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THE EFFECTS OF THE M-CYCLE ON THE PERFORMANCE OF A GAS TURBINE

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ABSTRACT

The M-Cycle represents one of the best heat recovery technologies known today. Technically, it can be incorporated into the Brayton cycle by replacing the heat recuperator with a recuperator-humidifier and directing high humidity compressed air into the combustion chamber. This provides a large increase in the cycle efficiency due to the reduction of the ambient temperature (from dry bulb to the dew point temperature), a reduction in compressor work, an increase in the volumetric flow, increase in combustion efficiency (increased fuel economy), and a large reduction in NO_x.These combined improvements provide an improved Brayton cycle efficiency.

By adding steam to the fuel gas stream in the combustion chamber of a Brayton cycle turbine the operating power will increase substantially. Experiments with innovative gas turbine cycles like Evaporative Gas Turbine (EvGT)[1], the Humid Air Turbines (HAT)[2], or the Cascade Humidified Advanced Turbine (CHAT) cycle[3], where instead of steam an equivalent amount of water vapor is created from waste heat from stack gases, and in some cases intercompressor coolers, have shown improvements in efficiency. Using reduced compressor power for the same mass flow rate causes the largest efficiency gain. However, efforts to commercialize these advanced turbine cycles have been stymied by the difficulty in maintaining the air to humidity ratio, and the added capital equipment cost such as the saturating tower, boilers and numerous heat exchangers. The development of the M-Cycle offers a cost-effective solution to these issues by presenting the opportunity to realize the thermodynamic advantages of these high performance cycles [4].

This paper analyzes the performance of the M-Cycle when operating with a Rolls-Royce 250 gas turbine and compares the output to the standard Brayton Cycle while varying the inlet humidity, temperature and air and fuel flow rates.

NOMENCLATURE

h	*[Btu/lb]*	Enthalpy
m	*[lb/sec]*	Mass Flow Rate
p	[psia]	Pressure
r_p		Pressure Ratio
t	[°F]	Dry-bulb Temperature
t_d	[°F]	Dew-point Temperature
v	*[ft^3/lbm]*	Specific Volume
C		Constant
P		Pressure
R	[ft*lb_f/lb_m*°R]	Specific Gas Constant

Subscripts

1,2		Subscripts indicating state
F		Fuel
W	*[psia]*	Water Vapor Partial Pressure
Ws	[psia]	Saturation Pressure

BACKGROUND

The M-Cycle [5] applied to gas turbines is a three step process. After the compressor, the hot, compressed gas enters the M-Cycle heat exchanger and is divided into many small channels in a plate and frame arrangement. The hot gas travels the length of the exchanger, losing heat through the plates to the wet side. The gas is cooled down to the dew-point of the gas at that pressure. The second step has the gas passing through the wet side channels. The heat from the incoming gas, as well as the heat from the exhaust gas heats the liquid water and produces steam. This steam adds mass flow and energy to the gas stream and it exits the exchanger for the combustion chamber. The third step comes after expansion, as the hot exhaust gas passes through the exchanger to boil water for the entering gas stream. The M-Cycle details are outlined in the section below.

PSYCHROMETRICS

Psychometrics in this report are based on the ASHRAE 1992 Fundamentals book. The equations are processed into a spreadsheet and analyzed based on each point in the Brayton cycle and are presented here as a reference for the later discussion of the M-Cycle.

Psychrometric Equations

Saturation Pressure given over liquid water:

$$\ln(p_{ws}) = \frac{C_8}{T} + C_9 + C_{10}T + C_{11}T^2 + C_{12}T^3 + C_{13}\ln T \quad (1)$$

Where: $C_8 =$ -1.0440397E+04

$C_9 =$ -1.1294650E+01

$C_{10} =$ -2.7022355E-02

$C_{11} =$ 1.2890.360E-05

$C_{12} =$ -2.4780681E-09

$C_{13} =$ 6.5459673E+00

Humidity Ratio is given by:

$$W = 0.62198\frac{p_w}{p-p_w} \quad (2)$$

Enthalpy of the mixture is given by:

$$h = 0.240t + W(1061 + 0.444t) \quad (3)$$

Where the first term represents the enthalpy of the dry air, the second represents the enthalpy of the saturated water vapor.

Dew Point is given by:

$$t_d = C_{14} + C_{15}\alpha + C_{16}\alpha^2 + C_{17}\alpha^3 + C_{18}(p_w)^{0.1984} \quad (4)$$

Where: $C_{14} = 100.45$

$C_{15} = 33.193$

$C_{16} = 2.319$

$C_{17} = 0.17074$

$C_{18} = 1.2063$

PSYCHROMETRIC CHARTS

The Psychrometric Charts provide the user with a quick way to determine property states of a given air-vapor mixture given any two conditions. An example chart is given in the following for condition of 1 atmosphere.

The graph layout is given with dry bulb temperature (°F) versus humidity ratio given in pounds of water vapor per pound of dry air. The curved lines represent lines of relative humidity, given in 10% increments. The diagonal lines are lines of constant enthalpy given in Btu per pound.

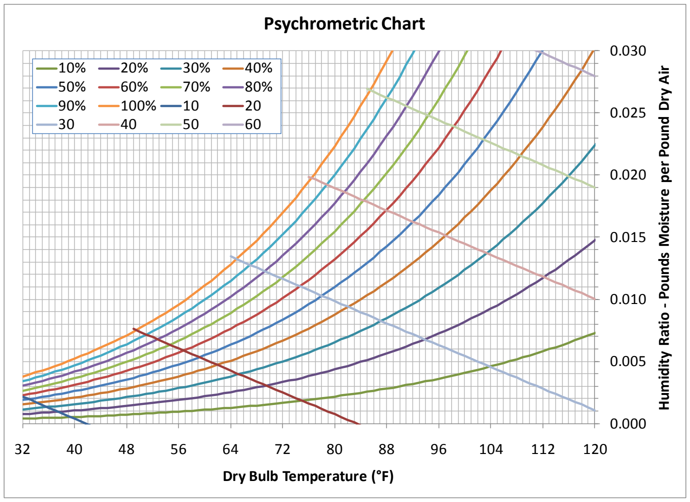


Figure 1 Psychrometric Chart (1 atm)

BRAYTON CYCLE

The Brayton Cycle is the fundamental thermodynamics cycle for modeling the performance of gas turbines. The combustion occurs at a constant pressure state, and the inlet and exhaust are at the same pressures. Again, as a reference for the understanding of the M-Cycle, the Brayton Cycle is described in four steps, as shown below:

Process 1-2: Isentropic Compression

The equations relating these two states can be seen below. This assumes a given pressure ratio, humidity ratio, standard atmospheric conditions at inlet, and air and water vapor as an ideal gas.

$$\dot{m}_1 = \dot{m}_2 \quad (5)$$

$$P_2 = P_1 * r_p \quad (6)$$

$$v_2 = v_1 / r_p^{\left(\frac{1}{\gamma-1}\right)} \quad (7)$$

$$T_2 = \frac{P_1 v_1}{R} \quad (8)$$

$$h_2 = 0.240T_2 + W(1061 + 0.444T_2) \quad (9)$$

Process 2-3: Isobaric Combustion

In the next step, the mass of the fuel adds to the total mixture, the pressure is stable, and the energy of the burned fuel is added to the enthalpy exiting the combustor.

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_f \quad (10)$$

$$P_3 = P_2 \quad (11)$$

$$v_3 = v_2 \quad (12)$$

$$T_3 = T_2 + \frac{h_3 - h_2}{c_p} \quad (13)$$

$$h_3 = h_2 + \frac{\dot{m}_f Q_{LHV}}{\dot{m}_3} \quad (14)$$

Process 3-4: Isentropic Expansion

The hot gas passes from the combustion chamber and expands through the power turbine.

$$\dot{m}_4 = \dot{m}_3 \quad (15)$$

$$P_4 = P_1 \quad (16)$$

$$v_4 = \frac{RT_4}{P_4} \quad (17)$$

$$T_4 = T_3/r_p^{(\frac{\gamma-1}{\gamma})} \quad (18)$$

$$h_4 = 0.240T_4 + W(1061 + 0.444T_4) \quad (19)$$

Process 4-1: Isobaric Heat Rejection

This is not an actual process, since the Brayton cycle is an open cycle. The exhaust gases are dumped to atmosphere while the fresh air begins the cycle again. Several processes, including the M-Cycle, rely on this “waste” heat for higher efficiencies in their modified cycles.

The figure below demonstrates the basic Brayton cycle on a P-v and T-s diagram.

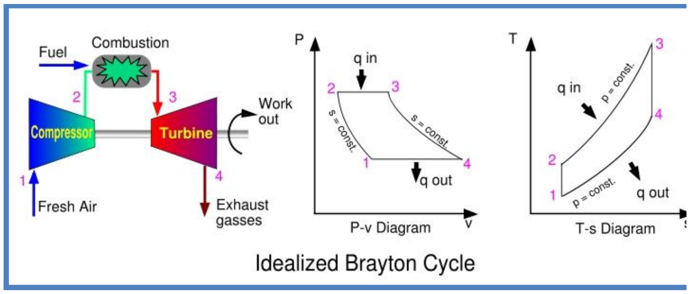


Figure 2 Basic Brayton Cycle

M- CYCLE

The M-Cycle is a modification of the state of the gas after compression to before combustion. The diagram below shows the flow path of the compressed gas as it flows from compressor to combustor. The gas passes through the dry side of the heat exchanger, then to the wet side, causing the flow to cool to saturation and then adding energy from the steam. The exhaust gas stream adds more energy to the incoming flow, as it passes heat through the liquid water in the exchanger.

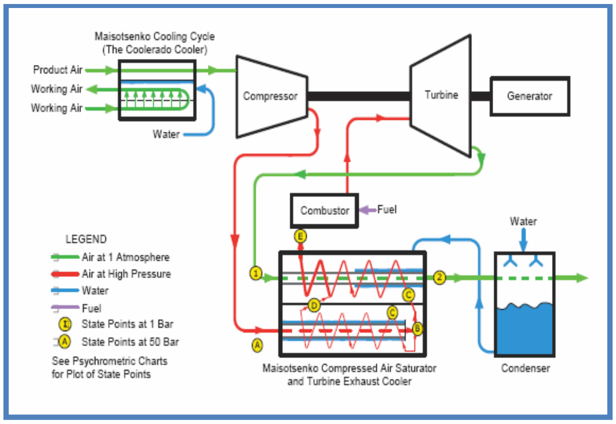


Figure 3 M-Cycle Diagram

M-Cycle Illustration on a Psychrometric Chart

The M-Cycle is described in three steps:

State A to B: Cooling of the flow down to the dew point

State C to D: Addition of saturated water vapor

State C to E: Heating of the combined Gas Stream

This can be shown on a Psychrometric chart as follows: (example at 7 atm compression)

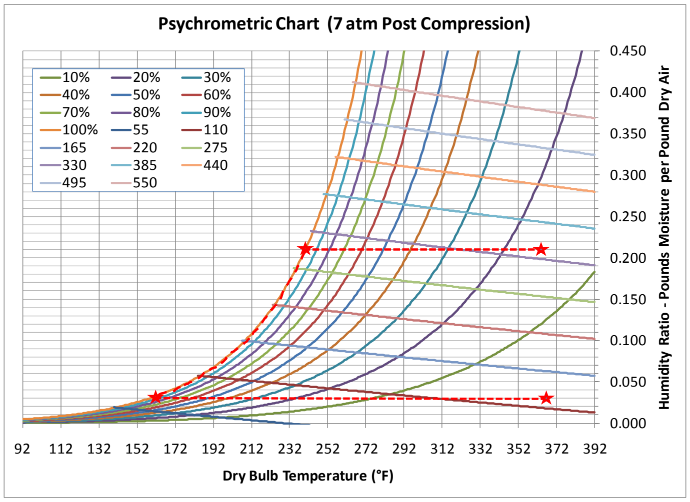


Figure 4 Compressor Psychrometric Chart

M-Cycle Equations

To solve for the M-Cycle states, we assume any extra energy is transferred at 100% efficiency and compare it to the unmodified cycle.

State A to B: Cooling of the compressed flow down to the dew point.

To calculate this temperature, we use the saturated vapor line at the specific humidity ratio coming into the cycle. The enthalpy difference is assumed to be transferred into the water on the wet side of the exchanger.

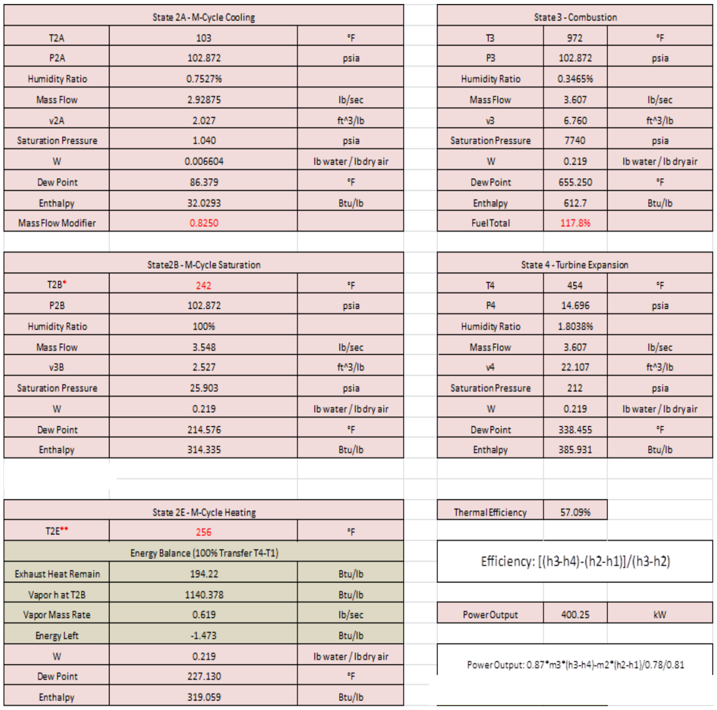
State C to D: Addition of Saturated Water Vapor

By using an iterative process, the heat from the inlet stream and exhaust stream is used to transfer liquid water into steam. This mass flow is added to the gas stream.

State C to E: Heating of Gas Stream

Using the last of the energy, the stream is heated back to near the original value of the inlet cycle. The iterative steps assume temperatures at D and E, then seek agreement at the end of the cycle by exhaust heat return.

EXAMPLE 1: M-CYCLE EXAMPLE (IDENTICAL FUEL FLOW RATE)



State 2A - M-Cycle Cooling		
T2A	103	°F
P2A	102.872	psia
Humidity Ratio	0.7527%	
Mass Flow	2.92875	lb/sec
v2A	2.027	ft^3/lb
Saturation Pressure	1.040	psia
W	0.006604	lb water / lb dry air
Dew Point	86.379	°F
Enthalpy	32.0293	Btu/lb
Mass Flow Modifier	0.8250	

State 3 - Combustion		
T3	972	°F
P3	102.872	psia
Humidity Ratio	0.3465%	
Mass Flow	3.607	lb/sec
v3	6.760	ft^3/lb
Saturation Pressure	7740	psia
W	0.219	lb water / lb dry air
Dew Point	655.250	°F
Enthalpy	612.7	Btu/lb
Fuel Total	117.8%	

State2B - M-Cycle Saturation		
T2B*	242	°F
P2B	102.872	psia
Humidity Ratio	100%	
Mass Flow	3.548	lb/sec
v3B	2.527	ft^3/lb
Saturation Pressure	25.903	psia
W	0.219	lb water / lb dry air
Dew Point	214.576	°F
Enthalpy	314.335	Btu/lb

State 4 - Turbine Expansion		
T4	454	°F
P4	14.696	psia
Humidity Ratio	1.8038%	
Mass Flow	3.607	lb/sec
v4	22.107	ft^3/lb
Saturation Pressure	212	psia
W	0.219	lb water / lb dry air
Dew Point	338.455	°F
Enthalpy	385.931	Btu/lb

State 2E - M-Cycle Heating		
T2E**	256	°F
Energy Balance (100% Transfer T4-T1)		
Exhaust Heat Remain	194.22	Btu/lb
Vapor h at T2B	1140.378	Btu/lb
Vapor Mass Rate	0.619	lb/sec
Energy Left	-1.473	Btu/lb
W	0.219	lb water / lb dry air
Dew Point	227.130	°F
Enthalpy	319.059	Btu/lb

Thermal Efficiency	57.09%

Efficiency: [(h3-h4)-(h2-h1)]/(h3-h2)

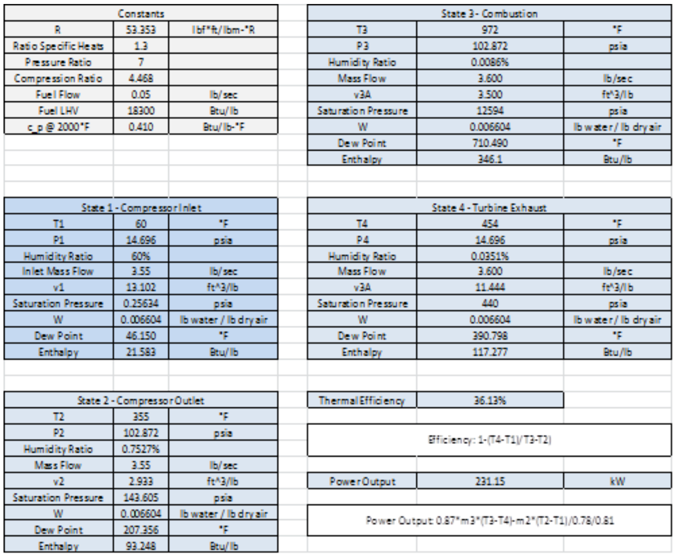
Power Output	400.25	kW

Power Output: 0.87*m3*(h3-h4)-m2*(h2-h1)/0.78/0.81

ROLLS ROYCE ALLISON 250 The following spreadsheets outline the effects of the M-Cycl on a given gas turbine.

The **BASIC BRAYTON CYCLE** gives the basic operating parameters of the cycle. The data presented is for the operation of the Allison 250 using the Brayton Cycle.

BASIC BRAYTON CYCLE



Constants		
R	53.353	lbf*ft/lbm-°R
Ratio Specific Heats	1.3	
Pressure Ratio	7	
Compression Ratio	4.468	
Fuel Flow	0.05	lb/sec
Fuel LHV	18300	Btu/lb
c_p @ 2000°F	0.410	Btu/lb-°F

State 3- Combustion		
T3	972	°F
P3	102.872	psia
Humidity Ratio	0.0086%	
Mass Flow	3.600	lb/sec
v3A	3.500	ft^3/lb
Saturation Pressure	12594	psia
W	0.006604	lb water / lb dry air
Dew Point	710.490	°F
Enthalpy	346.1	Btu/lb

State 1 - Compressor Inlet		
T1	60	°F
P1	14.696	psia
Humidity Ratio	60%	
Inlet Mass Flow	3.55	lb/sec
v1	13.102	ft^3/lb
Saturation Pressure	0.25634	psia
W	0.006604	lb water / lb dry air
Dew Point	46.150	°F
Enthalpy	21.583	Btu/lb

State 4 - Turbine Exhaust		
T4	454	°F
P4	14.696	psia
Humidity Ratio	0.0351%	
Mass Flow	3.600	lb/sec
v3A	11.444	ft^3/lb
Saturation Pressure	440	psia
W	0.006604	lb water / lb dry air
Dew Point	390.798	°F
Enthalpy	117.277	Btu/lb

State 2 - Compressor Outlet		
T2	355	°F
P2	102.872	psia
Humidity Ratio	0.7527%	
Mass Flow	3.55	lb/sec
v2	2.933	ft^3/lb
Saturation Pressure	143.605	psia
W	0.006604	lb water / lb dry air
Dew Point	207.356	°F
Enthalpy	93.248	Btu/lb

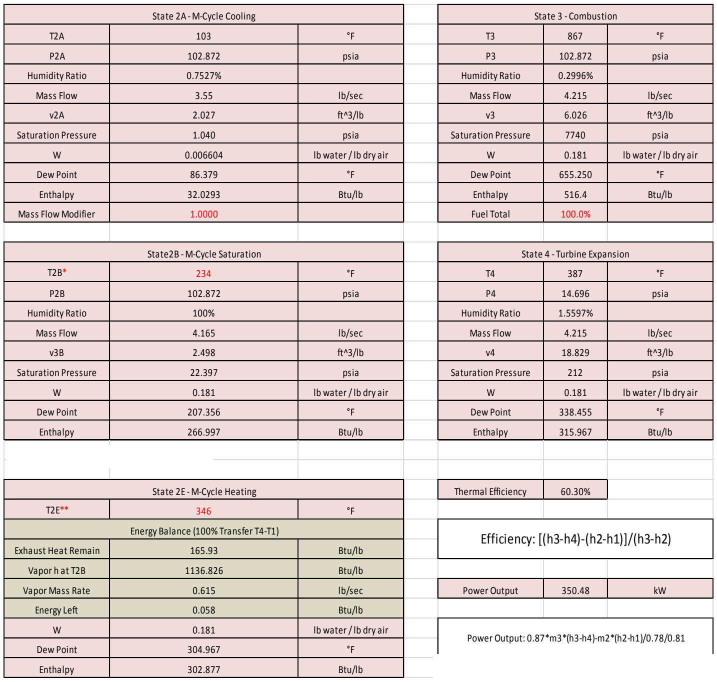
Thermal Efficiency	36.13%

Efficiency: 1-(T4-T1)/T3-T2)

Power Output	231.15	kW

Power Output: 0.87*m3*(T3-T4)-m2*(T2-T1)/0.78/0.81

EXAMPLE 2: M-CYCLE EXAMPLE (IDENTICAL AIR FLOW RATE)



State 2A - M-Cycle Cooling		
T2A	103	°F
P2A	102.872	psia
Humidity Ratio	0.7527%	
Mass Flow	3.55	lb/sec
v2A	2.027	ft^3/lb
Saturation Pressure	1.040	psia
W	0.006604	lb water / lb dry air
Dew Point	86.379	°F
Enthalpy	32.0293	Btu/lb
Mass Flow Modifier	1.0000	

State 3 - Combustion		
T3	867	°F
P3	102.872	psia
Humidity Ratio	0.2996%	
Mass Flow	4.215	lb/sec
v3	6.026	ft^3/lb
Saturation Pressure	7740	psia
W	0.181	lb water / lb dry air
Dew Point	655.250	°F
Enthalpy	516.4	Btu/lb
Fuel Total	100.0%	

State2B - M-Cycle Saturation		
T2B*	234	°F
P2B	102.872	psia
Humidity Ratio	100%	
Mass Flow	4.165	lb/sec
v3B	2.498	ft^3/lb
Saturation Pressure	22.397	psia
W	0.181	lb water / lb dry air
Dew Point	207.356	°F
Enthalpy	266.997	Btu/lb

State 4 - Turbine Expansion		
T4	387	°F
P4	14.696	psia
Humidity Ratio	1.5597%	
Mass Flow	4.215	lb/sec
v4	18.829	ft^3/lb
Saturation Pressure	212	psia
W	0.181	lb water / lb dry air
Dew Point	338.455	°F
Enthalpy	315.967	Btu/lb

State 2E - M-Cycle Heating		
T2E**	346	°F
Energy Balance (100% Transfer T4-T1)		
Exhaust Heat Remain	165.93	Btu/lb
Vapor h at T2B	1136.826	Btu/lb
Vapor Mass Rate	0.615	lb/sec
Energy Left	0.058	Btu/lb
W	0.181	lb water / lb dry air
Dew Point	304.967	°F
Enthalpy	302.877	Btu/lb

Thermal Efficiency	60.30%

Efficiency: [(h3-h4)-(h2-h1)]/(h3-h2)

Power Output	350.48	kW

Power Output: 0.87*m3*(h3-h4)-m2*(h2-h1)/0.78/0.81

DISCUSSION

The **FIRST EXAMPLE** shows the M-Cycle with the fuel flow rates the same as in the Basic Brayton Cycle. This results in an increase in efficiency of 36.7% and a 42.2% increase in power over the Basic Brayton Cycle. These improvements result because of improved heat exchange process in the M-Cycle heat exchanger, lower compressor exit temperature (increase in compressor efficiency), and the lowering of the turbine exhaust temperature (larger expansion ratio).

The **SECOND EXAMPLE** shows the M-Cycle with identical inlet air flow rates compared to the Basic Brayton Cycle. This results in an increase in efficiency of 40% and an increase in power output of 34%. The increase in power results because of the increase in fuel flow, but the increase in efficiency results because of the better utilization of the heat transfer in the M-Cycle exchanger, and the lower compressor operating temperature.

CONCLUSIONS

The M-Cycle could be used for many applications in the power industry to increase the overall system efficiencies. There are several modes of operation of the cycle that can be considered. There are two evaluated in this paper, but several other are being evaluated and will be discussed in a later paper.

The Constant Air Flow (CAF) and the Constant Fuel Flow (CFF) modes of operation offer insight into the operating characteristics of the M-Cycle coupled to a gas turbine. Optimization techniques have been used to determine the most efficient operating points for the turbine-M-Cycle combination. In the CAF mode of operation, the combined system efficiency was 60.3%. This is a significant improvement over conventional gas turbine efficiencies. In the CFF mode of operation, the combined system efficiency was 57%.

The flexibility of the M-Cycle for use with gas turbines can be found in performance improving applications, such as, to cool compressor intercoolers, to saturate the compressed air while reducing the temperature of the exhaust gases toward the dew point temperature of the compressed air, and to saturate the fuel with water vapor while further cooling the turbine exhaust gases.

The M-Cycle is a thermodynamic development for energy recovery, which utilizes the an efficient heat (enthalpy) recovery process for any combustion engine. It is possible to significantly increase the thermal efficiency for the turbine (> 50%). The gas turbine, using humidified compressed air through the M-Cycle, becomes less than half the size of a standard turbine to generate the same amount of power. This is possible because much more of the power produced by the turbine goes to the drive train rather than to compress air as in a standard turbine because of the lower cycle operating temperatures. In addition and because of the higher efficiency in this M-Cycle turbine, pollution is dramatically reduced due to the water vapor creating a more even burning process during combustion. Additional water will not be needed because the M-Cycle heat exchanger constantly recycles water in the exhaust gas.

The new M-Cycle gas turbine could provide the following improvements to a gas turbine sytem:

1. Approximately double fuel efficiency compared to a simple gas turbine.
2. Reduce the CO_2 by approx.half
3. Could reduce NOx related pollution by up to 90% (no hot spots).
4. Could reduce the size of the gas turbine by approx. half for the same usable shaft power out.
5. Require a recuperative plate heat exchanger.
6. Require an air-cooled condenser.
7. Creates a much wider power band.

REFERENCES

[1] Dalili, F., and Westermark,M., First Experimental Results on Humidification of Pressurized Air in Evaporative Power Cycles, Dept. of Chemical Engr., Royal Institute of Technology, Stockholm, Sweden, 2001
[2] Rao, A.D., Process for Producing Power, US Patent No. 4,829,763, 1989.
[3] Cohn, A., Scheibel, J, Current Gas Turbine Development and Future Projects, EPRI, Palo Alto, CA, 1997.
[4] Kohlenberg, C.R. Method and Apparatus for Cooling the Inlet Air of Gas Turbine and Internal Combustion Engine Prime Movers, US Patent No. 5,790,972.
[5] Gillan, L, and Maisotsenko, V, Maisotsenko Open Cycle Used for Gas Turbine Power Generation, GT 2003-38080, ASME/IGTI Turbo Expo, June 16-19, 2003.