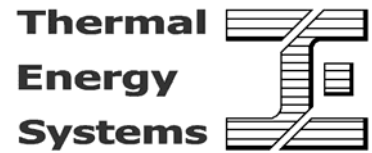


# Co-generation Thermodynamics Revisited

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## Abstract

*Whilst most engineers understand that higher HP steam conditions result in a more efficient power station, a wide range of different conditions have been selected around the sugar industry's export co-generation stations. This paper examines the thermodynamics of such a station and provides some help in optimising it. It also offers some practical advice for the designers of new stations.*

## Untersuchung zur Thermodynamik der Kraft-Wärme-Kopplung

*Während die meisten Ingenieure sich der Tatsache bewusst sind, dass höhere Hochdruckdampfbedingungen zu einem effizienteren Kraftwerk führen, ist bei den Export-Kraft-Wärme-Kopplungsanlagen der Zuckerindustrie ein breites Spektrum von unterschiedlichen Bedingungen gewählt worden. Dieses Referat untersucht die Thermodynamik eines solchen Werks und liefert hilfreiche Hinweise zu ihrer Optimierung. Es gibt zudem einige praktische Ratschläge für die Konstrukteure neuer Werke.*

## Se reexamina el tema de la termodinámica de la cogeneración

*A pesar de que la mayoría de los ingenieros entienden que condiciones de vapor a alta presión (HP) dan lugar a una central eléctrica más eficiente, muchas centrales de cogeneración para la exportación, en la industria azucarera, utilizan aún una gran variedad de condiciones. En este artículo se examina la termodinámica en dichas centrales y se sugieren métodos para su optimización. Se ofrecen también algunos consejos prácticos para los diseñadores de centrales nuevas.*

## Introduction

The sugar industry has been co-generating for many decades and is now moving towards substantially improved power stations so that it can export surplus energy when prices are right. However, too many engineers in the industry do not understand even the key aspects of co-generation and find it difficult to specify, let alone design, an export power station.

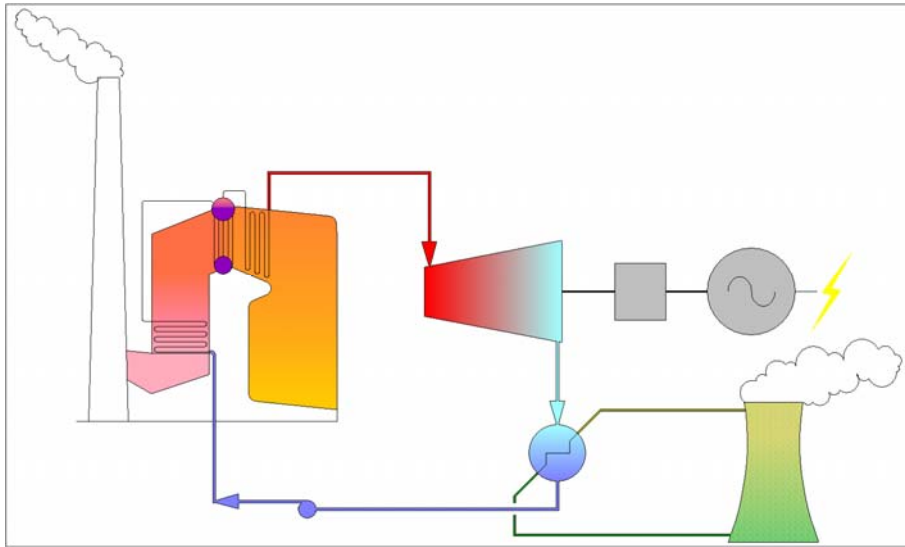
Once the fundamental principles are understood, it becomes much easier to apply those principles and achieve the optimum solution. Having reminded the reader of the advantages of co-generation, this paper provides a refresher course of the thermodynamics before showing how to apply that knowledge to optimise the conditions within the cycle. Finally it examines the implications of selecting high steam conditions so that initial selections may be changed if necessary to avoid major cost increase for little gain.

## Co-generation

It is perhaps better to start by discussing what co-generation is not in order to better understand what it is.

Conventional power generation is not a particularly efficient process because of the fundamental physics involved. Figure 1 [over] shows a typical station. The efficiency of the station depends on the sum of all the losses in the cycle.

A typical boiler might be 85% efficient when fossil fuel fired but only say 67 % when firing a fibrous fuel like bagasse because boiler efficiency depends largely on fuel moisture. About  $\frac{1}{3}$ <sup>rd</sup> of the bagasse energy is lost, mainly up the stack. The turbine is relatively efficient in converting the energy in the steam to mechanical energy. It is usually about 85 % efficient for the machines used in the sugar industry. Taking into account small gearbox and generator losses the overall efficiency of the turbo-generator set, as a unit, is about 80%.



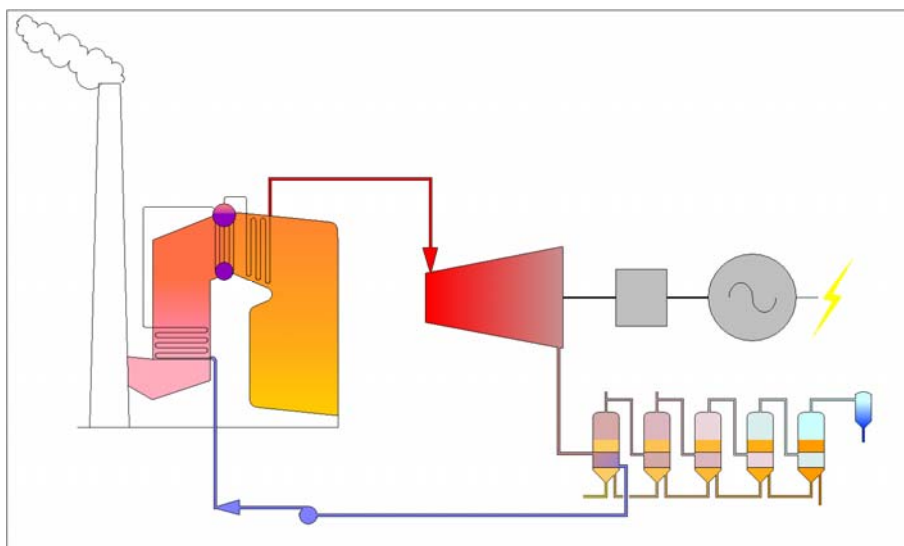
**Figure 1 : Conventional Power Generation**

The problem is that most of the energy used in a boiler is required to turn water into steam, not in raising the steam pressure and temperature. Most of that energy cannot be used by the turbine and must be dumped in the condenser and hence to atmosphere through a cooling tower.

Therefore, what appears to be a somewhat efficient cycle (67 % of 80 % = 53.6 %) is actually only maybe 22 % efficient because it cannot make use of most of the latent heat in the steam. [These figures are very broad but give an indication of the nature of the problem.]

If it were possible to use the low-grade heat rejected in the condenser then the whole system becomes more efficient. Co-generation does that.

Co-generation, as the name implies, couples a user of low-grade heat to the power station. Many of these users are industrial but there have been examples of district heating schemes [Battersea in London comes to mind] using the latent heat. The user is referred to as the 'host'.



**Figure 2 : Co-generation**

Many co-generation systems are purely in-house arrangements and most sugar factories are prime examples. The systems are designed so that the electricity generated supplies all of the host factory's needs and the low-pressure steam from the turbine supplies all of its heating needs.

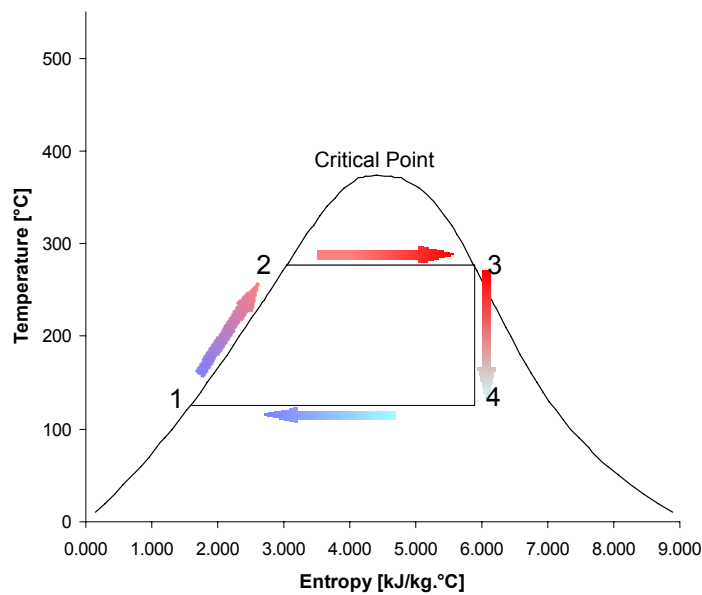
Because a co-generation station does make use of the latent heat, the process plant becoming the condenser, overall efficiency improves dramatically. The extent to which a station uses the latent heat is called the utilisation factor. In the example cited above the utilisation factor is about 33% so the overall station efficiency does become 53.6 % [say 50% in practice] which is more than double the efficiency of the equivalent conventional station.

Note that co-generation is not the use of a waste material or by-product such as bagasse as fuel for the station. A beet factory firing fossil fuel is as much a co-generation site as a cane factory burning bagasse. Equally, that same cane factory burning surplus bagasse during off-crop in full condensing mode is not co-generating. [In fact, because it is burning a fibrous fuel and because it is impossible to justify some of the expensive cycle enhancements found on large utility stations, it is substantially less efficient than a typical conventional station; only its 'free' fuel and the idle cost are in its favour.]

In summary then, most of the losses in a co-generation cycle are from the boiler. However, as it can be readily demonstrated that about 90% of those losses are stack losses in a bagasse fired boiler and hence proportional to the gas exit temperature, we know how to optimise those. This paper examines the rest of the cycle.

### The Thermodynamic Cycle

Steam turbine power generation is a Rankine Cycle. Plotting the theoretical process on a temperature/entropy [T/s] diagram as shown in Fig. 3 emphasises why it is called a cycle :



**Figure 3 : Temperature/Entropy Diagram**

The curve is the saturation curve for water so the peak of the curve is the critical point with liquid water to the left of the curve and water vapour [steam] to the right. Starting with the boiler feedwater, the water is heated to boiling point (1 to 2) then evaporated (2 to 3) before being expanded in the turbine which exhausts a wet LP vapour (3 to 4) that is then condensed (4 to 1) to make feedwater again.

Newton's First and Second laws tell us that, for reversible processes, the work done in the cycle is equal to the heat flux :

$$\oint \delta W = \oint \delta Q$$

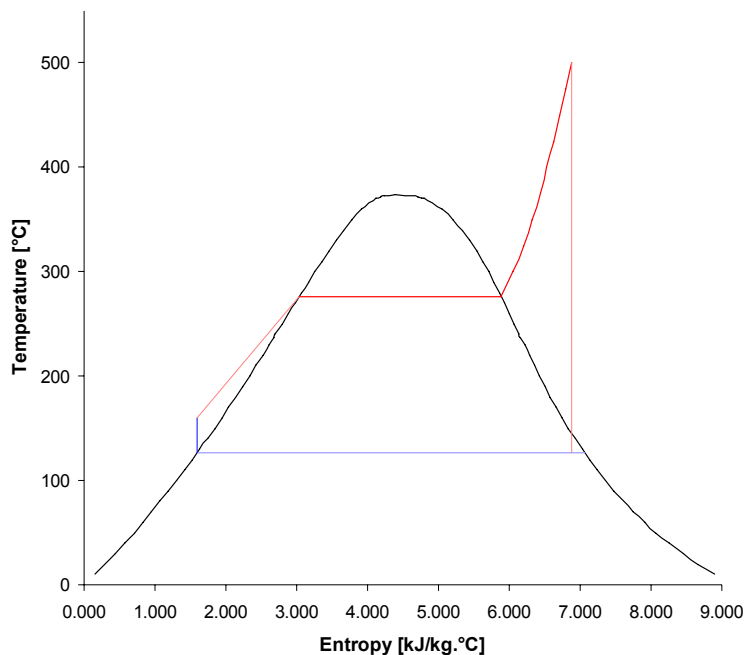
and that the heat flux is related to the temperature and the change in entropy :

$$\oint \delta Q = \oint T ds$$

Resolving the two equations, the work done in the cycle is related to the temperature and the change in entropy. As an integral is the area under a curve, the work done during theoretical steam turbine power generation – because it is reversible – is represented by the area bounded by the lines on the diagram. It follows, therefore, that the higher the steam temperature and the lower the exhaust temperature, the more work that can be obtained from the cycle. The steam temperature is the saturation temperature at the boiler pressure of course so the higher the boiler pressure, the more work is obtained from the cycle. Similarly, the exhaust temperature is a function of the condenser pressure so the lower the condenser pressure, the more work is obtained from the cycle.

Note that while a T/s diagram represents the work obtained from a cycle it does not represent the utilisation factor of a power station.

The T/s diagram [Fig. 4] also shows us why superheat is used in most cycles as the area bounded is substantially increased :



**Figure 4 : Cycle with Superheat and BFW Pump**

In this particular example the area bounded is almost 50% more than the previous example without superheat.

*[In modern power stations a vapour reheater is incorporated in the boiler so that as the vapour approaches saturation it is again superheated and expanded to obtain more work. That gives a second saw-tooth curve on the T/s diagram, again increasing the area bounded and showing that more work still is obtained from the cycle. However, the system is expensive and not very practical for small co-generation stations and we are not aware of any in the sugar industry.]*

We have seen that the vertical line on the right represents the isentropic [adiabatic] expansion of the steam in the turbine as it does work. However, the feedwater pump is also putting work into the cycle and that means that the water temperature increases as the pressure increases. Theoretically, it too is an isentropic process so is represented by a vertical line on the T/s diagram.

### **Practicalities**

All of the above is theoretical and does not, of course, correctly represent what happens in real life. The two most important issues are that neither the feedwater pump nor the turbine are perfectly efficient and that there are practical limits for conditions at various parts of the cycle.

### Mechanical Efficiencies

It is the turbine efficiency that is the more important in the mechanical efficiency issue. The reasons are best discussed initially using a Mollier Chart which is an enthalpy/entropy [H/s] diagram. An isentropic expansion still shows as a vertical line on a Mollier Chart from the HP to the LP condition. However, the less than perfect efficiency means that only a percentage of the enthalpy is actually converted to work. That means, in turn, that for any particular exhaust pressure one must track to the right and back up the pressure curve on the chart to intersect the actual enthalpy: the entropy has increased. The mathematics is simple: if the turbine barrel efficiency is say 85% then 85% of the isentropic enthalpy change is actually achieved.

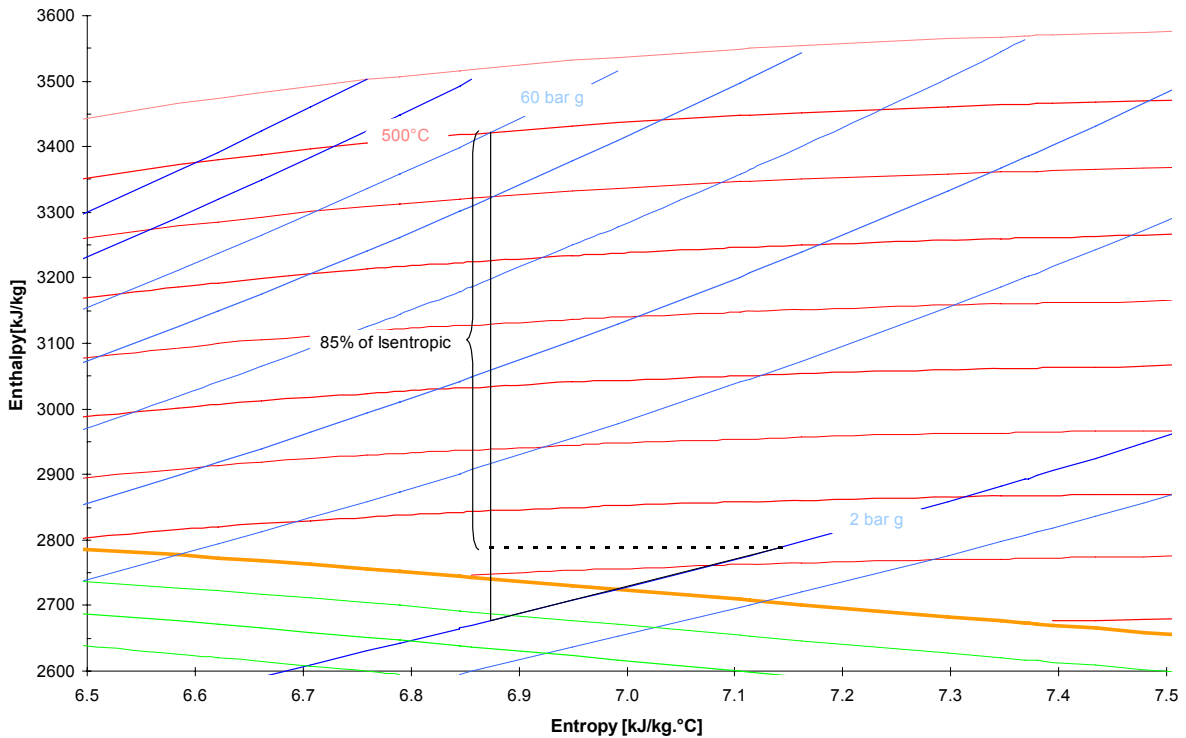


Figure 5 : Mollier Chart

Returning to the T/s diagram then, it follows that the practical expansion line shows increasing rather than fixed entropy. It must be to the right of the isentropic expansion line. Similarly, the practical compression line of the pump must be to the right of the isentropic compression line because entropy will also have increased in that case. This can be seen in Fig. 6.

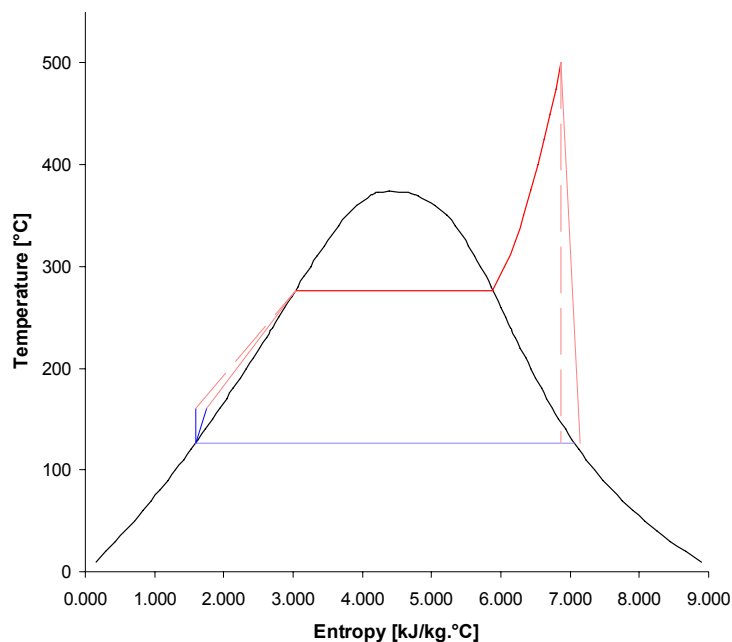


Figure 6 : A Practical Cycle [85%  $\eta_t$ ]

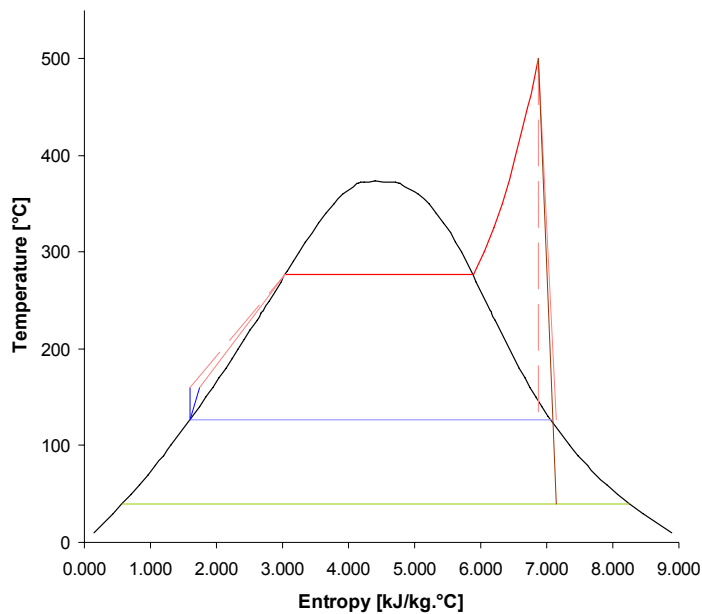
[The practical expansion and compression lines in the figure and the subsequent figures are drawn as straight lines although in real terms they are curves as the efficiency loss is compound. It is the net result which is of interest in this paper. Similarly, the evaporation and condensation lines are drawn as straight, horizontal lines although in practice there are losses to be considered here too.]

Note that the real processes are not isentropic and therefore not completely reversible and that the theoretical relationship between the area bounded and the work done is therefore no longer relevant. For example, the small increase in the area bounded resulting from a sloping expansion line cannot be considered as an increase in work output.

### Practical Limits

The most important practical limit in co-generation is the need to have a suitable temperature to drive the process that is the effective condenser. Most engineers in the sugar industry will discuss exhaust steam in terms of its pressure but the correct approach would be to discuss its saturation temperature. Exhaust is only used for heating, not for further expansion.

A typical sugar factory might have an exhaust condition of 125 °C saturation temperature : compare that with a condensing turbine's final temperature of perhaps 40 °C as shown in Fig 7.



**Figure 7 : Exhaust vs Condensing**

The much lower temperature of the LP/LT part of a condensing turbine means that the work obtained is much greater – about 60% greater in the example above. [Remember that the T/s diagram only shows turbine work output, not the utilisation factor].

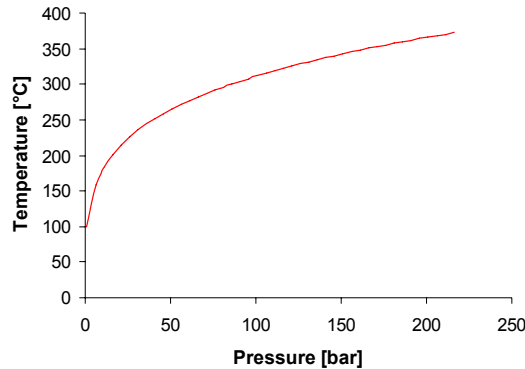
There is a second aspect to the practicality of the exhaust point. The exhaust steam should, ideally, be slightly superheated to allow for line losses so that it is just on saturation at the first effect and any other exhaust users. Exhaust quality is also an issue within the turbine as it leads to erosion of the blades. On the T/s diagram that means that the exhaust point should be slightly above the saturation temperature and hence to the right of the vapour saturation curve as the area between the two sides of the curve implies a dryness fraction of less than 1.0.

The third aspect to practicality is also at the LP/LT end of the cycle : in most pass-out turbines [and that is what is usually best if power export is required beyond the end of crop] it is necessary to keep the back end cool by passing some steam through to the condenser. A typical figure would be 15 to 20% of the full flow steam rate.

Note that dryness is less of an issue with a condensing turbine, provided that one stays within the limits set by the turbine manufacturer : usually somewhere between 0.95 and 0.90 dryness fraction.

The other practical limit is at the HP/HT end of the cycle. Not only do higher pressures mean thicker materials of construction but the higher temperatures mean that the strength of the materials deteriorates, forcing the selection of more exotic [and therefore more expensive] materials. There are, in fact, two cases to consider when discussing this issue : the saturation condition which affects the entire pressure part system and the superheated condition which only affects the superheater, steam pipework and turbine.

It is useful to remind ourselves of the temperature/pressure relationship of saturated steam :



**Figure 8 : Saturated Temperature of Steam**

As the pressure rises the temperature becomes more asymptotic so that it takes substantial pressure increases to make relatively modest gains in temperature. As we are, in any case, moving into the narrowing part of the T/s diagram the law of diminishing returns starts to apply in earnest.

In the end though, the issue is only a matter of economics : can the extra mass and/or more expensive materials of construction of the pressure parts be justified in the light of the extra work gained from the cycle? This and the other practicalities are discussed in the next section.

### **Applying the Lessons**

Clearly it is not possible to play with the saturation curve on the T/s diagram. Therefore it is only the evaporation and condensation temperatures plus the amount of superheat which must be decided when developing a co-generation scheme.

Most industry engineers will start discussing an export co-generation project [the usual circumstance in which high efficiency cycles come into play] by trying to select the HP/HT end of the cycle. However, the correct place to start is the exhaust point and the exhaust condensing temperature. From there, the next step is to discuss the upper end of the cycle but, as we shall see, in close co-operation with the turbine supplier.

#### *Exhaust Steam Temperature*

The exhaust steam saturation temperature – the condensing temperature – dictates the heat transfer surface area of the first effect of the evaporator and any other users. As one of the first requirements of export co-generation is to have an efficient factory, there are likely to be few such ‘other users’ of exhaust. The key, as with any optimisation, is the approach to the operating temperature of the first effect.

As the condensing temperature is reduced to improve the work output from the cycle, the cost of the evaporator goes up as more heat transfer surface area is required for the same duty. Whether the typical sugar factory approach of 10 °C is optimal needs to be questioned.

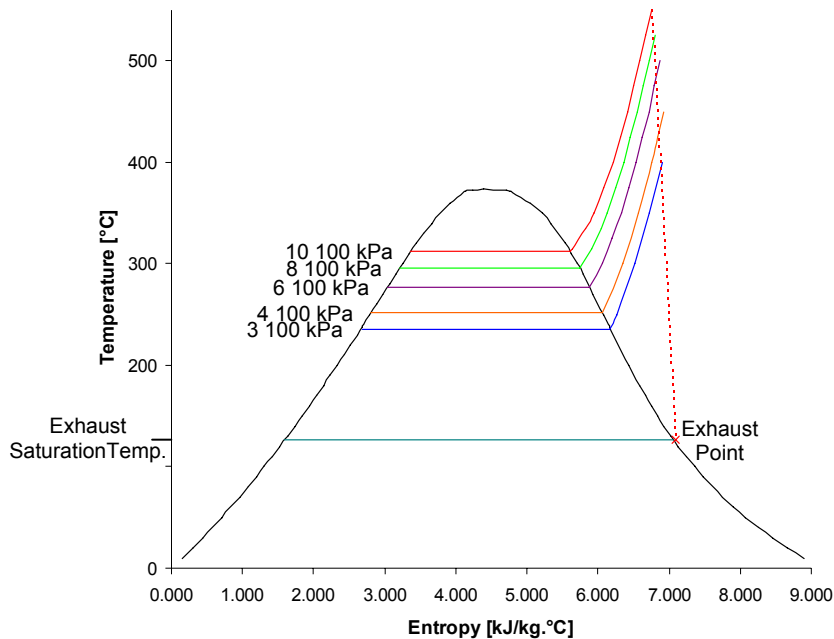
With a typical cycle having a 6 000 kPa HP condition, reducing the condensing temperature from 124 to 122 °C increases the work output by 1%. If the electrical output is say 30 MW then that 300 kW might be worth \$21 every hour.

Once the condensing temperature has been agreed then the preferred exhaust point can be determined. The actual exhaust point will depend on the turbine selected but a preferred point is required before starting a dialogue with turbine suppliers. A typical amount of superheat for the exhaust point might be 5 °C so the exhaust point is 5 °C above the selected condensing temperature, along the constant pressure curve to the right of the vapour saturation curve.

Working backwards from the exhaust point, the inlet condition can be derived if the barrel efficiency of the turbine is known. As a first approximation, assume an efficiency of 85% and establish the expansion line. Now it is possible to consider the high end of the cycle.

*HP/HT Steam Condition*

For any selected boiler operating pressure there is a unique curve of evaporation and superheating. There is no theoretical limit to the amount of superheat if the boiler can create it.



**Figure 9 : Optimum Superheat**

What the T/s diagram in Fig. 9 shows, however, is that there is an optimal amount of superheat for any boiler operating pressure. For the conditions selected for the diagram above [126 °C exhaust steam saturation temperature, 2 °C of superheat, 85% barrel efficiency] the optimal HP/HT steam condition is :

Steam Pressure	3 100	4 100	6 100	8 100	10 100	kPa a
Steam Temperature	388	423	475	513	545	°C

Any lower steam temperature for the pressure selected means either a lower efficiency turbine to achieve the same exhaust point or less superheat – more probably a dryness fraction of less than 1.0 – in the exhaust steam. Any higher HP/HT steam temperature for the pressure selected means either a higher efficiency turbine [if available] or more superheat in the exhaust steam.

The optimum conditions are quite dependent on the barrel efficiency of the turbine :

Steam Pressure	3 100	4 100	6 100	8 100	10 100	kPa a
Steam Temperature [80% $\eta_t$ ]	370	402	450	486	515	°C
Steam Temperature [85% $\eta_t$ ]	388	423	475	513	545	°C
Steam Temperature [90% $\eta_t$ ]	407	445	502	545	579	°C

All of the above assumes that the turbine efficiency is not affected by the inlet steam conditions but that is not true. One of the main influences on barrel efficiency [as distinct from the losses and other turbine inefficiencies] is the volume flow rate of the steam and hence the specific volume of the inlet steam. That is because lower volume flows mean shorter blades and the blade 'end effects' [both the root effect and the tip effect] become more significant.

A typical efficiency calculation for a turbine might look like this :

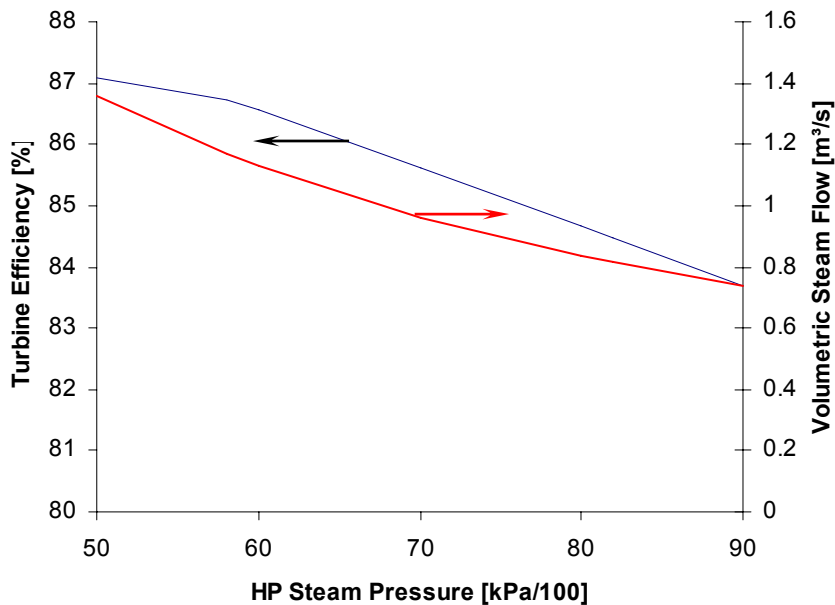
$$\eta_t = \eta_n - \frac{k}{CT^{1.11}}$$

Where  $\eta_n$  is the nominal efficiency of the machine  
 $k$  is a constant for the machine

The third factor,  $CT$ , is a function of the steam conditions :

$$CT = \frac{\text{mass flow}}{\text{pressure}} \times \sqrt{\frac{\text{pressure}}{\text{specific volume}}}$$

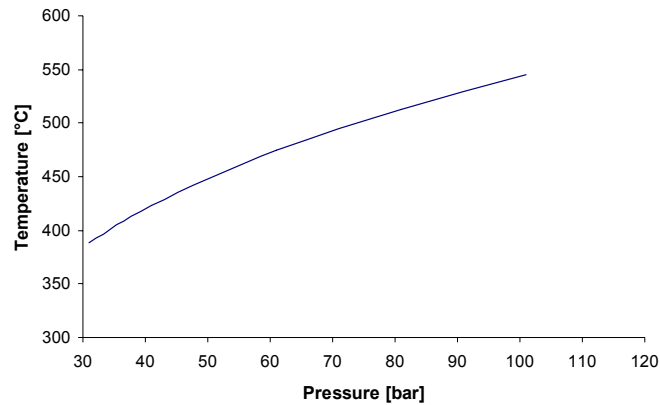
Plotting all this out shows a reasonable correlation between efficiency and volume flow :



The reducing efficiency at higher pressure means that, in practice, the increasing optimum temperatures are not quite as high as indicated by assuming a constant barrel efficiency.

In theory, having selected the exhaust point and the HP/HT steam conditions it is now possible to approach potential turbine suppliers. However, there is first a need to understand the implications of the HP/HT conditions with respect to both the turbine and the boiler in case it makes sense to modify the selections.

Plotting the relationship between HP/HT steam temperature and pressure in the first table above indicates a negative 2<sup>nd</sup> derivative leading to a point of inflection at some higher pressure but that would be certainly outside of the normal pressure range used in sugar industry co-generation stations. Therefore both the temperature and the pressure implications need to be fully understood.



**Figure 10 : Optimum Steam Temperature**

There is one other aspect to be considered. If one returns to the Mollier diagram [Figure 5], it can be seen that as the pressure rises for any one temperature, the total enthalpy falls. In other words, higher steam pressures require less fuel burn at any particular steam temperature. It therefore pays to fine tune the cycle with increasing pressure rather than with reducing temperature. This can be seen in the following two tables :

all at 480 °C ex boiler				
Steam Bar a	Bagasse kg/h	Power kW	S/H °C	Ratio kg/kW
62	49,297	16,234	12	3.037
63	49,276	16,286	11	3.026
64	49,254	16,337	10	3.015
65	49,233	16,387	8	3.004
66	49,211	16,435	7	2.994
67	49,190	16,483	5	2.984
68	49,168	16,529	4	2.975
69	49,147	16,575	3	2.965

all at 62 bar a ex boiler				
Steam °C	Bagasse kg/h	Power kW	S/H °C	Ratio kg/kW
480	49,297	16,234	12	3.037
478	49,216	16,185	11	3.041
476	49,135	16,135	10	3.045
474	49,054	16,086	8	3.050
472	48,972	16,036	7	3.054
470	48,891	15,987	6	3.058
468	48,810	15,938	4	3.063
466	48,728	15,889	3	3.067

The base HP/HT steam condition in each case is 62 bar a and 480 °C which, for the circumstances assumed, gives 12 °C of superheat in the exhaust. If the amount of superheat is reduced by increasing the steam pressure then the bagasse requirement falls while the power output increases so the specific bagasse consumption falls. However, if the amount of superheat is reduced by reducing the steam temperature then the bagasse requirement falls but the power output falls marginally faster and the specific bagasse consumption actually increases slightly.

*Implications for Boiler*

The high pressure implications for a boiler's pressure parts are relatively straight forward but it is easy to overlook them sometimes, particularly if the engineering of the boiler is undertaken in isolation of the thermodynamics and decisions on cycle conditions. In addition, preliminary thinking taking pressure alone into consideration can be upset once the temperature has also been considered.

Drum thickness is calculated from first principles but flanges, piping and tubing, for instance, come in standard sizes and thicknesses so there are step changes in capabilities not dissimilar to the standard frame size issue with mechanical equipment.

As an example, there may be little difference in the cost of the pressure parts for a boiler designed for 4 000 kPa compared to a 3 000 kPa one of the same capacity but a substantial difference for a 4 200 kPa one because the manifolds all go up one schedule in thickness and the boiler flanges, valves and mountings have to change from a Class 300 rating to a Class 600 rating.

We all tend to think of boiler pressures in standard steps, often linked to old imperial standards : 2 100 kPa [~300 psig], 3 100 kPa [~450 psig], 4 100 kPa [~600 psig] and so on. In fact that has been the case in this paper. However, it is more important to think in terms of the cost steps imposed by readily available materials than it is to think in terms of such historical pressure steps. Remember too that it is not the operating pressure that counts, it is the highest pressure in the system, usually many hundreds of kilopascals above the operating pressure because of the pressure drop across the superheater(s), the static head, operating margins and allowances for safety valve settings. In addition, do not be misled into thinking that standard flange ratings equate to those steps : those ratings must be modified in accordance with the temperature and material selections.

Pressure part components must be designed for the maximum mean wall temperature that the components are exposed to. For boiler tubes exposed to furnace radiation this is in the order of 50°C higher than the saturation temperature at the highest steam drum safety valve setting. For drums and manifolds which are heated by hot gasses, including the convection bank tubes, the design temperature may be +25°C higher than the saturation temperature at design pressure. For superheater tubing the design temperatures are even higher to take into account the following:

- Lower internal heat transfer coefficients
- Variations in steam flow during sudden load changes
- Variations in steam flow due to poor flow distribution
- Variations in heat fluxes resulting from uneven gas flows in the boiler

The selection of materials for the boiler pressure parts (with the exception of the superheater/s) are therefore primarily based upon the selected operating pressure. The materials selection for the superheater/s and its components is more complicated and is influenced by other factors such as creep and high temperature corrosion resistance.

Following these principles and ignoring any potential corrosion issues from auxiliary fuels, the following materials requirements become evident in the following table :

Steam Conditions		Boiler Components	Superheater
3,100 kPa	400°C	C or C-Mn steel Class 300	C & Low alloy steel Class 600
4,100 kPa	440°C	C or C-Mn steel Class 600	Low alloy steel Class 600
6,100 kPa	480°C	C-Mn steel Class 600	Low alloy steel Class 900
8,100 kPa	520°C	C-Mn & low alloy steel Class 900	High chrome alloys Class 1500
10,100 kPa	550°C	C-Mn & low alloy steel Class 1500	Stainless steel and high chrome alloys Class 1500 to Class 2500

C : Carbon

C-Mn : Carbon Manganese

Low Alloy : Carbon Molybdenum, Chrome Molybdenum, Manganese-Chrome-Molybdenum-Vanadium

High chrome Alloy : 2 ¼ Chrome, 9 Chrome and 12 Chrome alloys

The other major implication for higher HP/HT conditions is the quality of feedwater, boiler water and steam. Above about 60 bar the water quality requirements become far more stringent and demineralisation and volatile treatments becoming a necessity. The control of boiler water quality becomes critical to minimise steam impurities and prevent deposits forming in the superheater(s) control devices and turbine blades.

### *Implications for Turbine*

Similar implications exist for the turbine although the turbine inlet conditions are slightly below the boiler outlet conditions.

These limitations are, of course, a function of material selection as well as material thickness in the inlet sections and they vary from manufacturer to manufacturer. For some manufacturers the first temperature break point is at around 480 °C, whereas for others it seems to be around 510 °C.

Overall, the material limitations on turbine inlet sections follow the same engineering issues as for steam piping, i.e. the temperature limitations of a given material quality of a specified thickness is basically a function of the operating pressure. However, in addition to the pressure/temperature limitations on the materials from a creep life and softening point of view, additional limitations may be set by the physical design of the inlet valves and distribution pipes and wheel chamber design limitations come into play – above certain pressures and/or temperatures, a manufacturer's design might require an inner casing to protect the outer casing from excessive stresses. Higher temperatures may also require a change in rotor material – and in particular temperatures above 525 or 530 °C may require relatively advanced alloys to maintain the integrity of the equipment.

### *HP/HT Optimisation*

In the light of the above it may well be that the original thinking for the HP/HT steam conditions will be modified as the extra work gained by higher conditions might not justify the higher cost of boiler and turbine. How this affects the thinking will depend on the particular project, its economics and the availability of funds.

### *Condenser Temperature*

There is one other aspect of the cycle which has to be considered : the condenser temperature.

Many so-called co-generation stations have to operate as conventional condensing stations out of crop in order to be able to offer 'firm' power [power available for typically 11 months a year and therefore allowing the utility to avoid the capital cost of that capacity]. One option would be to have exhaust turbines for co-generation and condensing turbines for generation but the usual approach is to have pass-out machines which can do both. [Twin shaft machines where the condensing barrel can be uncoupled is another, probably better, solution but outside the scope of this paper.]

The need to keep the backend of such a machine cool while 'co-generating' has already been discussed. All that need be decided is the condenser temperature and the answer is simple : as low as possible. In other words the cooling tower and the condenser itself should each be designed for low approach temperatures so that the operating condition within the condenser is as low as possible. Two approach temperatures of 5 °C each will give a condenser temperature of 10 °C above the dew point but if each were only 4 °C the extra work obtained from the 2 °C lower condenser temperature might justify the extra capital costs. We have already seen that a 2 °C reduction in exhaust temperature gives about 1% improvement in work output : there will be a slightly better improvement at the condensing condition because the two saturation curves are further apart.

### *Overall Optimisation*

Having made the preliminary decisions, it is now possible to approach potential turbine manufacturers and start to optimise the cycle. From a thermodynamic point of view, the barrel efficiency has to be the starting point. It may even be of benefit to pay more for the turbine if more work is obtained. Remember too that higher barrel efficiencies will allow lower steam inlet pressures which might bring down the cost of the boiler.

## **Conclusions**

T/s diagrams are a useful means of visualising the steam power generation cycle and the amount of work obtainable from it. They are less useful for visualising the overall benefits of co-generation because it is the utilisation factor of the enthalpy [and the latent heat of evaporation in particular] that is the key to that. However, the principles are still applicable and can be used in optimising a co-generation project.

The starting point for the design of a co-generation cycle is the selection of the exhaust point and the lower the exhaust temperature, the more efficient the system. It probably pays to invest in first effect evaporator heat transfer surface area to achieve this, whether starting with a green field project or converting an existing factory to co-generation. Once that decision is taken it is possible to work backwards to the HP/HT steam conditions. To do that, the turbine barrel efficiency must be known.

For any combination of exhaust point and barrel efficiency there is an optimum steam temperature for the selected inlet steam pressure. However, as it is the steam temperature rather than the pressure which more strongly dictates the changes in materials of construction for both turbine and boiler, it is easier to select the inlet temperature and hence derive the optimum pressure.

It quickly becomes apparent that the boiler engineer and the turbine supplier need to work closely together with the sugar factory management if a co-generation project is to be fully optimised and unless it is optimised it will not be as profitable as it might.

## **Acknowledgements**

I would like to thank my colleague in Capetown, Ben Misplon, for all of his assistance in the preparation of this paper. I would also like to thank Tomas Hertzell of Alstom for his assistance with the implications of high HP/HT conditions for turbines.