Steam cycles for waste heat recovery: A case study

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Synopsis

The paper describes a thermodynamic study of single pressure and dual pressure steam cycles for electrical power recovery from the exhaust gases of a slow speed diesel engine. The study was for an engine mcr output of 5850 kW. In the single-pressure cycles, fixed and floating pressures were evaluated together with an external feed heating option. The dual-pressure cycles were also evaluated with an external feed heating option. Engine manufacturer's exhaust gas data was compared with in-service results and minimum stack temperatures were examined.

1. Introduction

The engine chosen for the study was one of the new generation of slow speed diesels, far smaller than is now considered viable for efficient power recovery from waste heat in general purpose cargo ships where a normal service base electrical load of approximately 270 kWe is expected. An engine of this output may also be considered for any application with a maximum continuous load not exceeding 6 MW and where suitable fuels are available. In such cases the overall economics may justify the additional capital outlay of a Waste Heat Recovery Plant (WHRP) by reducing generation cost. Substantial amounts of heat are also available from the engine cooling systems and these of course can be used for suitable processes but this study is limited to the thermodynamic design of steam cycles for WHRP for the efficient recovery of power.

There are a number of factors of fundamental importance when defining the design parameters of the cycles and, perhaps unavoidably, there are uncertainties associated with them. These are described in the following paragraphs.

Considerable uncertainty is associated with engine manufacturers' data for exhaust gas temperatures and mass flow rates at "in-service" conditions. The results of a detailed study of operational data from B & W and Sulzer engines, were given by Owen [1], and corroborate the findings of an earlier study by Cusdin and Virr [2], i.e. the exhaust gas temperature is generally some 33-35 °C higher than the design figure. Cusdin and Virr also found higher than expected exhaust gas quantities and velocities however no details of mass flow rates are given in the later study. Later results from a Mitsui study [3] showed a similar increase in temperature over a period of several months from commissioning.

To avoid low temperature acid corrosion in the economizer section of a WHRP boiler, a stack temperature of at least 169 °C has been used in the past. Results shown in [1] indicate that the dew point temperature in the exhaust gases of the main engines observed, was approximately 122 °C. It was also stated that the maximum deposition of sulphuric acid occurs at about 25 K below the dew point and that it appears feasible to use a WHRP boiler outlet temperature of 140 °C without resource to expensive corrosion-resistant materials. It should be noted that this analysis was of the exhaust gases of marine diesel engines and does not apply to fired boilers. In a Japanese study (4) the corrosion rates of several materials: mild steel, stainless steel, lead alloy plated steel, and fluorine synthetic resin coated steel, were measured in a simulated diesel exhaust gas environment at temperatues of 60-120 °C. Only the lead alloy plated and synthetic coated steels had adequate resistance. The study was connected with a bottoming cycle using a fluoro-carbon (R-11) as the working fluid. Lower boiler inlet temperatures from the new generation of diesels forces consideration of lower stack temperatures to maintain the quantity of heat available for power generation.

The minimum gas/water temperature difference is of prime importance for both the performance of the plant and the cost of the boiler. Foster-Pegg [5] suggests pinch points of 11-20 K as being optimal for unfired WHRP boilers. In a more recent study [6] of WHRP for a 25 MW medium speed diesel power station, an optimum pinch point of about 30 K was suggested. In previous studies Morton [7] and the author [8] have used pinch points of 15-30 K.

2. Thermodynamics

2.1 Engine data

Exhaust gas temperatures and mass flow quantities for a M.A.N.-B & W 5L60MCE engine are given in [9]. The data used was for an engine running at ISO reference conditions (air temperature 27 °C at the turbo-charger inlet, cooling water inlet temperature 27 °C and burning fuel oil with a LCV of 42.7 MJ/kg). Temperatures together with an estimation of the variations which occur in service are shown in Fig. 1. Reasons for the variations were given by the author in [10] and briefly they are:

(a) Fitting an uncooled turbo-charger increases the temperature at least 10 K. Partially bypassing exhaust gas around the turbine increases the temperature approximately 20 K but doing so can incur a penalty of approximately 1.3 g/kWh on the specific fuel consumption.
(b) Exhaust temperature increases approximately 30 K

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when burning residual fuel oils compared to distillate fuel oil. ISO condition fuel may not equate to distillate fuel but neither does it equate to poor quality residual fuel oil and it is believed that a temperature increase of 10-20 K can be reasonably expected when burning a residual fuel.

(c) The average air temperature in many enclosed machinery spaces is nearer to 40 °C than 27 °C and a corresponding increase in exhaust temperature was noted by Owen. Similar ambient conditions may prevail in many industrial installations.

(d) A temperature increase due to a gradual deterioration in main engine components between overhauls and for which an estimate of 5-20 K is believed to be fair.

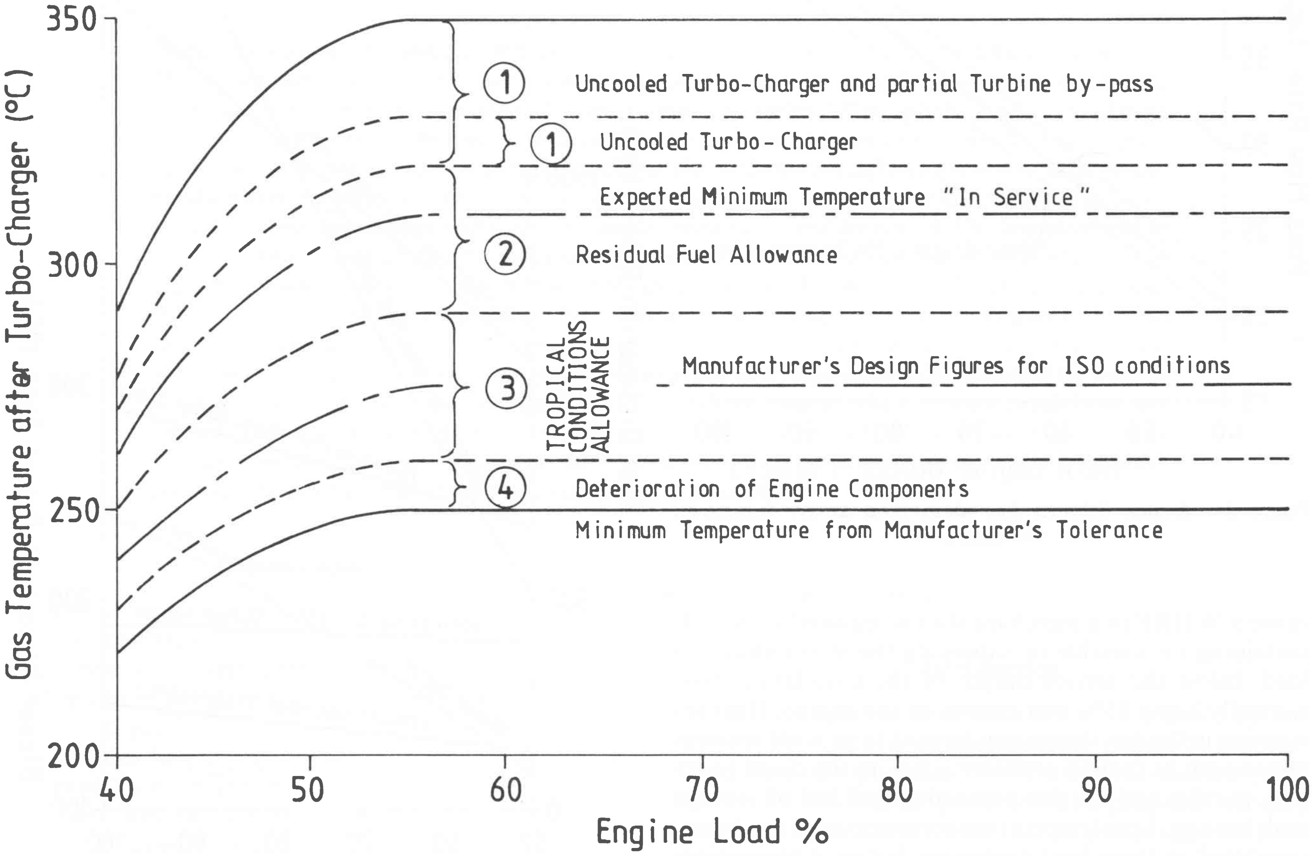

Figure 1 – Explanation of the Variations in the Exhaust Gas Temperature of a Large Diesel Engine

M.A.N. give a gas stream temperature of 275 °C at the inlet to the exhaust boiler for an engine with an uncooled turbo-charger, and 265 °C for an engine with a fully water-cooled turbo-charger. If a WHRP is to be installed it is logical to select an uncooled turbo-charger. For design purposes it is believed that a gas stream temperature at the boiler 35 K above the engine manufacturer's figure could be accepted, i.e. 310 °C however, as a precaution the thermodynamic properties of the steam cycles were optimized for an initial gas temperature of 300 °C.

M.A.N. data for the exhaust gas mass flow are shown in Fig. 2 and the design line was accepted for heat transfer purposes. Any reservations felt in this are offset by the use of a conservative value for the initial gas temperature. The specific fuel consumption of the engine at ISO reference conditions is given as 167 g/kWh and the corresponding thermal efficiency is 0.505.

2.2 Insulation of combustion chamber components

The insulating effect of ceramic coated piston crowns upper sections of cylinder liners, cylinder heads and the faces of exhaust valves was recently described by Kyrtatos [11]. A reduction in the amount of energy formerly rejected to the surroundings is expected to primarily increase the energy available in the exhaust gases for power recovery from waste heat. An increase of almost 17% was predicted for a two-stroke uniflow diesel engine with a mcr output of 9 000 kW. When the current materials problems, relating to adhesion of the ceramic coatings and obtaining acceptable corrosion and wear resistance in them, are overcome, the potential of efficient power recovery will be enhanced. Advanced materials problems rarely have rapid solutions, however that should not inhibit the development of WHRP for the current generation of engines.

2.3 Boiler temperatures

Although it appears feasible to have a WHRP boiler outlet temperature of 140 °C and still use conventional materials for the economizer section, the steam cycles were optimized for a stack temperature of 160 °C and the engine running at 70% full load. The latter was chosen be-

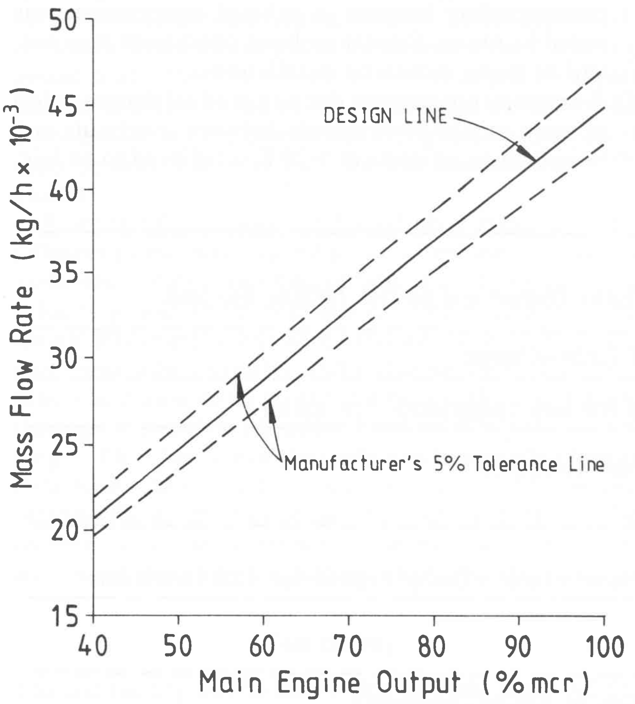

Figure 2 – Engine Exhaust Gas Mass Flow at ISO Reference Conditions

cause a WHRP in a merchant ship is required to be self-sustaining i.e. capable of supplying the ship's electrical load, below the service output of the installation, that normally being 85% mcr output of the engine. Heat remaining in the gas stream can be used to provide requirements such as fuel oil preheating before the diesel injectors, purifier and clarifier preheating and fuel oil storage tank heating. Low temperature corrosion need not be encountered in these heat exchanges but as a precaution, resistant materials could be used in them. Separating the power recovery and heating systems has additional advantages:

(a) As Morton states, it is a cardinal thermodynamic error to use heat from the hot end of the gas stream for any purpose other than power recovery, and this is avoided.
(b) If steam is to be used for fuel oil heating and leakage of oil into the condensate occurred, contamination of boiler heating surfaces is avoided.

A pinch point of 25 K was adopted as being suitable for conceptual design purposes.

2.4 Feed heating

Morton also points out that the feed water temperature at the economizer inlet must be maintained at about 120 °C to minimize corrosion. This can be achieved by recirculating saturated water from the drum, or, by fitting a direct-contract dearator, or, by using an external source of heat to raise the temperature of the feed water. There are considerable amounts of heat available at the various temperatures in the cooling systems of large diesel engines and estimated data for this engine are shown in Fig. 3. It can be seen that at the engine mcr output there is approximately the same amount of heat energy available from the charge air as there is from the exhaust gas. A temperature of 120 °C at the economizer inlet was adopted for all of

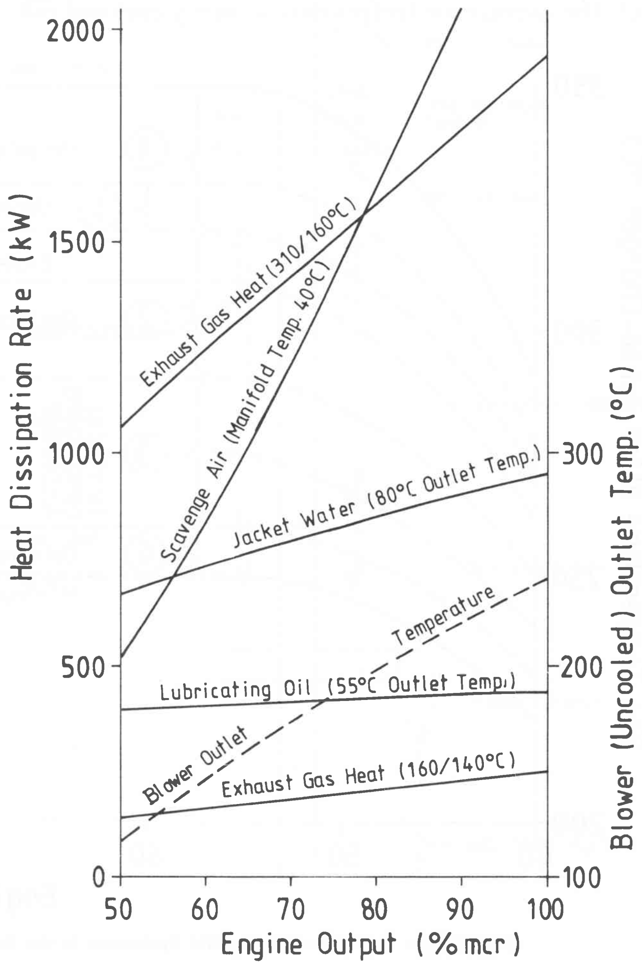

Figure 3 – Estimated Heat Dissipation Rates and Turbo-Charger (uncooled) Air Outlet Temp.

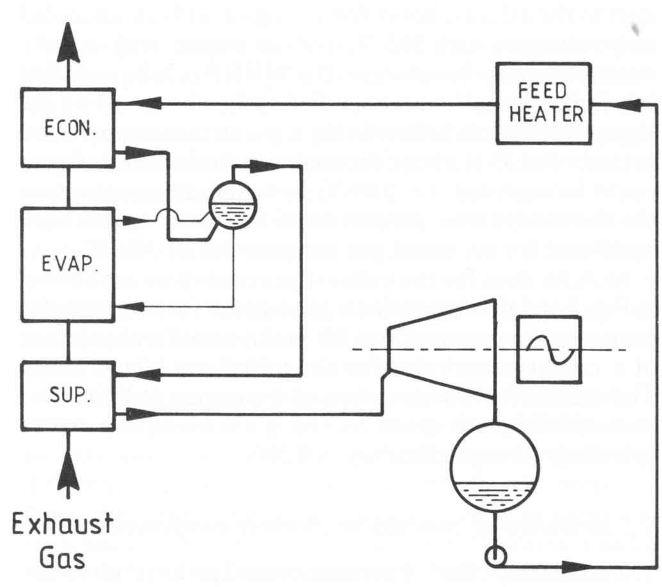

Figure 4 – Single – Pressure System Diagram

the steam cycles and in those which include an external feed heating option, this was taken as being provided by the charge air system.

2.5 Steam cycles

The same thermodynamic cycle as for previous studies [7] [8] of Rankine cycles with superheat and with the option of a single stage of external feed heating, was adopted, see Fig. 4.

The dual-pressure steam cycles were modified from the previous studies as suggested (reply to contribution p 67 [7]) by omitting the LP superheater, see Fig. 5. Only the HP steam supply to the turbine would be superheated. The system includes the option of a single stage of external feed heating.

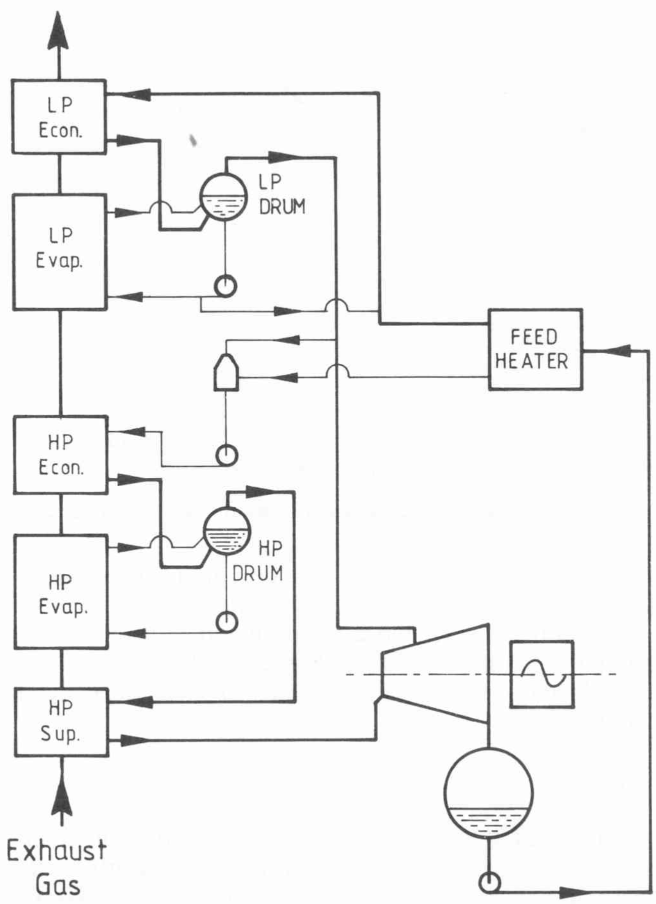

Figure 5 – Dual – Pressure System Diagram

2.6 Power recovery gas turbines

The latest engines in the series have as an option, a gas turbine connected in parallel with the turbo-charger and approximately 10% of the total exhaust gas flow passes through it. The power turbine is geared to the main engine and the power recovered is believed to be 2-3% that of the engine output, however, this option was not studied here.

3. Results

Now, the optimizing of thermodynamic cycles inevitably involves computational methods and a detailed analysis of each cycle was carried out. The results are described in the following sections. The basic parameters assumed for the cycles are shown in Table 1. The values for the boiler heat duty refer to heat transfer at a gas inlet temperature of 350 °C. In the results for the steam cycles:

$$\text{recovered power ratio} = \frac{\text{recovered power}}{\text{main engine power}}$$

Recovered power ratio as a measure was used by Morton [7], the recovered power term refers to the mechanical output of the turbine and this was used here. This ratio can be applied to all combined cycles. It will not diminish the importance of overall thermal efficiency, η_0, as the fundamental measure of efficiency for combined cycle plants.

3.1 Perfect power recovery cycle

The maximum possible efficiency of a WHRP receiving heat from a fluid of specific heat C_p at an initial temperature T_1, cooled to a temperature T_2 as it passes through the heat exchanger and rejecting heat at absolute temperature T_0, is

$$\eta_{\text{Ideal}} = \frac{W_{\text{Carnot}}}{Q} = 1 - \frac{T_0}{(T_1 - T_2)} \log_e \frac{T_1}{T_2}$$

per unit mass of fluid (8).

The recovered power for a thermodynamically perfect cycle would be

$$W_{\max} = C_p\left(T_1 - T_2 - T_0 \log_e \frac{T_1}{T_2}\right)$$

per unit mass of fluid.

The ideal efficiency and the perfect power recovery for a M.A.N.-B & W 5L60MCE diesel at typical exhaust gas temperatures and engines outputs are shown in Fig. 6. In each case a stack temperature of 160 °C was assumed.

3.2 Single-pressure steam cycles

The higher grade of heat available in exhaust gases of the past generations of slow speed engines enabled sufficient power to be recovered by a single-pressure steam cycle plant for installations of 10 000 kW and above. Below that output comparatively cheap fuels made waste heat recovery unattractive economically. That situation has, of course, changed and is unlikely to return in the foreseeable future. Single-pressure steam cycle plants are comparatively simple to construct and control, and have widespread use.

Analysis of the power recovery potential from single-pressure steam cycles for the installation under consideration are shown in Figs. 7/8. The fixed-pressure steam cycles were optimized for an initial gas temperature of 300 °C. As could be reasonably expected, floating-pressure cycles offer improved power of recovery of approximately 5% at both temperature extremals. External feed heating is slightly advantageous at low temperature but at

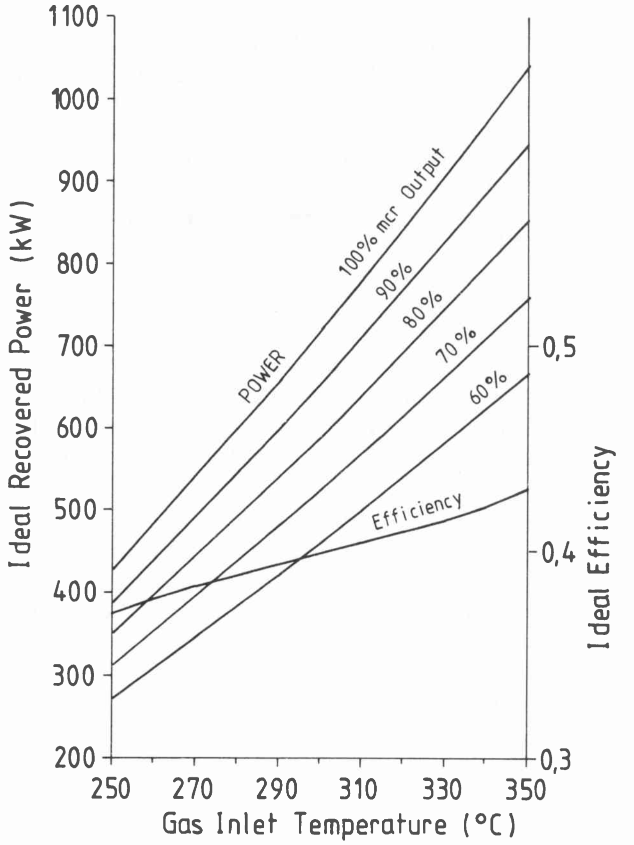

Figure 6 – Perfect Power Recovery Cycle: Power and Efficiency

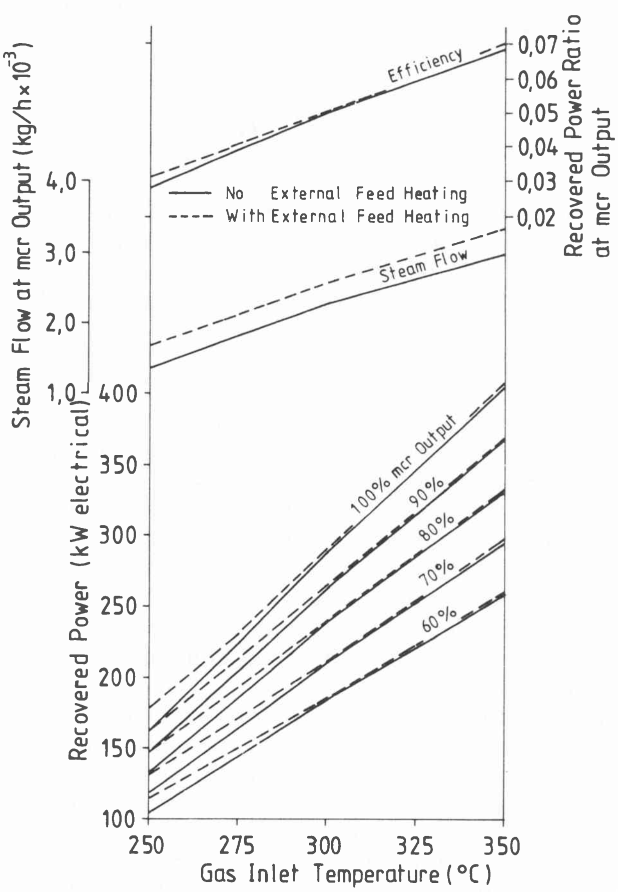

Figure 7 – Fixed-Single-Pressure Steam Cycle Performance

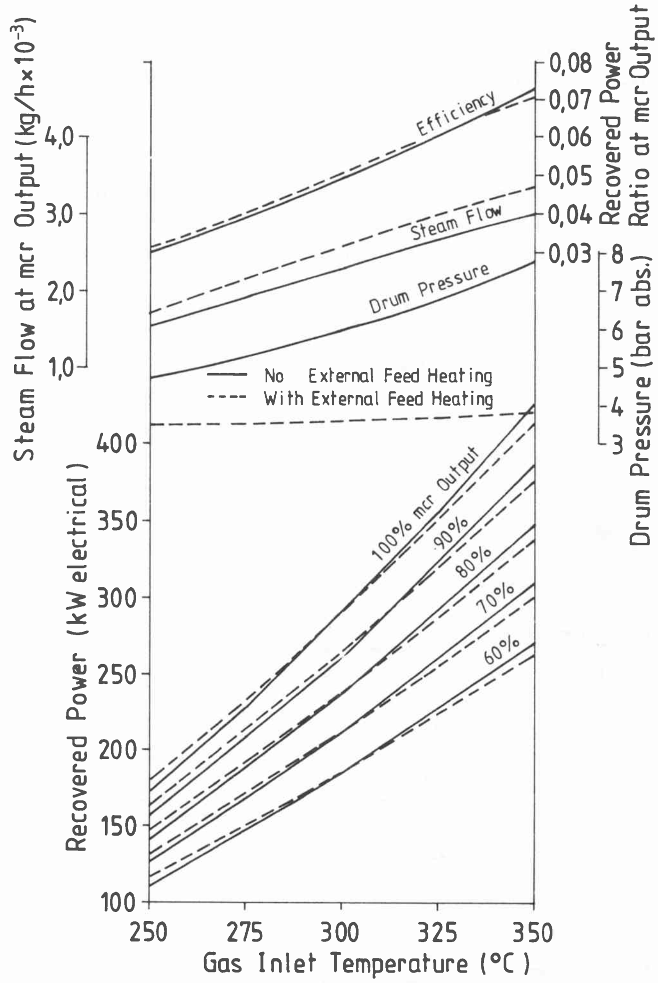

Figure 8 – Floating-Single-Pressure Steam Cycle Performance

the higher temperatures it can be counter-productive. This is substantiated by earlier results [8].

Running at the engine mcr output and with an exhaust gas temperature of 310 °C, the thermal efficiency of a combined cycle plant using the most efficient steam cycle would be 0.535.

3.3 Dual-pressure steam cycles

Analyses of the power recovery potential from dual-pressure steam cycles are shown in Fig. 9. Both cycles were optimized for an initial gas temperature of 300 °C. The thermodynamic advantages of using a dual pressure cycle are clearly shown in Fig. 10. A dual pressure system with external feed heating offers power gains of 20-24% over a simple single-fixed-pressure system, whilst a dual-pressure system without external feed heating offers power gains of 11-15%.

At the end of the isentropic expansions the dryness fraction of the steam in the HP and LP exhausts are 0,80 and 0,84 respectively, and at the end of actual expansions are exptected to be approximately 0,95 and 0,92 respectively. The latter being satisfactory for the operation of the turbines.

It is interesting to compare pressures: HP 19 and 16 bar; LP 4.3 and 3.3 bar for cycles with and without feed heating respectively, to a Mitsubishi design [12]. The pressure ranges quoted for the dual-pressure economizer were: HP 6.9-16.7 bar; LP 2.9-4 bar.

Running at the engine mcr output and with an exhaust gas temperature of 310 °C, the thermal efficiency of a combined cycle plant using the most efficient steam cycle would be 0.545.

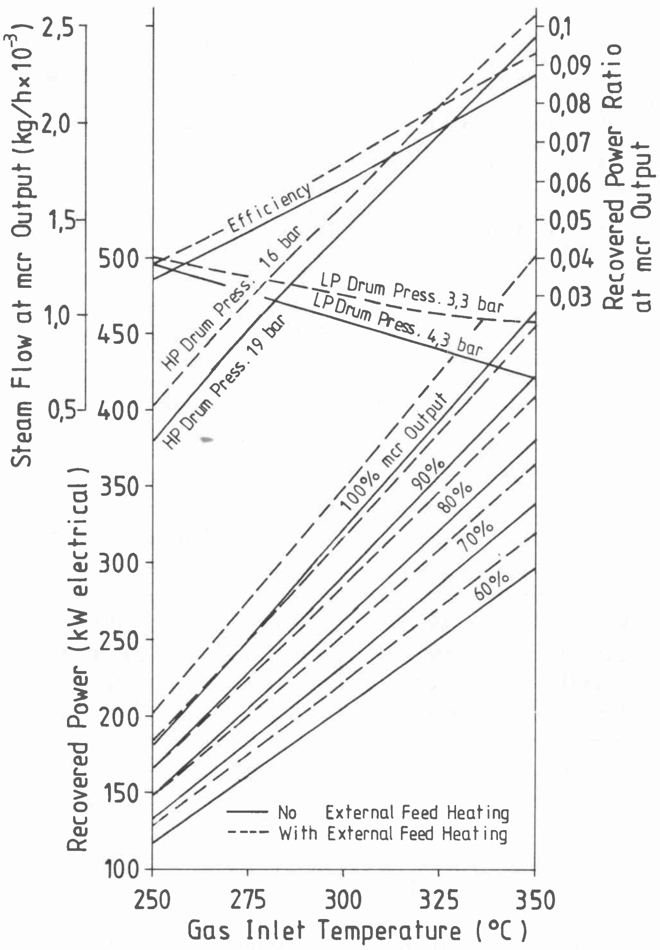

Figure 9 – Dual-Pressure Steam Cycle Performance

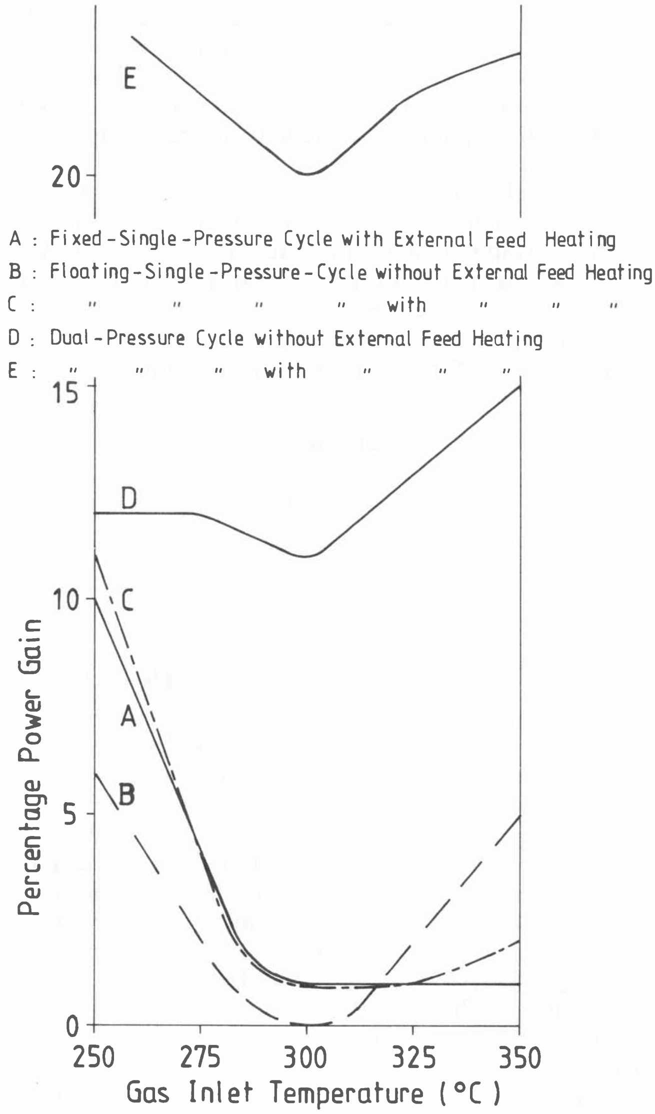

Figure 10 – Steam Cycles Performance relative to Fixed-Single-Pressure Cycle without External Feed Heating

3.4 Organic fluid cycles

The thermodynamic and physical and chemical requirements of organic fluids suitable for use in WHRP's have been well described [1], [7], [13]. However it was shown in Tables II/IV (7), that, dual pressure steam cycles can offer thermal efficiencies comparable to those of organic Rankine cycles: 0.21-0.25 cf. 0.2-0.29. At this early stage in the development of these fluids and systems for them, there does not appear to be a valid case for their use, neither thermodynamically nor economically. That does not preclude future changes, such as the lowering of the initial temperature of the gas stream which could alter the balance.

Accordingly, the performance of only single- and dual-pressure steam cycles were evaluated.

4. Conclusions

In the aproaching decades mankind faces two problems of global magnitude; environmental pollution and diminishing fossil fuel resources. In response to both of these, power and heat recovery from the waste heat sources of devices which convert the chemical energy of reactants into mechanical or electrical power, can only assume a far greater role.

The following comments appertain only to a M.A.N.-B & W 5L60MCE slow speed diesel of 5850 kW mcr output running at ISO reference conditions. Similar considerations would apply to similar enginges but it is emphasized that fixed-pressure steam cycles can only be optimized for particular thermodynamic conditions.

(a) A temperature of 310 °C could be expected in service, 3-4 months after commissioning, for the gas stream after the turbo-charger, this being 35 K above the temperature given in the engine manufacturer's data. Thus a WHRP may not attain full-performance until the end of this period. The reality of this situation is that at commissioning a diesel exhaust gas temperature may be only 250 °C and there will be a deficiency in output from a WHRP. This can lead to disputes over liability between the engine builder, WHRP manufacturer, and installation owner. Obviously it is in the best interests of all the parties to obtain the maximum thermal efficiency for the operational life of the installation and there may not be a simple solution to this problem.

(b) The ideal thermal efficiency is approximately 0.4 at the 70% engine load and expected service conditions, the corresponding ideal recovered power is 570 kW. At the engine mcr output these increase to 0.43 and the 780 kW respectively.

(c) A dual-pressure system with external feed heating would be by far, the most efficient steam plant.

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Table 1 Data for the steam cycles

	Fixed single-pressure cycles		Floating single-pressure cycles		Dual-pressure cycles	
	Without external feed heating	With external feed heating	Without external feed heating	With external feed heating	Without external feed heating	With external feed heating
Final gas temp. (°C)	160	160	160	160	160	160
Minimum pinch point (°C)	25	25	25	25	25	25
Superheated steam temp. (°C)	275	275	275	275	275 LP dry sat.	275 LP dry sat.
Superheater and line pressure drop (%)	10	10	10	10	10	10
Combined turbine and alternator efficiency	0.66	0.66	0.66	0.66	0.66	0.66
Minimum dryness of exhaust steam HP					0.78	0.8
Minimum dryness of exhaust steam LP	0.87	0.9	0.88	0.9	0.88	0.84
Condenser pressure (bar)	0.1	0.1	0.1	0.1	0.1	0.1
Boiler heat duty (kW)	2 445	2 445	2 460	2 460	2 441	2 495
Feed inlet temp. to economizer (°C)	46	120	46	120	46	120
Drum pressure (bar) HP					19	16
Drum pressure (bar) LP	6.0	3.6	4.7-7.8	3.5-3.8	4.3	3.3
Feed heater duty (kW)	0	290	0	295	0	315