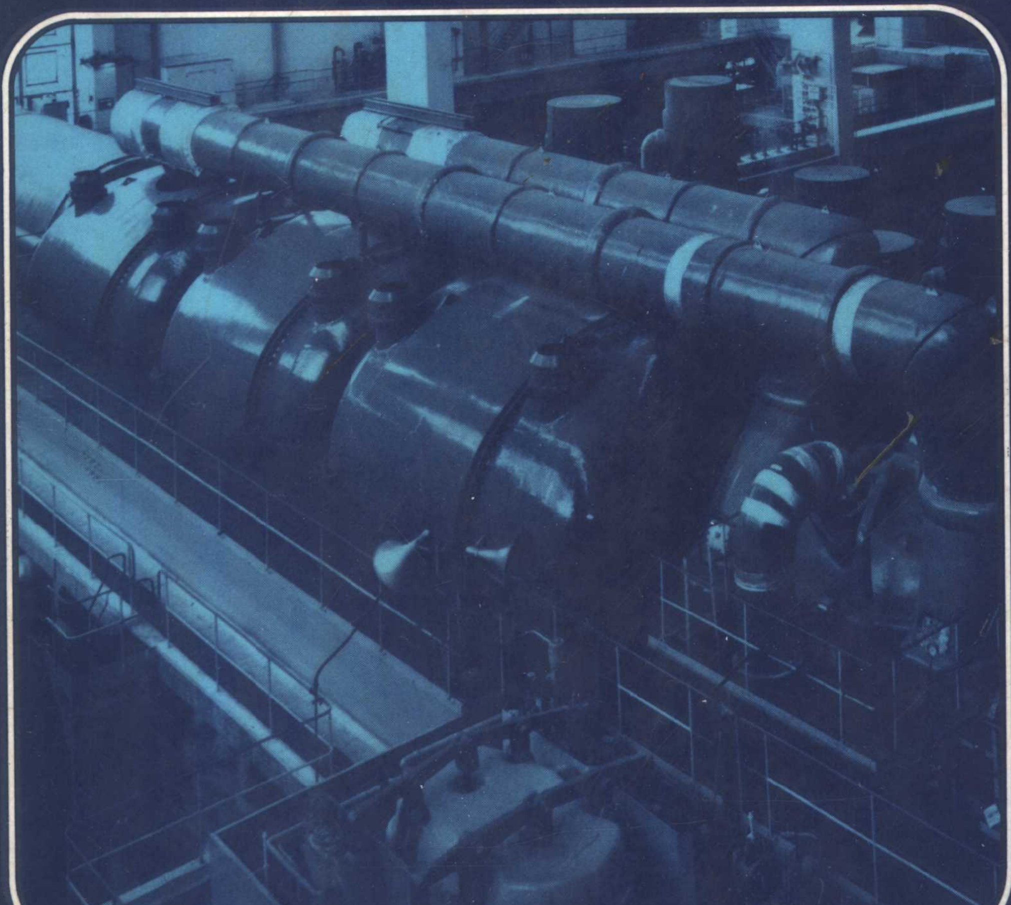
Power Plant Performance

A.B. Gill



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Power Plant Performance

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Preface

The considerable rise in electricity demand throughout the world has resulted in an enormous increase in the size of power plant installations. The capacity of a typical turbo-alternator thirty-five years ago was about 30 MW. Today, machines rated at 1300 MW are in service and the capacity of boiler and ancillary plant has kept pace with this rapid growth.

This expansion, coupled with the escalating cost of fuel, has imposed an increasingly urgent need to ensure that the plant is operated and maintained as near to optimum conditions as possible. At current UK fuel prices, if a large unit is operated at an efficiency one percentage point lower than design it will incur an additional fuel cost of over two million pounds per year. It is clear that it is of great importance that all power engineers should be aware of the causes of poor efficiency and the means to rectify such trends. That is the purpose of the book. Although written primarily for power engineers, others, such as marine engineers and engineering students, will find the material of interest.

As far as possible mathematics and thermodynamic theory have been kept to a minimum, the aim being to stress the practical application of the subject matter. Each chapter contains self-test questions, exercises and projects. The exercises are quite straightforward and should present little difficulty, whereas the projects are more demanding.

SI units have been standard within the CEGB for several years now so it is appropriate that they are used throughout the book. Also, the CEGB publications *Steam Tables in SI Units* and *Abridged Steam Tables in SI Units* have been used.

The basic model used is a British coal-fired station but, of course, much of the material is also applicable to other stations, whether nuclear or fired by oil or gas and no matter what the country. British practice may differ from that in other countries somewhat in particular aspects of design and, where this is so, reference is made to it.

The 'Performance' referred to in the title is thermal, as opposed to mechanical, performance. Thus, purely mechanical subjects such as vibration, turbine line-out and so on are not covered. However, pumps are so important in power station work that a chapter has been devoted to the subject.

The views expressed are those of the author and he is solely responsible for any errors or omissions.

Sincere thanks are due to the many people who have helped in the preparation of the work. They are too numerous to mention individually, but a special word of thanks is due to the following:

Mr J. Porteous, Director-General, Midlands Region, CEGB, for his kind permission to publish the book, to use illustrations from CEGB sources and for other facilities; Mr R. Mason and Mr T. Westcombe of the Efficiency Department at Drakelow Power Station for their help in reading the text and the preparation of many of the illustrations; Mr J.R. Jackson of the CEGB National Training Resources Unit for his valuable advice and encouragement; the Management and various staff at Drakelow Power Station for their help and co-operation; Mr F. Carlin, Midlands Region Scientific Services Department, Mr C. Clag and Mr P. Adams of the Hydraulic and Temperature Calibration Centre at Hams Hall for checking the material in Part 2, and last, but by no means least, the various organizations who have given permission to publish illustrations and other material.

Allan B. Gill

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Figures 7.1, 7.5 and 7.14 are from BS 1042, Part 1, 1964, and are reproduced by kind permission of the British Standards Institution, 2 Park Steet, London W1A 2BS, from whom complete copies of the Standard can be obtained.

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Part I

General Plant Considerations

1 Ideal steam cycles

The temperature–entropy (T–S) diagram

The basic diagram is shown in *Figure 1.1.* Note that the boiling water line

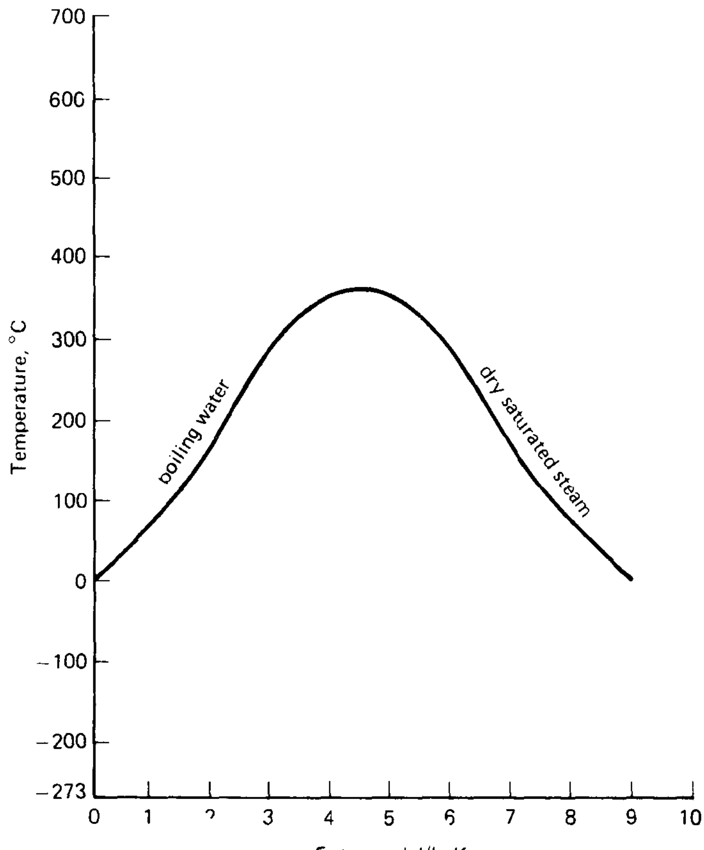

Figure 1.1. Temperature – entropy diagram SI Units

starts at 0°C and the temperature scale at absolute zero, i.e. (−)273.15°C. Also note that the entropy scale starts from zero.

Heat is the product of absolute temperature and change of entropy, and so on a T–S diagram it is represented by an area.

For example, consider the latent heat required to convert 1 kg of boiling water at 100 bar abs. into dry saturated steam.

The boiling temperature is 311.0°C = 584.111 K. The entropy of the boiling water is 3.3605 kJ/kgK and of the dry saturated steam 5.6198 kJ/kgK.

$$\begin{aligned}\text{So latent heat required} &= T(S_2 - S_1)\\ &= 583.111\,(5.6198\text{-}3.3605)\\ &= 583.111 \times 2.2593\\ &= 1319.7\ \text{kJ/kg}\end{aligned}$$

This is illustrated in *Figure 1.2*.

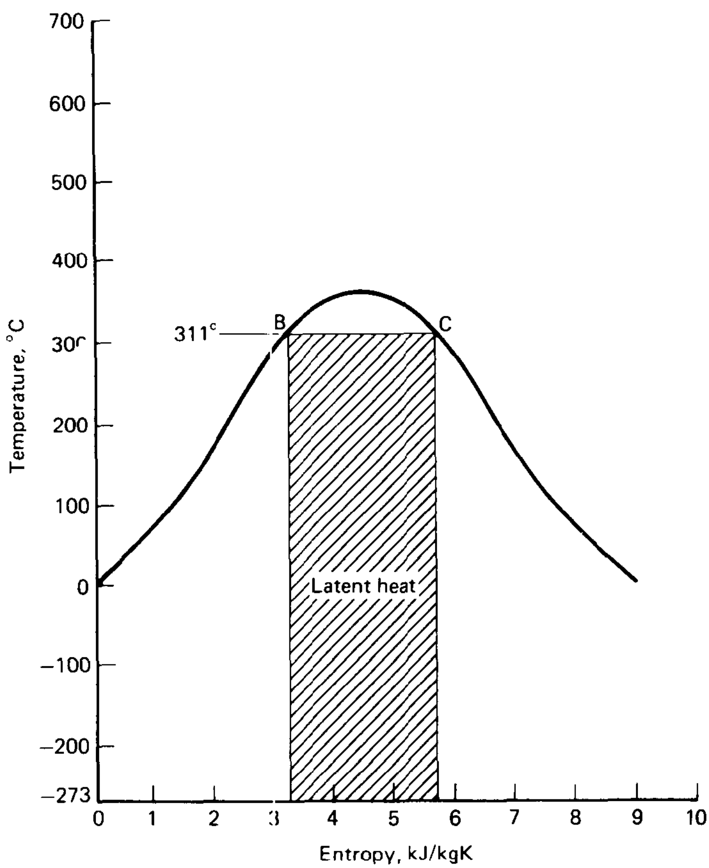

Figure 1.2. Representation of latent heat

Lines of constant pressure are horizontal in the water/steam zone and rise rapidly in the superheat region as shown in *Figure 1.3(a)*, while lines of

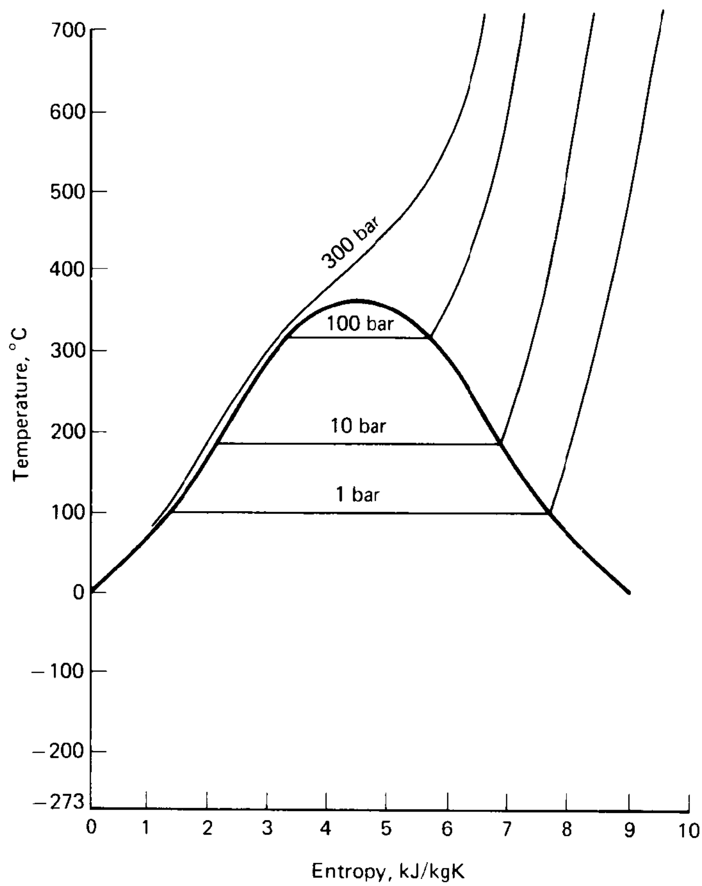

Figure 1.3 (a). Lines of constant pressure

constant wetness are shown in *Figure 1.3(b)*. Of course a line of constant dryness of, say, 90% is the same as a line of constant wetness of 10%.

Supercritical conditions

Reference to the temperature–entropy diagram shows that the 'boiling water line' and the 'dry saturated steam lines' come closer together, the higher the temperature. In other words, the quantity of latent heat required to convert the boiling water to steam becomes less.

A point is reached where the boiling water and dry saturated steam lines meet and so the associated latent heat is zero. This is known as the 'Critical

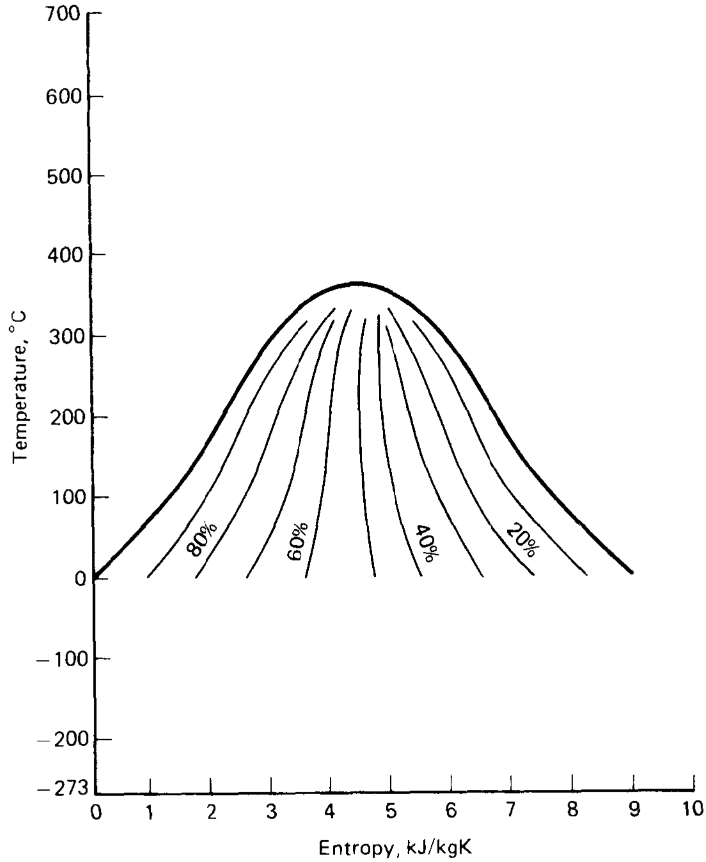

Figure 1.3 (b). Lines of constant wetness

Point' and occurs at the following conditions:

Critical pressure	221.2 bar absolute
Critical temperature	374.15°C
Critical volume	3.17 dm^3/kg

At more elevated conditions the steam is 'supercritical'. Thus, if water is at a supercritical pressure and is heated the temperature will increase until, at a particular value, the water will flash instantaneously into steam and superheating will commence. There is no change of specific volume from the liquid to the dry steam state.

The temperature at which water at a given supercritical pressure will flash to steam is not precisely known, but the pseudo transition locus shown in *Figure 1.4* gives an indication. The supercritical boilers at Drakelow operate at about 250 bar and so the transition from water to steam takes place at about 385°C.

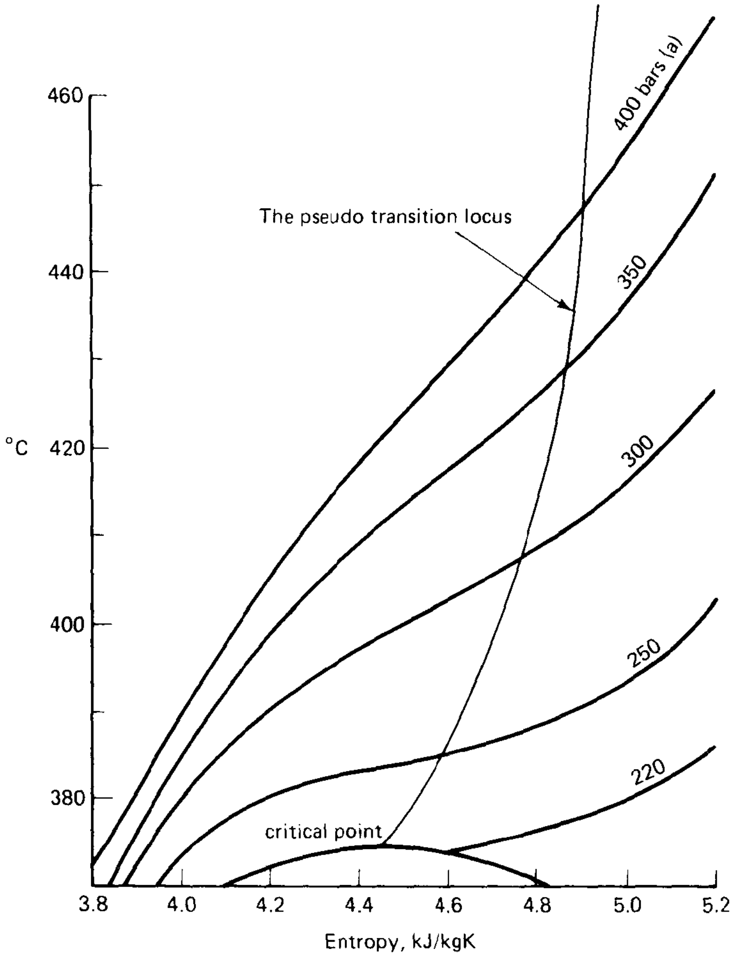

Figure 1.4. The pseudo transition locus

The Ideal (or Rankine) cycle

It is axiomatic that every component in an ideal cycle works perfectly. Thus, the condensate is heated such that it is continuously at saturation temperature from the time it leaves the condenser until it is at boiler pressure; work is done in the turbine by isentropic expansion of the steam; in the condenser only the latent heat of the steam is removed, and so the condensate is at the boiling temperature corresponding to the back pressure, and so on.

The basic cycle is shown in *Figure 1.5*, which shows the conditions for:

Steam pressure	100 bar absolute
Steam temperature	566°C (839K)
Back pressure	30 mbar (saturation temp. 24.1°C)

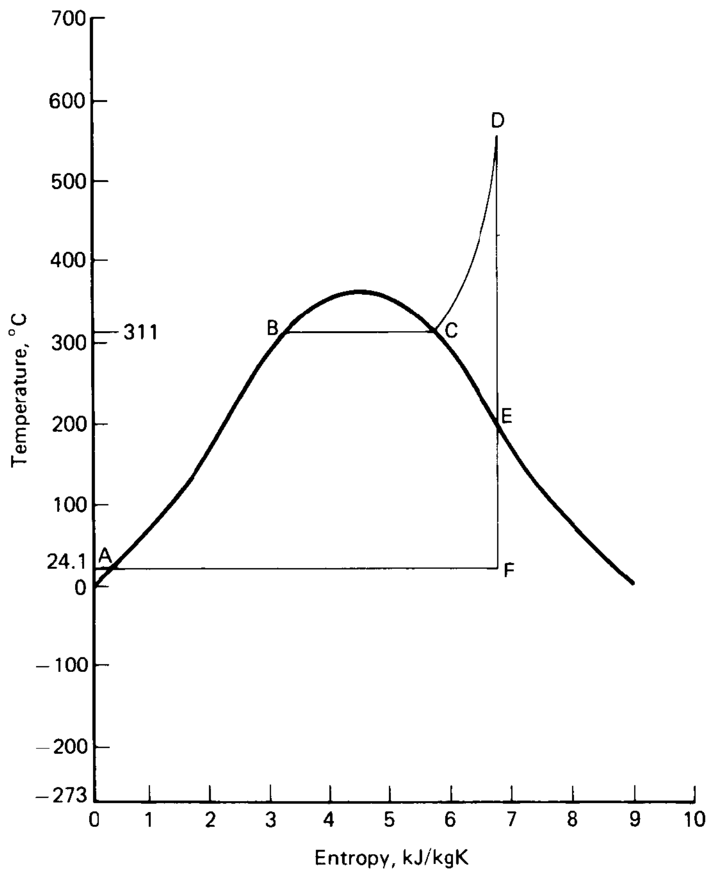

Figure 1.5. Basic Rankine cycle

At point A the condensate is at the boiling temperature corresponding to the back pressure. It is then returned to the boiler via a heating system such that as the temperature increases so does the pressure, and so the feed is continuously at the boiling temperature corresponding to the pressure. At B the boiling water is at a pressure of 100 bar where it is evaporated in the boiler. The latent heat addition is represented by the line BC and at C all the water has been evaporated and superheating commences. This is shown by CD and at point D the superheated steam temperature is 566°C.

The steam then expands isentropically inside the turbine as shown by the line DEF. At point E there is no superheat left in the steam and so from E to F there is increasing wetness. At F the steam is at a pressure of 30 mbar and is passed out of the turbine to the condenser and condensation of the steam takes place as represented by the line FA. At point A the steam has all been condensed and the condensate is at boiling temperature ready to begin another cycle.

To summarize the above:

AB – heating of feed water (i.e. sensible heat addition)
BC – evaporation of water in boiler (i.e. latent heat addition)
CD – superheating of steam (i.e. superheat addition)
DF – expansion of steam in turbine. Point E is the demarcation between superheated and wet steam
FA – condensation of the steam in the condenser

More detailed study of the ideal cycle

(a) *Sensible heat addition*

In *Figure 1.6* the sensible heat is represented by the area under the line AB. At point A the temperature is 24.1°C and at B it is 310.961°C, the values being obtained from steam tables. Also from the tables the sensible heat from A to B is $1408 - 101 = 1307$ kJ/kg.

If, however, the heat represented by the area 0°C YB in *Figure 1.6* were known, the heat at B could have been calculated from:

Heat at B = area under YB − area 0°C YB
= (absolute temperature at B × entropy change from 0 to B) − area 0°C YB

Willard Gibbs determined values for the areas such as 0°C YB and he called them the 'Negative Thermodynamic Potential', but the common name is 'Gibbs Function'. In the 1939 Callender's steam tables it is denoted by G; in the 1972 *C.E.G.B. Abridged Steam Tables* it is denoted by g and called 'specific free enthalpy'.

To find the sensible heat using Gibbs' function in the above case first calculate the heat at point B, i.e. 310.961°C and 3.3605 entropy:

$$\therefore \text{Sensible heat at B} = [(310.961 + 273.15) \times 3.3605] - g$$

From steam tables the value of $g = 554.9$ kJ/kg.
So heat at B $= 1962.90 - 554.9$
$= 1408.00$ kJ/kg

But the water was already at 24.1°C before heating took place, so it already contained the equivalent heat of water at 297.25 K, 0.3544 entropy.

So initial heat = $(297.25 \times 0.3544) - g$
From steam tables $g = 4.4$ kJ/kg
.. initial heat = $105.4 - 4.4$
= 101.0 kJ/kg
Therefore the added heat to raise the water from 24.1°C to 310.961°C
= $1408.00 - 101.0$
= 1307 kJ/kg as before

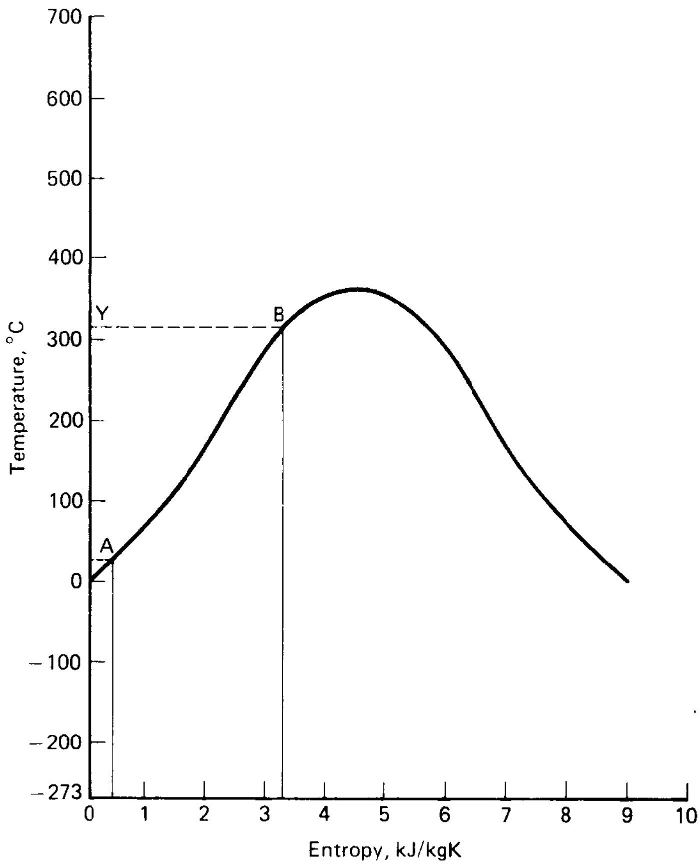

Figure 1.6. Specific free enthalpy

Increasing the pressure increases the quantity of sensible heat per kilogram of water as shown in the random examples shown in *Table 1.1* and in *Figure 1.7*.

Usually, increased pressure is associated with increased output of the turbo-alternator, so not only does increasing the pressure mean more sensible heat