Cogeneration 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 HP and exhaust steam conditions have been selected around the sugar industry's export co-generation stations. This paper examines the thermodynamics of such a station and then sets out a method to optimise it.

Nature is, of course, never that simple. There also needs to be a discussion on how to optimise the thermodynamics when, as is often the case with our export stations, there are two entirely different sets of conditions: in-crop export cogeneration and out of crop generation. The paper also offers some practical advice for the conceptual design of new stations.

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 any surplus. 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.

Conventional power generation is inefficient because the low grade heat dumped to atmosphere is the latent heat: the majority of the energy in the steam. 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. Note that co-generation is not the use of a waste material or by-product such as bagasse as fuel. 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.

Once the fundamental principles are understood, it becomes much easier to apply those principles and achieve the optimum solution. This paper provides a refresher course of the thermodynamics before showing how to apply that knowledge to optimise the conditions within the cycle.

The Thermodynamic Cycle

Steam turbine power generation is a Rankine Cycle, best plotted on a temperature/entropy [T/s] diagram :

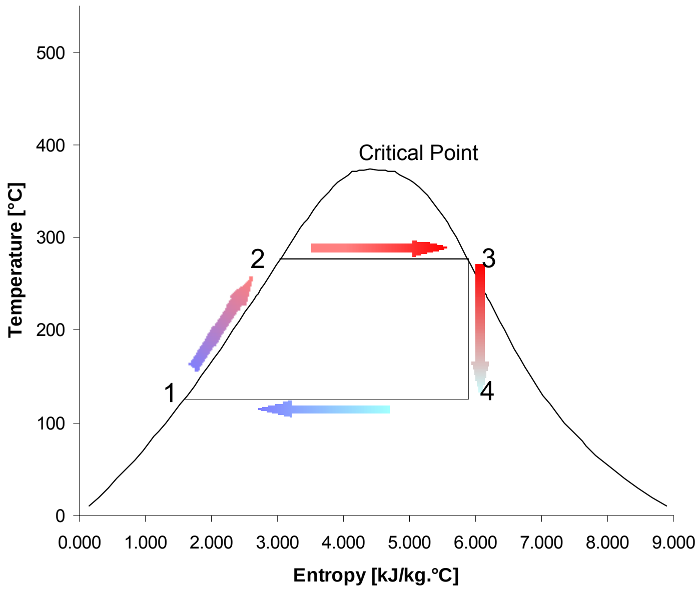

Fig. 1 : Temperature/Entropy Diagram

The First and Second laws of Thermodynamics tell us that, for reversible processes, the work done in the cycle is equal to the net 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 one temperature and the lower the other, the more work that can be obtained. As the upper temperature is the saturation temperature, higher boiler pressures give more work. Similarly, the lower the condenser pressure, the more work is obtained. Note that while a T/s diagram represents the work obtained it does not represent the utilisation factor of a power station. The T/s diagram also shows why superheat is used in most cycles:

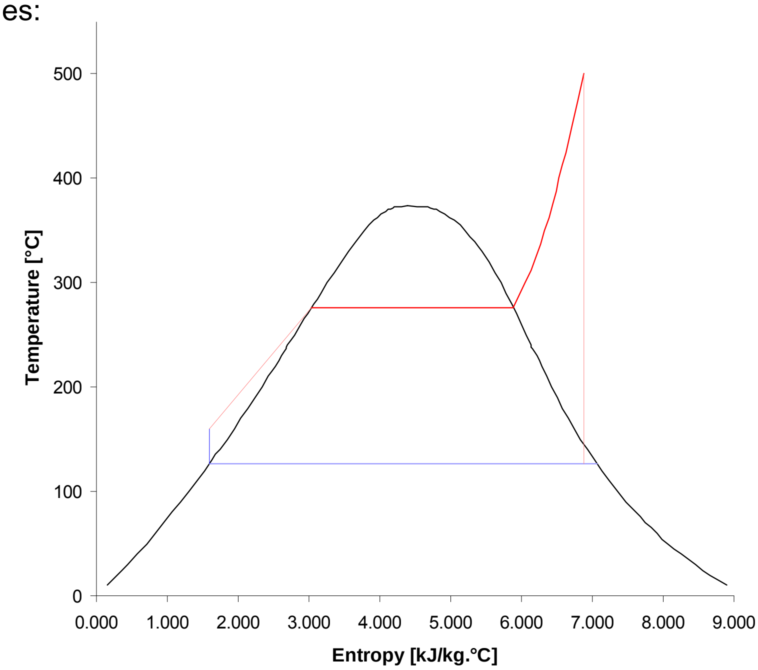

Fig. 2 : Cycle with Superheat and BFW Pump

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

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 not what happens in practice: neither the turbine nor the feedwater pump is perfectly efficient and there are practical limits for conditions at various parts of the cycle.

Mechanical Efficiencies

It is the turbine efficiency that is the most important issue. This is best discussed initially using a Mollier Chart, an enthalpy/entropy [H/s] diagram :

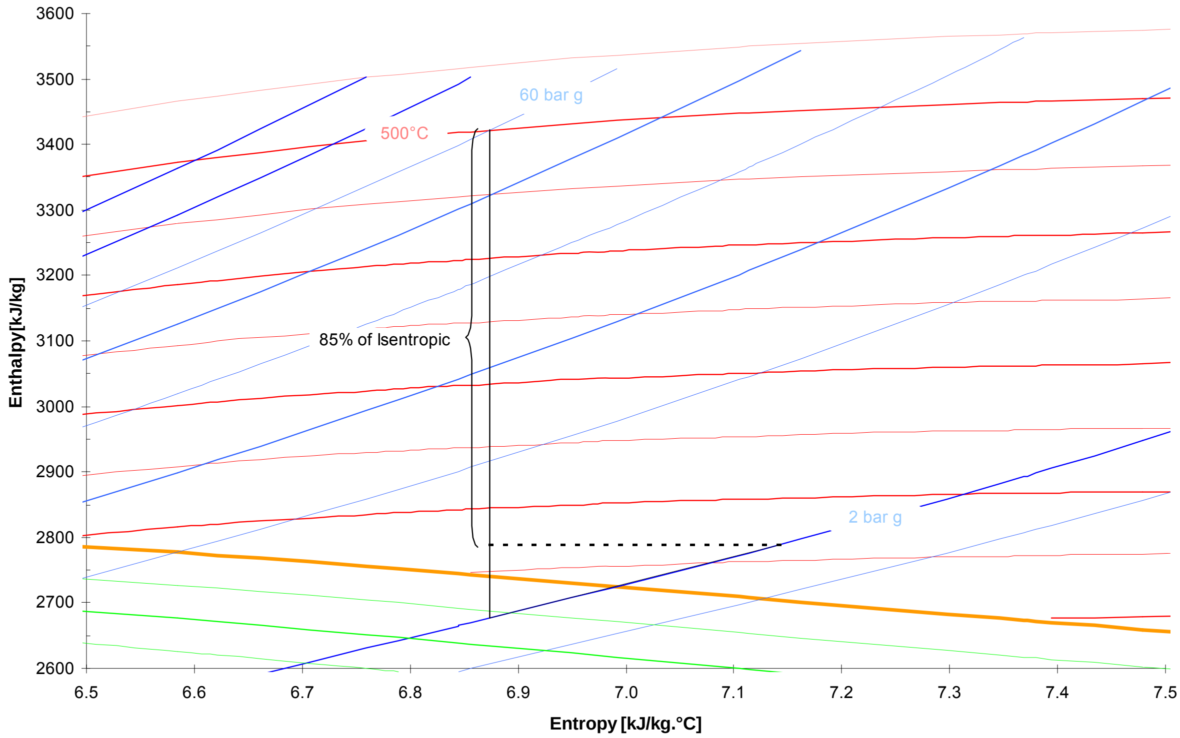

Fig. 3 : Mollier Chart

Isentropic expansion still shows as a vertical line from the HP to the LP condition. However, a less than perfect efficiency means that only some of the enthalpy is actually converted to work so, for any particular exhaust pressure, one must track back up the pressure curve to intersect the actual exhaust enthalpy: the entropy has increased.

Returning to the T/s diagram then, it follows that the practical expansion line shows increasing entropy, i.e. 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. This can be seen in Fig. 4 :

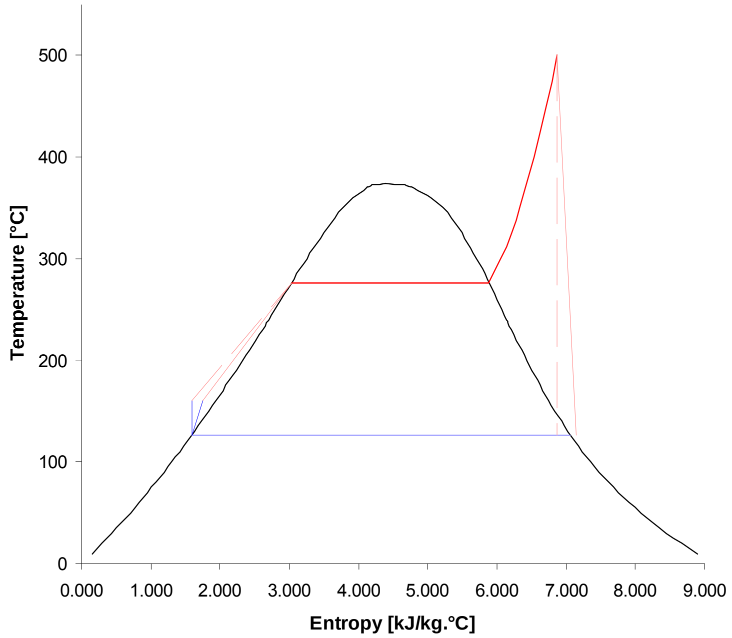

Fig. 4 : A Practical Cycle [85% η_t]

Note that the real processes are not isentropic and therefore not completely reversible, so the theoretical relationship between the area bounded and the work done no longer applies: 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 for a suitable temperature to drive the process that is the effective condenser.

A typical sugar factory might have an exhaust condition of 125°C saturation temperature compared with a condensing turbine's final temperature of perhaps 40°C:

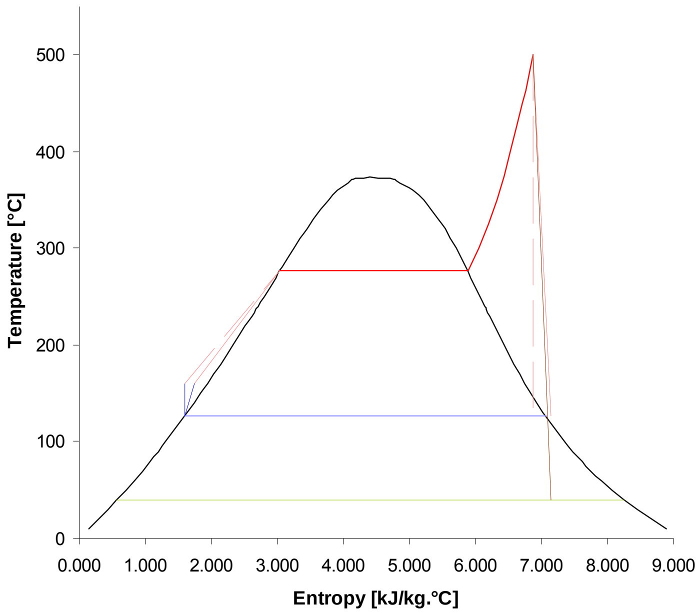

Fig. 5 : Exhaust vs Condensing

The reduced lower temperature means that the work obtained is much greater – about 60% greater in the example above – but 40°C cannot be used to heat the process. [Remember that the T/s diagram only shows turbine work output, not the utilisation factor].

The exhaust steam should, ideally, be slightly superheated to allow for line losses so that it is just on saturation when used. Exhaust quality is also an issue within the turbine as it leads to erosion of the blades. On the T/s diagram, the exhaust point should therefore be slightly above the saturation temperature and hence to the right of the vapour saturation curve.

The third aspect to practicality is also at the lower end of the cycle : in most pass-out turbines, it is necessary to keep the back end cool by passing some steam through to the condenser.

The other practical limit is at the HP/HT end of the cycle. Not only do higher pressures mean thicker materials of construction but, as the strength of the materials deteriorates, more exotic materials are required.

In addition, the temperature/pressure relationship of saturated steam is not useful :

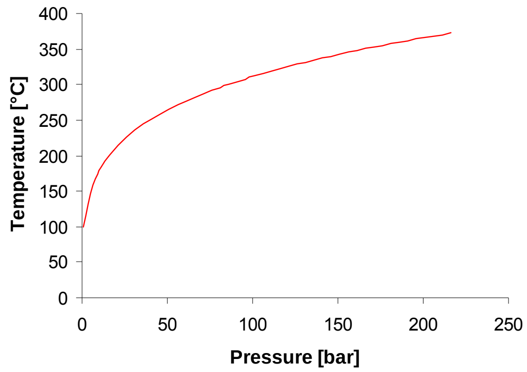

Fig. 6 : Saturated Temperature of Steam

It takes substantial pressure increases to make relatively modest gains in temperature and as we are 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 so only the evaporation and condensation temperatures plus the amount of superheat need 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 dictates the heat transfer surface area of the first effect. As one of the first requirements of export co-generation is to have an efficient factory, there should be no other users of exhaust.

The key is the approach to the operating temperature of the first effect. As the condensing temperature is reduced to improve the work output, 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 6000 kPa cycle, reducing the condensing temperature from 124 to 122°C increases the work output by 1%. If the electrical output is 30 MW, that 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. (desuperheater is normally used to guarantee saturated steam at the inlet of process equipment).

Working backwards from the exhaust point, the inlet condition can be derived if the barrel efficiency of the turbine is known.

HP/HT Steam Condition

For any selected boiler operating pressure there is a unique curve of evaporation and superheating.

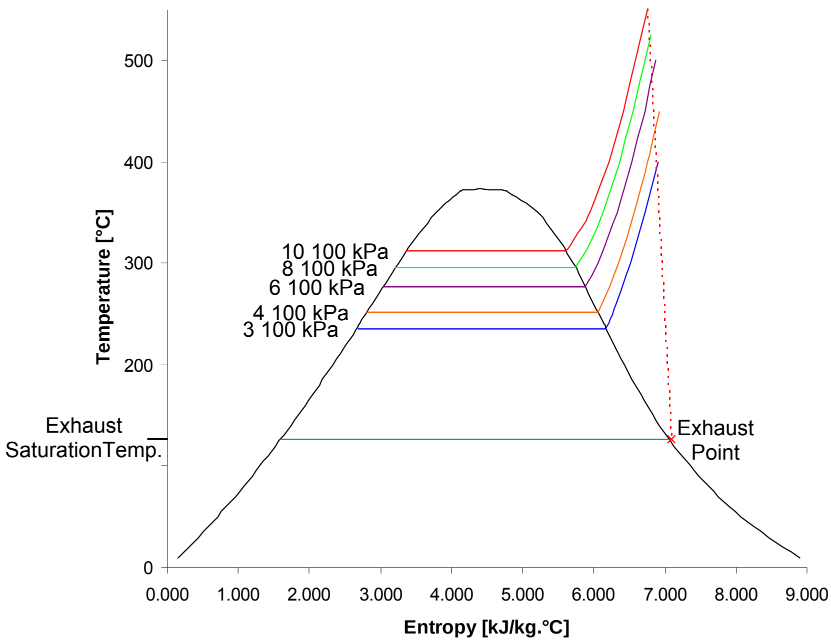

Fig. 7 : Optimum Superheat

The T/s diagram shows 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	3100	4100	6100	8100	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	3100	4100	6100	8100	10 100	kPa a
Temperature [80% η_t]	370	402	450	486	515	°C
Temperature [85% η_t]	388	423	475	513	545	°C
Temperature [90% η_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 one of the main influences on barrel efficiency [as distinct from the losses and other turbine inefficiencies] is the volume flow rate 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 η_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 :

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

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

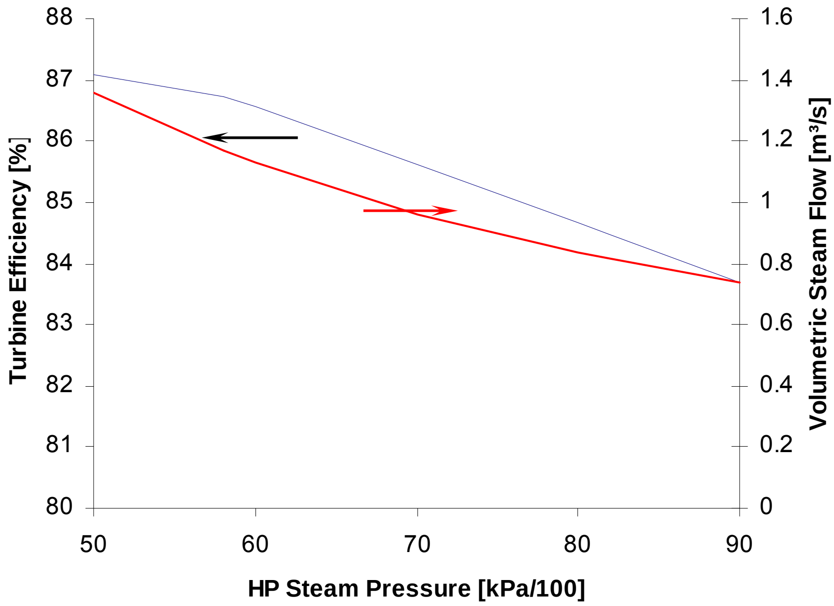

Fig. 8: Turbine Efficiency and Volumetric Flow vs Inlet Condition

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.

There is a need to understand the implications of the selected temperature and pressure conditions before contacting the turbine suppliers.

There is one other aspect to be considered. The Mollier diagram shows that, as the pressure rises for any one temperature, the total enthalpy falls: 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 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, 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 implications for a boiler are relatively straightforward provided that the boiler is not engineered in isolation and that pressure and temperature are considered simultaneously.

Drum thickness is calculated from first principles but flanges, piping and tubing come in standard ratings so there are step changes in capabilities not dissimilar to the standard frame size issue with mechanical equipment. There may, for instance, be little difference in the cost of the pressure parts for a boiler designed for 4000 kPa compared to a 3000 kPa one but a substantial difference for a 4200 kPa one because the manifolds all go up one schedule and fittings change from Class 300 to Class 600 rating.

Boiler pressures are often linked to old imperial standards: 2100 kPa [~300 psig], 3100 kPa [~450 psig], 4100 kPa [~600 psig] and so on. However, it is more important to think in terms of cost steps imposed by readily available materials than such historical pressure steps. Additionally, it is the highest pressure in the system, not the operating pressure, that counts. This is usually well 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, standard flange ratings do not equate to those steps: 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 typically 50°C higher than the saturation temperature at the highest steam drum safety valve setting. For parts which are heated by hot gases the design temperature may be 25°C higher than the saturation temperature at design pressure.

Superheater design temperatures are even higher to take into account:

- lower internal heat transfer coefficients
- steam flow variations during sudden load changes
- poor flow distribution
- heat flux variations resulting from uneven gas flows

Materials selection for most boiler pressure parts is therefore primarily operating pressure dependent. Materials selection for the superheater(s) 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, materials requirements are as shown over :

Steam Conditions		Boiler Components	Superheater
3100 kPa	400°C	C or C-Mn steel Cl. 300	C & Low alloy steel Cl. 600
4100 kPa	440°C	C or C-Mn steel Cl. 600	Low alloy steel Cl. 600
6100 kPa	480°C	C-Mn steel Cl. 600	Low alloy steel Cl. 900
8100 kPa	520°C	C-Mn & low alloy steel Cl. 900	High chrome alloys Cl. 1500
10 100 kPa	550°C	C-Mn & low alloy steel Cl. 1500	Stainless steel and high chrome alloys Cl. 1500 to 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 conditions is the quality of feedwater, boiler water and steam. Above about 60 bar, the water quality requirements become far more stringent with demineralisation and volatile treatments becoming necessary. 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. The limitations are 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. However, in addition to the 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 system, and wheel chamber design limitations come into play – above certain pressures and/or temperatures, the 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. One option is 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 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. The cooling tower and the condenser itself should therefore each be designed for low approach temperatures. 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.

Overall Optimisation

Having made the preliminary decisions, it is now possible to approach potential turbine manufacturers and start cycle optimisation. 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. Higher barrel efficiencies will also allow lower steam inlet pressures which might bring down the cost of the boiler.

What has also to be recognised is that the exercise as described only considers the situation during crop. A second optimisation is required for off-crop generation, made even more complicated if a refinery or an ethanol plant continues to operate out of crop.]

The principles described still apply but the exhaust point is replaced by a condenser point with the maximum wetness permitted by the turbine manufacturer and the lowest temperature achievable for the prevailing ambient conditions. This exercise will generate a different optimum HP steam condition from the crop one. The difficulty is in deciding which will prevail for design purposes as a boiler can only be optimised for one condition.

Economics will decide in the end and that will depend on the ratio of crop to off-crop length.

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 [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.