UNDERSTANDING THE EFFECTS OF ENVIRONMENTAL CONDITIONS AT MY COMBINED CYCLE POWER PLANT

By Miguel de la Cruz

Combined cycle power plant owners and operators understand that, on any given day, the performance of their power plant will vary with ambient conditions surrounding the plant. It is especially useful to understand, for a given set of ambient conditions, how much of a performance variation can be expected. This gives us insight on how much power we could be expecting to deliver to the grid on any given day, and on the cost of production for that day.

To even start trying to predict the change in performance of a power plant under any set of ambient conditions, we must first understand how and why the performance of the power plant is affected by each of these ambient parameters. That will be the primary focus of this series of articles, we will attempt to gain an intuitive understanding of WHY and HOW ambient conditions fluctuations influence performance. Along the way, we will try to better understand why it is that some ambient conditions affect performance in a dramatic way, while some other ambient conditions not so much.

We will focus on the two main primary power plant performance indicators that every power plant stakeholder is interested in: net power output (how much energy can we deliver to the grid?), and net efficiency (how much does this energy cost us to produce?). Also, we will develop some formulations that we can use to start predicting the performance fluctuations of a power plant at different atmospheric conditions.
In our attempt to understand these effects, we will use common thermodynamic formulations, and as our analysis becomes more complex, we will rely on some sophisticated computer simulations to try and better understand these behaviors.

The analysis will be done assuming that the reader already has a basic understanding of thermodynamic processes, but not much more. We will begin from the thermodynamic fundamentals of the combined cycle power plant, and methodically build on top of this foundation as we look at more atmospheric effects.

All in all, due to the amount of information and detail that we will tackle, this will be a long journey. Therefore, we will split this series into four parts, each covering a major component of the power plant. We first start by looking at what is the main driver in a fossil fuel power plant: the gas turbine. We will spend some time explaining the effects of atmospheric conditions on gas turbines since these effects are a major influence in the performance of combined cycles. We will then explore how these performance variations cascade into the bottoming cycle and the domino effects they have there. We will then take a detour into the performance of the different types of steam turbine condensers, and finally, we will put all this together into an overall power plant performance analysis. Each part attempting to build on top of the foundations of the previous.

Part 1: The Gas Turbine
The ideal Brayton Cycle

We begin by looking at the prime mover of any combined cycle: the gas turbine. A GT is a constant volume heat engine that works under the principle of the Brayton thermodynamic cycle. This thermal cycle in its ideal form consists of four thermodynamic processes, which are the following:

- Isentropic compression. The air, or working fluid, enters the gas turbine at a pressure P_1, the compressor will then do work on this air and compress to a given discharge pressure P_2. During this process, the work done by the compressor is transferred to the working fluid, in such a way that this work is used to increase the air pressure. At this point, the exact manner in which the compressor does the actual compression is not important. What is important to mention is that this compression process is what is called and isochoric one. Meaning that for a fixed compressor geometry and speed, the volume flow passing through the compressor will always be constant. Also important to note is that, because we are imagining an ideal process, entropy will not increase during this process, this is the definition of an isentropic process. Process line 1 -2 in Fig. 1.
- Isobaric heat input. After the air has been compressed and its internal energy will have increased. The compressed air receives and additional heat input in the combustion chamber via the fuel combustion. This enthalpy rise is usually controlled until a specific temperature T_3 is reached. The choice of T_3 will depend on the component's material constraints. The pressure during this heat input process remains

constant ($P_2 = P_3$) since air and fuel are injected into the combustion chamber at the same rate that the combustion gases leave the combustion chamber. Process line 2-3 in Fig. 1.

- Isentropic expansion. The high-pressure high-temperature working fluid is then expanded through an expansion turbine. During its expansion it surrenders its internal energy by doing work through the expansion turbine and the work is collected in the turbine shaft. Some of this work is used to drive the compressor which is attached to the expansion turbine. The rest is used to run the electrical generator. At the end of this process, the working fluid is expanded back to the initial pressure P_1. Process line 3-4 in Fig. 1.
- Isobaric heat rejection. After the fluid has done work upon the turbine, it will be left at a lower energy level. The expansion will bring the fluid back to the initial pressure ($P_4 = P_1$), and there will be some leftover heat in the fluid which will need to be rejected by cooling it down into a heat sink. This process will happen at constant pressure. By the end of this process, the working fluid will be back at state 1, and the entire cycle will commence again. Process line 4-1 in Fig. 1.

A useful way of representing this sequence of processes is to plot them in an Enthalpy-Entropy diagram, also known as an h-S diagram. Figure 1 shows exactly this for an imaginary ideal gas turbine. Let's try to analyze this chart and try to gain some insight into the thermodynamic relationships that it shows.

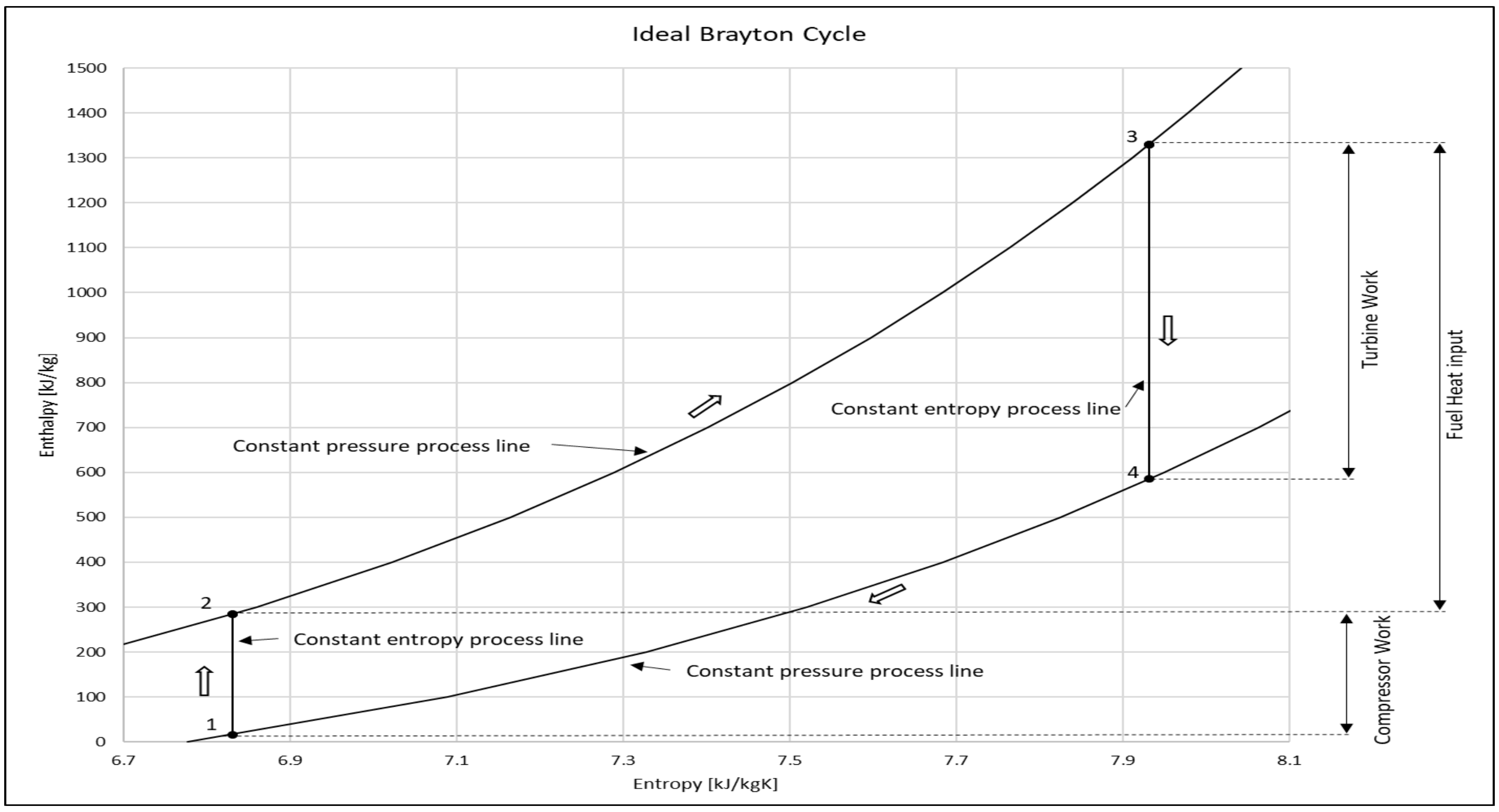

Figure 1. Ideal Brayton cycle, air as working fluid (T_3 = 1200 °C, P_1 = 1atm, P_2/P_1 = 10)

Figure 1 shows an h-S diagram of a sample ideal Brayton cycle that uses air as the working fluid. For this example, we have constrained the cycle by defining two key operation parameters: the turbine inlet temperature (TIT = T_3), and the compressor pressure ratio (P_2/P_1). A gas turbine designer initially chooses these parameters and designs the turbine around them, so it is reasonable for us to choose a TIT and pressure ratio to analyze our cycle as well.

The four thermodynamic processes of the Brayton cycle are plotted as lines connecting four states of the working fluid. We consider state 1 the initial condition of the air, we can imagine this as the intake air of the compressor. State 2 is the compressed state at the compressor discharge, state 3 is the high-pressure high-temperature state at the outlet of the combustion chamber, and state 4 is the discharge condition after the expansion turbine. Note that the cycle runs in a clockwise manner as shown by the arrows, the working fluid starts in state 1 (lower-left corner), and moves along the process lines through states 2, 3 and 4, before returning to state 1 and closing the cycle.

For this example let us say that the air in state 1 is at ISO conditions, meaning that the air temperature is 15 °C, and at one standard atmosphere of pressure, that is 101.325 kPa. Since we know the air temperature and pressure, we can calculate the enthalpy and entropy values of state 1 since these are functions of state.

The compression process is represented by the vertical line 1-2. Remember that the ideal compression process is isentropic, and therefore the process line must be vertical in this type of diagram. Since we have constrained the cycle by defining the pressure ratio P_2/P_1, we can easily find state 2 by moving vertically from state 1 and finding the intersection point with the P_2 isobaric process line. The vertical distance between state 1 and state 2 (the enthalpy difference) is equal to the compressor work.

The next process that occurs in our ideal gas turbine is the heat input that happens in the combustion chamber, we know this to be a constant pressure process, and therefore we must follow the isobaric process line and stop when we reach the enthalpy of air corresponding to our target T_3. Using the properties tables of air we know that when T_3 = 1200 °C and P_3 = 101.325 kPa the enthalpy of air is 1,330 kJ/kg. And so state 3 is found at the intersection point of the isobaric process line and this enthalpy value.

Finally, we know that the expansion process 3-4 is an isentropic process. Therefore, we once again move vertically in the chart. This time we move in a downward direction to intersect with the isobaric process line for pressure P_4 (which is the same as P_1). Just like that, we have now a completely defined thermodynamic cycle in our h-S diagram.
Here are some interesting things to notice in this diagram that may not be immediately obvious:

1. The isobaric process lines represent the thermal properties of the working fluid, in this case air, at a given pressure. There is an infinite amount of possible isobaric process lines, in our case we are only interested in the ones for the pressures that we use in our cycle. These process lines will continue extending well beyond the end margins of our diagram.
2. We can see that as we move further to the right side of the diagram into higher entropy states, the constant pressure process lines diverge from each other. It's because of this behavior that the enthalpy difference between states 1 and 2 (compressor work) is lower than the enthalpy difference between states 3 and 4 (expansion work). It logically follows that the further we increase T_3 (thereby increasing h_3), more work could be extracted in process 3-4, because states 3 and 4 would have diverged further from each other.
3. By the same logic, we can see that the compressor always does less work on the air than the air does on the turbine ($(h_2 - h_1) < (h_3 - h_4)$). This is the entire reason why a gas turbine can generate any useful work.
4. The entropy and enthalpy values are in a specific mass basis. That is to say that all the enthalpy and entropy values are per unit mass of air. For example in Figure 1 the enthalpy rise in the compressor is 268 kJ/kg ($h_2 - h_1 = 268$ kJ/kg), this means that for every unit of mass that passes through the compressor, the compressor must do 268 kJ of work to take it from state 1 to state 2. The same logic can be used when evaluating the work output of the turbine.

We can develop a few simple equations to define the performance of the cycle in more precise terms. As explained in point 4 above, the enthalpy differences are expressed in terms of unit mass. Therefore, we will express all our equations in specific terms, i.e. per unit mass.

The compressor specific work can be calculated as:

$$CW = (h_2 - h_1) \tag{1}$$

Where
CW is the specific compressor work needed to compress the working fluid and take it from state 1 to state 2 (kJ/kg),
h_2 is the specific enthalpy of the working fluid entering the compressor (state 2) (kJ/kg), and
h_3 is the specific enthalpy of the working fluid leaving the compressor (state 1) (kJ/kg).

The specific heat input to the cycle in process 2-3 is given by:

$$HI = (h_3 - h_2\ ) \tag{2}$$

Where
HI is the specific heat input required to raise the enthalpy of the working fluid and take if from state 2 to state 3 (kJ/kg),
h_2 is the specific enthalpy of the working fluid entering the combustion chamber (state 2) (kJ/kg), and
h_3 is the specific enthalpy of the working fluid leaving the combustion chamber (state 3) (kJ/kg).

We have simplified equation 2 by assuming that the mass flow rate at the inlet and outlet of the combustion chamber is the same ($W_2 = W_3$). In fact, $W_3 > W_2$ because a certain mass of fuel needs to be added during the combustion process. The mass of fuel is usually very small compared to the mass of air. For now, let's accept this simplification and move on.

The specific work of the isentropic expansion of the combustion gases through the expansion turbine is given by:

$$EW = (h_3 - h_4) \tag{3}$$

Where
EW is the specific work done by the fluid as it expands from state 3 to state 4 (kJ/kg),
h_3 is the specific enthalpy of the working fluid entering the expansion turbine (state 3) (kJ/kg), and
h_4 is the specific enthalpy of the working fluid leaving the expansion turbine (state 4) (kJ/kg).

Since the expansion turbine drives the compressor, the specific useful work that remains after the compressor work is discounted, and that is ultimately available to drive the generator is given by:

$$SUW = EW - CW \quad (4)$$

Where
SUW: is the specific useful work of the gas turbine (kJ/kg).
If we know the mass flow rate of the gas turbine, we can also calculate the useful power (UP), given by:

$$UP = SUW * W \quad (5)$$

Where
W: the mass flow rate of the turbine (kg/s).
UP: is the useful power output of the gas turbine, in standard power units (kJ/s).

Finally, the thermal efficiency of the turbine (ETA) is the ratio of the specific useful work and the specific heat input. It is given by:

$$ETA = \frac{SUW}{HI} * 10 \quad (6)$$

Previously we have mentioned that a gas turbine is a constant volume engine, let's take a minute now to explain what this means. For starters, we can assume that for every revolution that the turbine takes, a certain volume of air will be ingested into the compressor. The amount of volume will depend only on the physical dimensions of the turbine. Since these physical dimensions do not change (we're ignoring VIGVs for now), the volume admitted per revolution is constant. Since a gas turbine normally operates at a constant rotational speed, we can safely assume that the volume flow rate will also be constant. We know from the equations above, however, that the performance of the gas turbine is a function not of the volume flow rate, but of the mass flow rate. Thus, we can see that gas turbine performance, specifically Useful Power, will also be a function of fluid density. Let's keep this fact in mind as we continue our analysis.

Effect of ambient temperature on the ideal Brayton Cycle

We can now start to think about how variations of ambient conditions (variations of state 1) will affect the performance of our ideal gas turbine. Let's start with a change in ambient temperature (T_1). For now, let us assume that P_1 remains the same (i.e. atmospheric pressure remains constant). What would be the effect of increasing or decreasing temperature T_1? Will a change in T_1 affect useful power and thermal efficiency?

Let us imagine that the ambient air temperature for our ideal gas turbine from Figure 1 increases to 30 °C on a particularly hot day. Also, let's imagine that on a different, particularly cold day the temperature of the air drops to 0 °C. If we calculate the enthalpy and entropy state functions for air at these ambient conditions, and we would discover that on the hot day the air enthalpy is higher, and in the cold day the air enthalpy is lower. We would find out that the entropy of air during the cold day is lower than the entropy of the air during the hot day. This means that in the hot day state 1 will shift up and to the right in the h-S diagram, and in the cold day state 1 will move down and to the left. In both cases, the new position of state 1 would be somewhere on the same isobaric process line since atmospheric pressure has not changed. This is exactly what we can see in Figure 2.

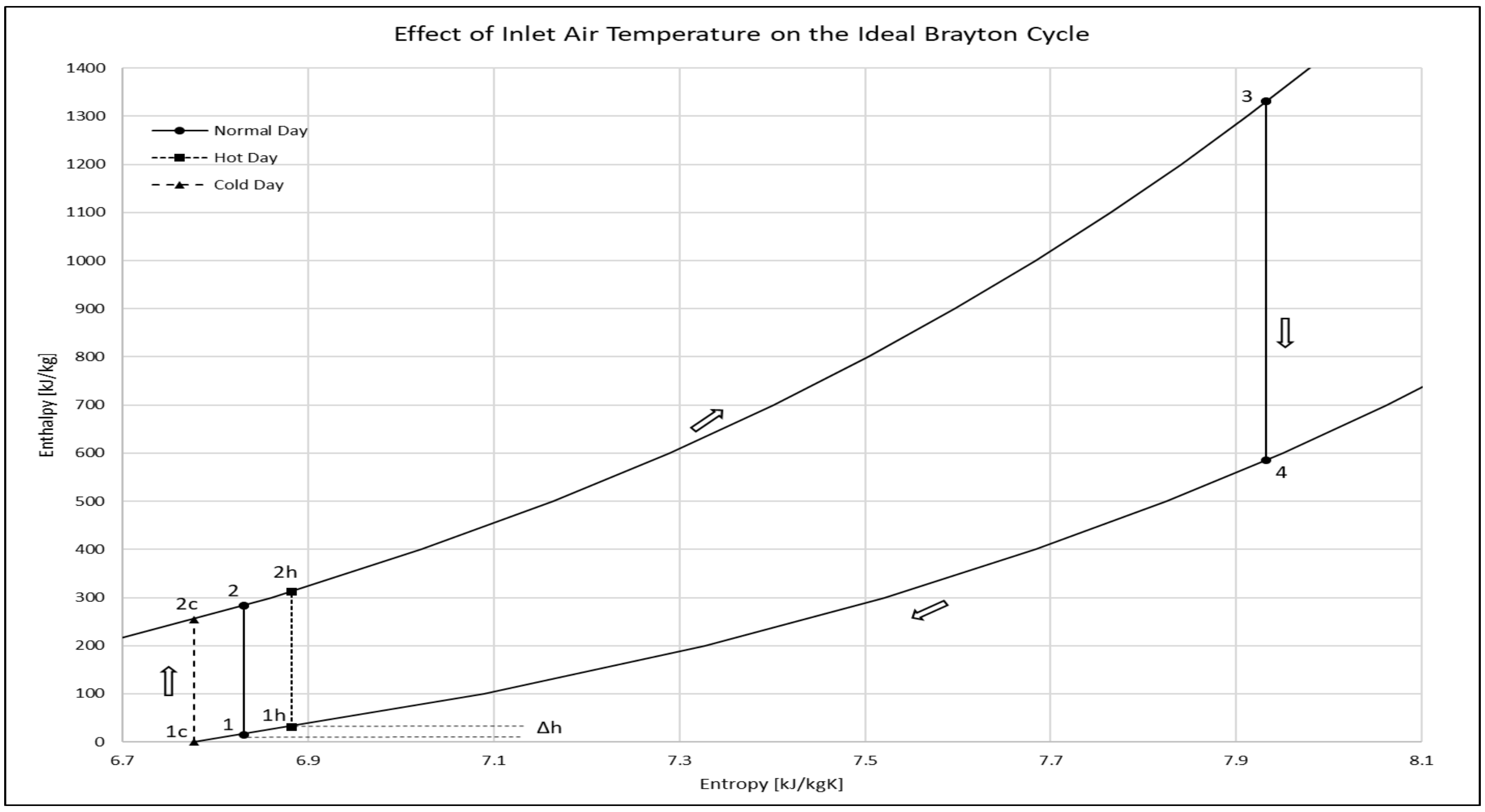

Figure 2 The ideal Brayton Cycle in a hot day and cold day, air as working fluid ($T3 = 1200$ °C, $P1 = 1$atm, $P_2/P_1 = 10$)

Note that because state 1 moves to a different entropy level, state 2 must also move to a different entropy level. Because of the exponential and diverging nature of the isobaric lines, the work required to isentropically compress the air will be lower on a cold day than on a hot day, $(h_{2c} - h_{1c}) < (h_{2h} - h_{2h})$. Also, because the expansion work in the turbine remains constant, the specific useful work will increase (thereby increasing ETA acc. to equation (6)).

Another thing to note is that since h_3 is the same both days (remember that this is an operational constant), the heat input required to bring the air to T_3 is higher in a cold day than in a hot day $(h_3 - h_{2h}) < (h_3 - h_{2c})$. In other words, because on a cold day the compressor discharge air temperature is lower, more heat input is required to raise its temperature to T_3 (thereby decreasing ETA acc. to equation (6)).

Putting it all together, we can see that on a cold day the thermal efficiency tends to increase due to the higher specific useful work, and at the same time tends to decrease due to the higher heat input requirement. It just so happens that for an ideal gas, these two effects perfectly cancel each other out, and thus thermal efficiency is unaffected (a more detailed explanation will come later, for now just trust that this is correct).

Something else to note that is not immediately obvious from Figure 2, as the air temperature changes necessarily so will its density. Since we have already established that the volume flow rate of the turbine is constant, the mass flow rate will only change in the same proportion of the density change. On a colder day, the density of the air will be higher than on a hotter day, therefore a larger mass flow will pass through the compressor and through the expansion turbine. Since useful power is a function of the mass flow (see equation (5), a gas turbine will generate more useful power in a colder day, and less useful power on a hotter day.

In Table 1 the performance equations are developed for this imaginary gas turbine working in a normal 15 °C day, a cold 0 °C day, and a hot 30 °C day. The difference between the cold day and the normal day is an increase of 8.5% power output. Of this 8.5%, we can attribute approximately 3% to the increased specific useful work, and 5.5% to the increase in air density.

Table 1: Performance table for the ideal Brayton cycle on a cold day, a normal day, and a hot day.

Parameter	Units	Cold Day	Normal Day	Hot Day
Cycle Parameters				
P_1	kPa	101.325	101.325	101.325
P_2/P_1	-	10.0	10.0	10.0
T_1	°C	0.0	15.0	30.0
T_2	°C	250.9	278.8	306.4
T_3	°C	1,200.0	1,200.0	1,200.0
T_4	°C	559.6	559.6	559.6
h_1	kJ/kg	0.0	15.1	30.2
h_2	kJ/kg	254.9	283.8	312.7
h_3	kJ/kg	1,330.5	1,330.5	1,330.5
h_4	kJ/kg	585.4	585.4	585.4

Parameter	Units	Cold Day	Normal Day	Hot Day
Cycle Performance				
CW	kJ/kg	254.9	268.8	282.5
HI	kJ/kg	1,075.6	1,046.7	1,017.8
EW	kJ/kg	745.1	745.1	745.1
SUW	kJ/kg	490.2	476.3	462.6
W	kg/s	158.2	150	142.6
UP	kJ/s	77,563	71,452	65,952
ETA*	%	45.57%	45.51%	45.45%

*Note: It can be proven that for an ideal Brayton cycle working with an ideal gas, the efficiency of the cycle at a constant T_3, is only a function of P_2/P_1. Since in this example the pressure ratio is unchanged, the efficiency of the cycle is unchanged. The very small differences seen here are due to the fact that real air does not behave as an ideal gas.

To summarize the effects of ambient temperature on the cycle performance:

1. If the ambient temperature decreases, the specific useful work of the gas turbine increases. If the ambient temperature increases, the specific useful work of the gas turbine decreases.
2. If the ambient temperature decreases, the mass flow rate of the gas turbine increases, which increases the useful power. If the ambient temperature increases, the mass flow rate of the gas turbine decreases, which decreases the useful power.
3. In the ideal cycle, the efficiency of the gas turbine does not depend on the ambient temperature.

The effect on efficiency

The reasoning behind item number 3 above is not immediately obvious. Let us take a moment now to try to understand why thermal efficiency has remained unaffected since this will tie into our discussion on the effects on performance of the other ambient conditions.

For an ideal gas undergoing an isentropic process, we can derive from the ideal gas equation that:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad (7)$$

And because the heat injection and rejection processes are isobaric processes, equation (7) can be expanded:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} \qquad (7a)$$

By expanding and rewriting equation (4) in terms of the state function of enthalpy, the equation takes the following form:

$$UW \;=\; Cp(T_3 - T_4) - Cp(T_2 - T_1) \;=\; Cp * T_3\left(1 - \frac{T_4}{T_3}\right) - T_2\left(1 - \frac{T_1}{T_2}\right) \qquad (8)$$

Substituting equation (7a) into (8) and rearranging we obtain:

$$UW = Cp * (T_3 - T_2)\left(1 - \frac{T_1}{T_2}\right) \qquad (9)$$

And from (6) and (9) we can finally obtain,

$$ETA = \frac{UW}{HI} = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad (10)$$

And so we arrive at an important fact of the Brayton cycle when working with an ideal gas, and that is that for a given T_3, the thermal efficiency of the cycle is a function of the pressure ratio only. Since the pressure ratio has remained constant in all our examples so far, the thermodynamic efficiency of our turbine will not change. We have seen in the performance data tables that the efficiency does vary very slightly, this is because real air does not behave like an ideal gas.

Effect of atmospheric pressure on the ideal Brayton Cycle

Let us now imagine our ideal gas turbine finds itself operating at a normal atmospheric pressure one day, and a slightly higher atmospheric pressure the next day. Assuming that T_1 is the same on both days, what would be the effect of this atmospheric pressure change in our gas turbine?

Equation (10) simplifies our work significantly because we know that efficiency only depends on the pressure ratio of the compressor, which for this example has remained constant. Therefore we know that atmospheric pressure will not affect the thermal efficiency of the gas turbine.

On the other hand, the one variable that *does* change as atmospheric pressure changes is air density. We know from the ideal gas equation that higher pressure will result in a higher density. Therefore, we can deduce that with higher atmospheric pressure, the mass flow of our turbine will increase and thus useful power too. Table 2 shows the cycle parameters for the same gas turbine, at three different atmospheric pressure, clearly as we have predicted, the only effect on performance is due to air density.

Table 2: Performance table for the ideal Brayton cycle in a low-pressure day, normal day, and high-pressure day.

Parameter	Units	Low-Pressure Day	Normal Day	High-Pressure Day
Cycle Parameters				
P_1	kPa	100.825	101.325	101.825
P_2/P_1	-	10.0	10.0	10.0
T_1	°C	15.0	15.0	15.0
T_2	°C	278.8	278.8	278.8
T_3	°C	1,200.0	1,200.0	1,200.0
T_4	°C	559.6	559.6	559.6
h_1	kJ/kg	15.1	15.1	15.1
h_2	kJ/kg	283.8	283.8	283.8
h_3	kJ/kg	1,330.5	1,330.5	1,330.5
h_4	kJ/kg	585.4	585.4	585.4
Cycle Performance				
CW	kJ/kg	268.8	268.8	268.8
HI	kJ/kg	1,046.7	1,046.7	1,046.7
EW	kJ/kg	745.1	745.1	745.1
SUW	kJ/kg	476.3	476.3	476.3
W	kg/s	149.26	150	150.74
UP	kJ/s	71,099	71,452	71,804
ETA	%	45.51%	45.51%	45.51%

To summarize the effects of atmospheric pressure on the cycle performance:

1. If atmospheric air pressure increases, the mass flow rate of the gas turbine increases, which increases the useful power. If the atmospheric pressure decreases, the mass flow rate of the gas turbine decreases, which decreases the useful power.
2. In the ideal cycle, the specific useful work of the gas turbine does not depend on the atmospheric pressure.

3. In the ideal cycle, the thermal efficiency of the gas turbine does not depend on the atmospheric pressure.

Effect of ambient relative humidity on the ideal Brayton Cycle

The next atmospheric condition that we will analyze is the moisture content of the atmospheric air. Until now we have assumed that our imaginary ideal gas turbine works on dry air. In reality, air will almost always include a percentage of moisture. This moisture content is usually measured in terms of the relative humidity of the air, or in terms of the wet-bulb temperature. For a given ambient air temperature and pressure, as the water content in the air increases, the air enthalpy increases and density decreases.

By now we already know that a decrease in density will translate into a reduction of useful power, but what will be the effect of the increased enthalpy? To help us answer this question, the effect of relative humidity on the ideal Brayton cycle is graphically shown in Figure 3. We can tell from this diagram that as the moisture content increases, the enthalpy increases (albeit too slightly to be noticeable in this chart), and that the whole cycle seems to shift rightwards. We can also see that the compressor work and turbine expansion work appear to remain the same, only shifted to higher entropy states.

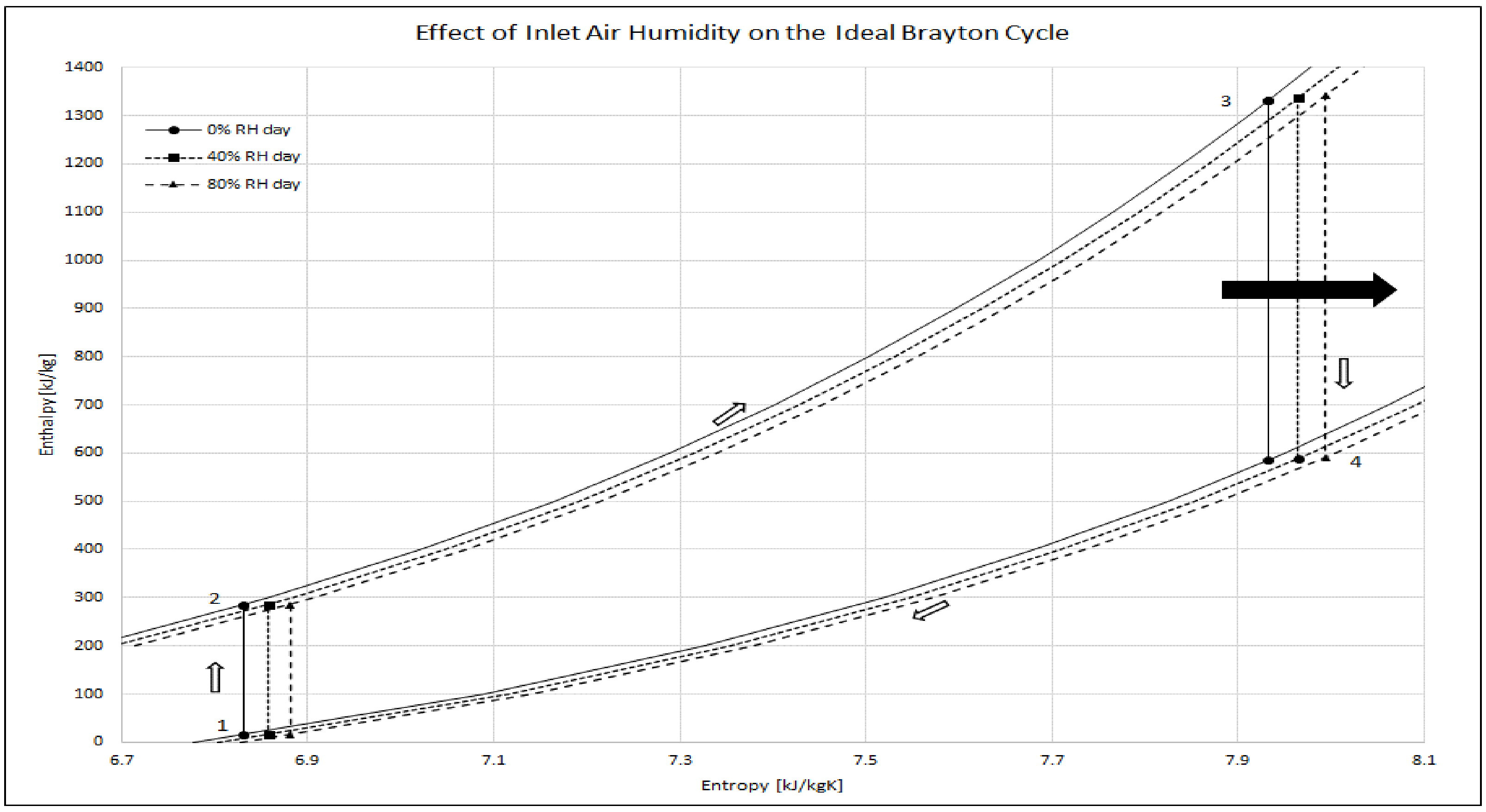

Figure 3 Effect of air humidity on the ideal Brayton cycle

By reading Table 3 we can see the decreased mass flow that we would be expected as moisture increases. Surprisingly, useful power shows small gains even as mass flow decreases. It turns out that as humidity increases, the expansion work in the turbine increases more than the compression work does, and therefore the specific useful work is higher. This improvement in specific useful work makes up for the reduced mass flow and ends up giving a small power output benefit. Still, the improvement is too small to be significant.

Table 3: Performance table for the ideal Brayton cycle in with dry air, and humid air.

Parameter	Units	Normal Dry Day	40% relative humidity day	80% relative humidity day
Cycle Parameters				
P_1	kPa	101.325	101.325	101.825
P_2/P_1	-	10.0	10.0	10.0
T_1	°C	15.0	15.0	15.0
T_2	°C	278.8	278.8	278.8
T_3	°C	1,200.0	1,200.0	1,200.0
T_4	°C	559.6	559.6	559.6
h_1	kJ/kg	15.1	15.1	15.2
h_2	kJ/kg	283.8	284.5	285.1

Parameter	Units	Normal Dry Day	40% relative humidity day	80% relative humidity day
h_3	kJ/kg	1,330.5	1,336.1	1,341.7
h_4	kJ/kg	585.4	588.7	592.0
Cycle Performance				
CW	kJ/kg	268.8	269.3	269.9
HI	kJ/kg	1,046.7	1,051.6	1,056.6
EW	kJ/kg	745.1	747.1	749.7
SUW	kJ/kg	476.3	478.1	479.8
W	kg/s	150	149.6	149.2
UW	kJ/s	71,452	71,526	71,600
ETA	%	45.51%	45.46%	45.41%

To summarize the effects of ambient air humidity on the cycle performance:

1. If ambient humidity decreases, the mass flow rate of the gas turbine increases, which increases the useful power. If the ambient humidity increases the opposite happens.
2. In the ideal cycle, the efficiency of the gas turbine does not depend on the ambient humidity.

The real Brayton Cycle at design conditions

At this point, we have a very good understanding of the effects of ambient conditions on the ideal Brayton cycle. Obviously, a real gas turbine will never achieve the ideal Brayton cycle. The viscous effects of real air manifest themselves and add some irreversibilities to the thermodynamic processes. These can be in the form of vortex flows in the compressor and expansion turbine, tip clearance leakages, pressure losses in the combustion chamber, pressure losses at the compressor inlet caused by air filters and ducting, pressure losses at the exhaust duct of the turbine, blade boundary layer separation, etc. All of this without even beginning to consider operational losses like extractions and auxiliary system power requirements.

A real Brayton cycle as it happens in a real gas turbine would look something more like what is shown in Figure 4.

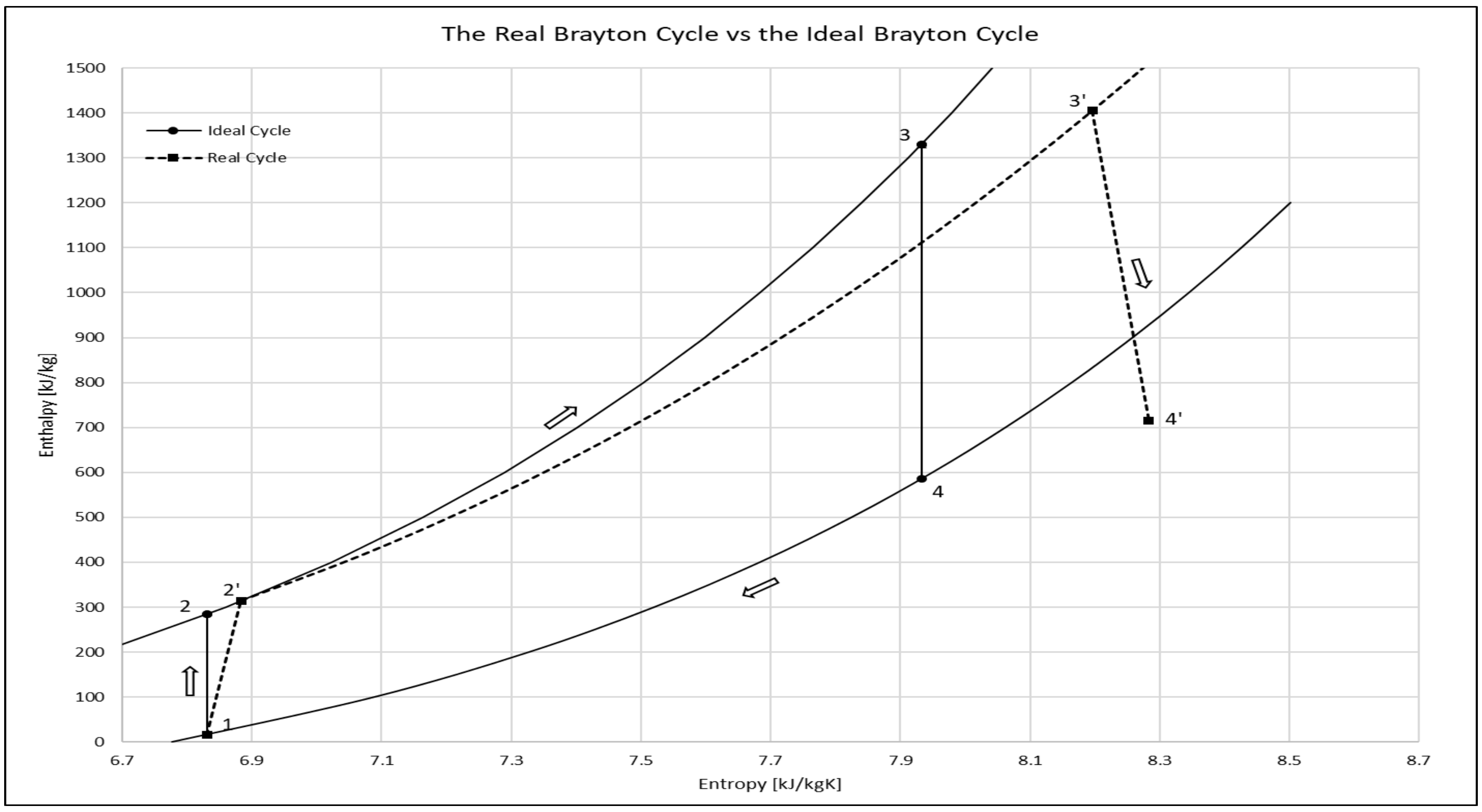

Figure 4 The ideal Brayton cycle and the real gas turbine Brayton cycle

Analyzing Figure 4, we can immediately highlight some obvious differences.

- Compression process: Because the compression process is not isentropic, the compressor needs to do additional work on the air to make up for these losses and still bring the air to the required compressor discharge pressure. In the h-S diagram, this is shown as process line 1-2'.
- Heat addition: In a real gas turbine, the heat input process is done via the combustion of a fuel. Fuel is injected in the combustion chamber, and the combustion reaction changes the molar composition of the combustion gases. This change in molar composition also changes the specific heat capacity of the gas, and therefore

the enthalpy of this gas that corresponds to the target T_3 will be different than that of pure air. The isobaric process line is changed because of the change in the working fluid composition, similar to the effect we saw when discussing the effects of moisture content of the air.

- Gas expansion in power turbine: the expansion process is not isentropic, a certain amount of entropy is generated as a function of the expansion turbine expansion efficiency. Therefore, the discharge of the gases at the exhaust of the expansion turbine will be at a higher enthalpy and higher temperature. We can think of this as work that the expansion turbine could not extract from the working fluid before reaching the exhaust pressure.
- Duct losses: not immediately obvious in Figure 4, are the pressure losses in the combustion chamber, $P_{2'} > P_{3'}$, and the higher exhaust pressure at state 4' due to the exhaust duct losses, $P_{4'} > P_1$. These pressure losses translate into an additional reduction of the specific useful work.
- Heat rejection: in the real implementation of the Brayton cycle with gas turbines, the exhaust mass flow is not recycled and brought back to state 1. This heat rejection process is omitted, and the exhaust gases are simply discharged into the atmosphere, or into a bottoming cycle heat recovery system. The Brayton cycle as seen in a real gas turbine is an open thermodynamic cycle.

Table 4: Performance table for the ideal Brayton cycle, and a real Brayton cycle.

Parameter	Units	Ideal cycle	Real cycle
Cycle Parameters			
P_1	kPa	101.325	101.325
P_2/P_1	-	10.0	10.0
T_1	°C	15.0	15.0
T_2	°C	278.8	307.4
T_3	°C	1,200.0	1,200.0
T_4	°C	559.6	645.2
h_1	kJ/kg	15.1	15.1
h_2	kJ/kg	283.8	313.7
h_3	kJ/kg	1,330.5	1,405.1
h_4	kJ/kg	585.4	714.9
Cycle Performance			
CW	kJ/kg/s	268.8	298.6
HI	kJ/kg/s	1,046.7	1,091.4
EW	kJ/kg/s	745.1	690.2
SUW	kJ/kg/s	476.3	391.6
W	kg/s	150	153.4
UW	kJ/s	71,452	61,058
ETA	%	45.51%	36.25%

The real Brayton Cycle at off-design conditions

So now we know how the real Brayton cycle behaves when compared to the ideal Brayton cycle at design conditions, i.e. at design ambient conditions, design pressure ratio, and design turbine inlet temperature. That still is not the whole story, we're not done yet! Another thing that we need to consider is that the compression efficiency and expansion efficiency are actually not constant! The performance of these components can vary substantially depending on the operational conditions of the gas turbine.

Mainly, the isentropic efficiency of these components is dependent on the mass flow passing through the system. As ambient conditions vary and air density varies with them, the component efficiencies will vary as well. And that is not all, to make matters even more complicated, it turns out that the pressure ratio of the compressor is also dependent on the air mass flow rate. And because from equation (10) we know that the thermal efficiency of the cycle is dependent on the pressure ratio, this means that a change in air density will end up affecting the thermal efficiency of the turbine as well.

The efficiency of the compressor and expansion turbine, and compressor pressure ratio as a function of mass flow cannot be modeled using simple thermodynamic formulations only. The ideal gas law formulations can only take us so far. The off-design performance of the compressor and expansion turbine are very heavily dependent on the physical arrangement of the components and other design choices. Therefore it becomes very difficult to accurately predict the isentropic efficiencies and pressure ratios of the gas turbine components without manufacturer data. The manufacturer supplies compressor maps, and turbine maps, to complement our thermal performance predictive calculations.

Let's take as an example the compressor component. In all of our design cases so far, we've considered a constant compressor pressure ratio, and therefore thermal efficiency of the gas turbine in those examples hasn't varied. In reality the pressure ratio is a function of the mass flow rate admitted into the compressor. An example of a typical compressor map showing

This relationship is illustrated in Figure 5. The relative mass flow will depend on the rotational speed of the shaft, and on the density of the air. It's safe to assume that during continuous stable operation the rotational speed won't vary too much since the turbine operates synchronous with the electric grid, therefore it is the density of the air that will have the dominant effect in the mass flow admitted into the turbine. The constant rotational speed lines have a very pronounced slope, specially as we move into higher mass flow values, and thus a small change in mass flow can have a big impact in the compressor pressure ratio.

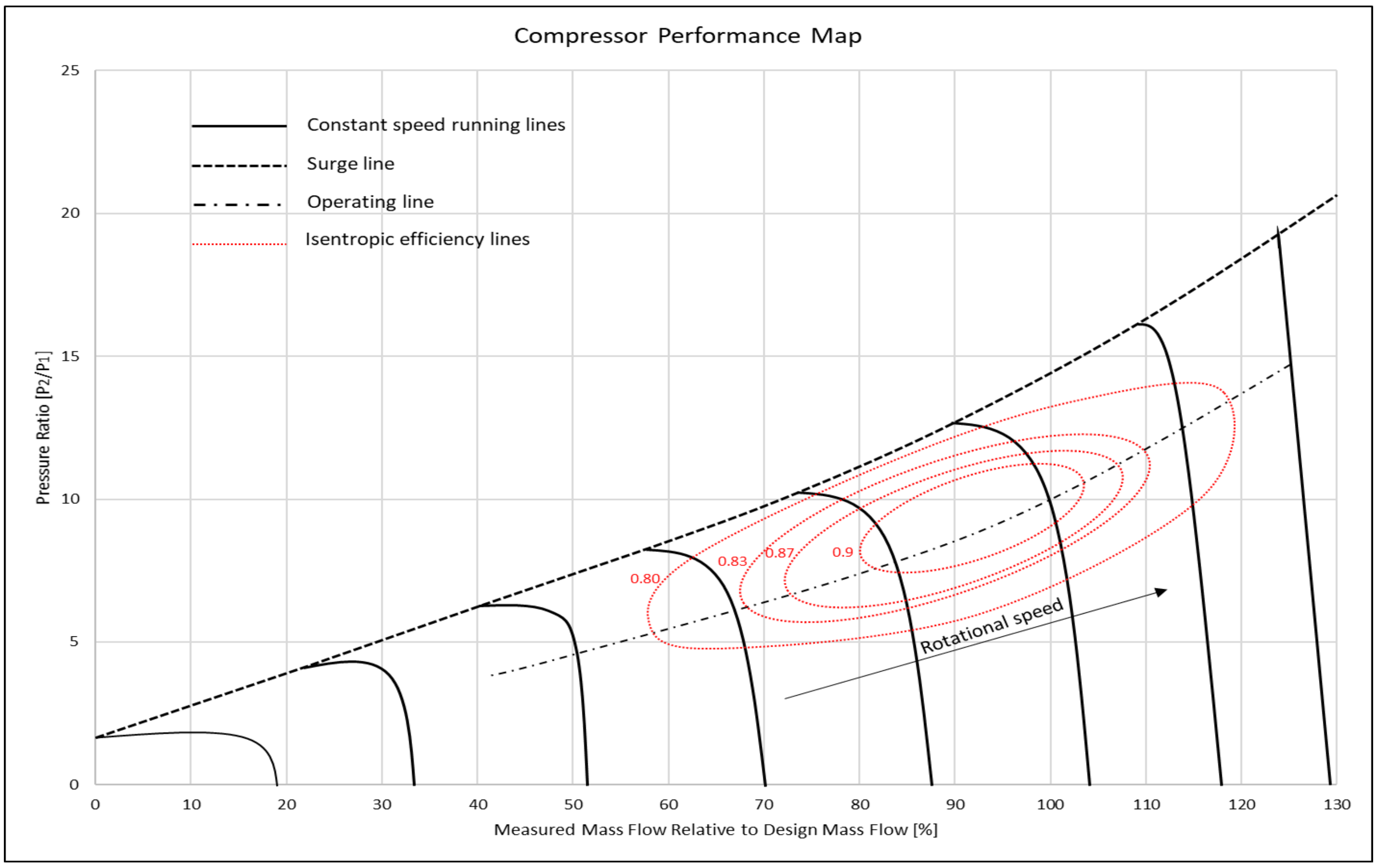

Figure 5. Sample compressor map

Note as well how the isentropic efficiency of the compressor is also a function of the mass flow. Normally the compressor is designed to reach a design pressure ratio at a design isentropic efficiency when operating at the specified design conditions, but any operation outside of these design conditions will move the operating point of the compressor to somewhere along the operating line, and therefore a different pressure ratio and isentropic efficiency. All these effects compound together, and at this point, it is easy to see that the impact in performance due to fluctuation of ambient conditions can get to be a complex matter. Clearly, to achieve a high accuracy prediction of performance, manufacturer data is required (otherwise experimentation and testing).

Normally the gas turbine manufacturer supplies what are known as performance correction curves. At their base, these are performance curves that consider all these individual component effects and combine them into one single diagram showing the overall behavior of a gas turbine for a given variation in the operational conditions. These can be variations in atmospheric conditions, but also in other conditions like grid frequency, fuel composition, generator power factor, etc. These component correction curves are developed using OEM proprietary cycle models.

To summarize the effects of ambient conditions on the real cycle performance:

1. The primary mechanism by which ambient conditions affect real cycle thermal efficiency is via the air density.
2. As air density changes, compressor and expansion turbine isentropic efficiencies will vary, which will impact the gas turbine specific useful work and specific heat input.
3. As air density changes, the compressor pressure ratio will vary, which will impact the gas turbine thermal efficiency.
4. The effects are complexly interrelated, and thus real gas turbine performance becomes difficult to predict without sufficient manufacturer data.
5. GT performance at different ambient operating conditions are estimated using OEM correction curve.