 | .336 | 0.441 |
| 30 | 15.76 | 9.16 | 4.92 | 3.50 | 4.56 | 5.62 | 8.44 | 10.56 | 85.21 | .331 | 0.434 |
| 32 | 16.43 | 9.34 | 4.92 | 3.17 | 4.22 | 5.22 | 7.72 | 9.77 | 93.84 | .325 | 0.425 |

Table 2. Result of steady state simulations at *12→7°C* chilled water temp, *85→80°C* driving hot water temp

| Tamb [°C] | $P_{GEN}$ [$bar_{abs}$] | Pm [$bar_{abs}$] | $P_{EVP}$ [$bar_{abs}$] | $Q_{EVP}$ [kW] | $Q_{RCA}$ [kW] | $Q_{ABS}$ [kW] | $Q_{COND}$ [kW] | $Q_{GEN}$ [kW] | Mlret kg $h^{-1}$ | COP | Sratio |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 28 | 15.44 | 9.24 | 4.92 | 3.53 | 4.56 | 5.82 | 8.54 | 10.83 | 68.68 | .326 | 0.441 |
| 30 | 16.12 | 9.46 | 4.92 | 3.53 | 4.63 | 5.94 | 8.55 | 10.97 | 78.02 | .321 | 0.434 |
| 32 | 16.82 | 9.66 | 4.92 | 3.52 | 4.71 | 6.08 | 8.58 | 11.13 | 90.84 | .316 | 0.427 |
| 35 | 18.06 | 9.89 | 4.92 | 2.64 | 3.69 | 4.79 | 6.50 | 8.66 | 94.17 | .305 | 0.411 |
| 37 | 19.00 | 9.98 | 4.92 | 1.87 | 2.78 | 3.61 | 4.73 | 6.47 | 94.05 | .289 | .393 |

Table 3. Result of steady state simulations at *12→7°C* chilled water temp, *90→85°C* driving hot water temp

| Tamb [°C] | $P_{GEN}$ [$bar_{abs}$] | Pm [$bar_{abs}$] | $P_{EVP}$ [$bar_{abs}$] | $Q_{EVP}$ [kW] | $Q_{RCA}$ [kW] | $Q_{ABS}$ [kW] | $Q_{COND}$ [kW] | $Q_{GEN}$ [kW] | Mlret kg $h^{-1}$ | COP | Sratio |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 28 | 15.79 | 9.34 | 4.92 | 3.49 | 4.60 | 5.95 | 8.60 | 11.060 | 62.80 | .316 | 0.441 |
| 30 | 16.48 | 9.60 | 4.92 | 3.49 | 4.65 | 6.07 | 8.60 | 11.17 | 69.87 | .313 | 0.436 |
| 32 | 17.20 | 9.82 | 4.92 | 3.49 | 4.72 | 6.20 | 8.60 | 11.31 | 78.87 | .309 | 0.429 |
| 35 | 18.46 | 10.10 | 4.92 | 3.29 | 4.56 | 6.08 | 8.13 | 10.92 | 94.49 | .301 | 0.417 |
| 37 | 19.43 | 10.24 | 4.92 | 2.62 | 3.74 | 5.04 | 6.53 | 8.95 | 94.41 | .292 | 0.407 |
| 40 | 21.00 | 10.34 | 4.92 | 1.51 | 2.39 | 3.27 | 3.97 | 5.73 | 94.17 | .264 | 0.377 |

The model assumes no constrain on heat exchanger ability to provide the entire amount of transfer required. The only limitation is on maximum flow rate at the solution pumps, which is set at *120 litres/hour* for each of the two pumps. In real test of course the components geometry plays a crucial role and determines the actual performance of the unit. Also in the above simulations the pressure at the evaporator is kept constant, but the refrigerant/solution charge and restrictor setup of an actual unit, once fixed, could result in a different evaporator pressure.

The simulations show the general behavior of the cycle and the amount of refrigerant for the evaporator and the amount for the *RCA*, and in particular when the thermal lift, i.e. the difference between condensing temperature and the evaporating temperature is larger, as expected a larger amount of refrigerant is used to cool the low pressure absorber *RCA*.

**2.3 Prototype design**

The flow rates and thermal loads of the above simulations have been taken as input condition for the design of each component of the 3.4 kW target cooling capacity prototype. Here below a list with a brief description of some relevant features for each component

- Generator: vertical tube in shell & liquid distributor, falling film mode
- *SHP, SHX, RCA, RHE*: tube in tube, horizontal flow
- Evaporator: mini tube in shell, vertical flow
- Air cooled Condenser/ Absorber: steel tubes and aluminum fins, horizontal flow
- Rectifier: pall ring packing and solution cooled coil
- Restrictor: series of drilled disks
- Pumps: two oil driven diaphragm with single electrical inverter controlled motor [100W max power consumption]
- Fan, axial 500 mm diameter blade, variable speed

Each single component has been built and assembled in a complete unit. The unit prototype is approx. 1000 mm width, 500 mm depth and 1100 mm high.

Figure 3 Complete prototype unit.

A hot water loop has been connected to the generator, to a boiler and to variable flow pump, water flow meter and temperature transducers. The evaporator has been connected to an electrical resistance set, to an air handler, to a variable flow pump, to a flow meter and temperature transducers.

The outdoor ambient temperature has been simulated increasing the testing room temperature with a space heater.

Four pressure transducers and 24 thermocouples have been located on the unit. A wattmeter has been connected to the power line to the unit (fan and pump). All the transducer data are collected and recorded on a PC via NI LabView hardware and software.

## 2.4 Test results to December 2011

The prototype unit has been charged with pure water and ammonia. After some adjustments made on the rectifier setup, the unit has been run in a number of test at different charge, restrictors setup, hot water temperatures and ambient conditions. Here below a few of the results at steady state test (following EN 12309, where applicable).

The result of the experimental tests shown below are obtained with the unit at a fixed setup and charge. The thermal dispersion at the envelope of the generator has been measured by running hot water at empty unit and it is typically *100 W to 200 W* at water temperatures ranging from *70 to 90°C,* thus meaning approx. 1 point in COP.

In the reported configuration the unit has a minimum start up temperature at *64°C*, at which cooling effect is starting being delivered, with ambient temperatures from *15 to 28°C.*

Table 3. Result of the prototype steady state test to December 2011

| Tamb [°C] | PGEN [$bar_{abs}$] | Pm [$bar_{abs}$] | PEVP [$bar_{abs}$] | QEVP [kW] | Tcool [°C] | Theat [°C] | QGEN [kW] | COP | Power [W] |
|---|---|---|---|---|---|---|---|---|---|
| 22.7 | 14.21 | 8.55 | 4.97 | 2.86 | 12.1→7.1 | 74.1→69.6 | 9.23 | .310 | 165 |
| 28.1 | 15.16 | 9.23 | 5.05 | 2.87 | 11.2→6.2 | 79.0→74.1 | 9.54 | .300 | 280 |
| 28.0 | 15.94 | 8.52 | 4.66 | 3.28 | 10.4.→4.7 | 83.5→78.0 | 11.32 | .289 | 300 |
| 28.2 | 16.51 | 8.59 | 4.41 | 3.42 | 10.2.→4.7 | 88.2→82.5 | 11.81 | .29 | 280 |
| 30.6 | 16.20 | 9.44 | 5.28 | 2.86 | 12.6→7.6 | 79.6→74.7 | 9.94 | .288 | 300 |
| 31.0 | 16.79 | 9.54 | 5.34 | 3.14 | 12.7→7.2 | 84.1→78.9 | 10.89 | .288 | 310 |
| 29.7 | 17.02 | 8.99 | 4.65 | 3.41 | 10.9→5.5 | 89.3→83.5 | 11.94 | .286 | 290 |
| 32.1 | 15.85 | 9.20 | 5.13 | 2.49 | 11.9→7.3 | 79.4→75.0 | 8.90 | .280 | 285 |
| 32.1 | 16.54 | 9.70 | 5.29 | 2.78 | 12.6→7.4 | 85.2→78.8 | 9.99 | .278 | 310 |
| 32.5 | 17.42 | 9.25 | 4.87 | 3.31 | 11.9→6.8 | 90.6→82.6 | 11.50 | .288 | 330 |
| 33.8 | 17.33 | 9.94 | 5.44 | 2.80 | 12.8→7.8 | 85.0→79.7 | 10.03 | .279 | 360 |
| 34.8 | 18.51 | 10.14 | 5.21 | 3.27 | 12.8→7.8 | 91.8→86.1 | 11.78 | .278 | 350 |
| 34.9 | 19.0 | 10.25 | 5.35 | 3.41 | 12.8→7.4 | 93.9→88.1 | 12.08 | .282 | 360 |
| 35.1 | 18.66 | 10.36 | 5.43 | 3.31 | 12.8→7.7 | 92.0→86.4 | 11.57 | .286 | 380 |
| 37.0 | 18.48 | 10.54 | 5.15 | 2.23 | 11.9→6.9 | 90.2→85.3 | 10.38 | .215 | 380 |
| 37.0 | 19.10 | 10.68 | 5.35 | 3.07 | 12.4→7.6 | 94.1→88.5 | 11.75 | .261 | 380 |
| 37.4 | 19.20 | 10.72 | 5.3 | 3.00 | 12.2→7.3 | 94.7→89.0 | 11.67 | .24 | 380 |
| 36.8 | 19.02 | 10.7 | 5.63 | 3.28 | 13.9→8.7 | 92.5→87.0 | 11.48 | .286 | 380 |

### 2.5 Discussion

The general way the unit operates, in startups and in transient conditions, in on and off cycles is not different from the stability of an ammonia water single stage absorption unit. It always finds a stable and repeatable equilibrium state from different starting conditions.

Hot water flow rates giving $\Delta T = 4°C$ to $\Delta T = 10°C$ have been tested. As expected the lower the $\Delta T$ the higher the input the generator can take. Evaporator capacities are lower than simulated with hot water temperatures below 80-82°C and ambient temperatures above 28°C. In these conditions with the given setup the generator cannot take the entire input required.

The COPs fall within approx. 10% difference versus the simulated data, which is a rather remarkable result for a first prototype setup of a double stage air cooled water ammonia absorption unit. By performing an energy balance on actual test, the Split ratio equivalent has been found to fall within approx. 5% of the simulated value. This validates the concept of avoiding liquid splits and replacing them by the new cycle design.

The air cooled unit still delivers cooling capacity at outdoor ambient temperatures at least up to 40°C. In this situation the effect of evaporating refrigerant in RCA gradually reduces the evaporator capacity but allows to extend the operations to wide ranges of temperatures.

The evaporator pressure is strongly depending on the generator temperature. This behavior was not shown in the simulations in that the evaporator pressure was kept constant. The evaporator geometry (and rectified vapor concentration ) allow full capacity at evaporating pressure 0.3 bar higher than simulated.

In the simulations if the given conditions do not allow to take the entire maximum generator input, the generator input and solution flow rates are scaled down to match the required heat loads. In the real unit this is not the case, as the flow rate is determined by the pressure difference in the stages and the amount of charge and restrictor setup. If extra input is given to the generator more refrigerant and solution will flow in the unit and the performance will fall very quickly. This can be seen when thermal lift and/or difference between generator and condenser temperature exceed a certain limit.

The actual solution pumps maximum rated capacity is *120 liters/hour*, and with increasing difference between generator and condenser temperature this limit happens to be reached.

The falling film generator is fast in responding to the thermal input; if hot water is provided at temperatures above 70°C then cooling effect is given in less than 3 minutes from pump and fan startup.

### 2.6 Future developments

A new set of tests is going to be performed at negative evaporator temperatures, after a new restrictor configuration and a new charge setup is made.

The unit is expected to be moved to a new lab site at Milan Politecnico University early spring 2012 for independent testing.

## 3. CONCLUSIONS

A new absorption double stage cycle based on the double stage half effect has been developed to allow operation with low generator temperature input. The main feature of this new design is to self adapt the amount of refrigerant to the evaporator and to the low pressure absorber to improve efficiency, stability and to extend the range of operations. An air cooled water ammonia unit prototype has been built according to the new double stage self adapting cycle. The unit has been tested and the test show the performance is within 10% the simulated data. Also the operation of the prototype has proven to be stable and repeatable.

## NOMENCLATURE

| | | | |
|---|---|---|---|
| *Mlret* | flow rate returning to *GEN* (kg $h^{-1}$) | *ABS* | air cooled absorber |
| $M^{c}_{H2O}$ | cooling water flow rate (kg $h^{-1}$) | *COND* | air cooled condenser |
| $M^{h}_{H2O}$ | hot driving water flow rate (kg $h^{-1}$) | *COP* | $Q_{EVP}/ Q_{GEN}$ |
| *p* | pressure (bar) | *EVP* | evaporator |
| *Q* | thermal power (kW) | *GEN* | generator |
| *T* | temperature (C) | *PMP* | solution pump |
| *Tamb* | outdoor ambient temperature (C) | *RCA/MPE* | refrigerant cooled absorber |
| $T^{c}_{H2Oin}$, | cooling water temp. (C) | *RCTF* | rectifier |
| $T^{h}_{H2Oin}$, | hot driving water temp. (C) | *RHE* | refrigerant heat exchanger |
| | | *SEP* | liquid-vapour separator |
| | | *SHP* | high pressure heat exchanger |
| | | *SHX* | medium pressure heat exchanger |
| | | *liq* | liquid |
| | | *m* | medium |
| | | *vap* | vapour |

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