ANALYSIS OF A TURBINE CYCLE
WITH A DRY COOLED CONDENSER

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ABSTRACT

The availability of low grade coal in areas with very little or no water has led the Electricity Supply Commission of South Africa to embark on a bold development to reduce the water consumption of individual power stations from the order of 2.5 to 0.5 liters/kWh. This requires a finned tube heat exchanger 70m wide and almost 500m long for the 3600 MW (nominal) Matimba Power Station, with 288 fans 9m in diameter to push more than 600 000 t/h of air through the system.

The optimisation of the cycle and studying the effect of various parameters on the cycle performance can be achieved using the PEPSE computer program. A PEPSE model for Matimba was set up which includes some 77 components and 101 streams. The turbines are modelled using the G.E. procedure for fossil turbines (GER-2007C), but the high pressure turbine is modelled using the General Turbine component (type 08) since this system does not include a governing stage. At present the dry condenser is modelled using the Standard Condenser component (type 10) with the cooling fluid source adapted to the specific requirements of the model. The cooling water, which represents the actual cooling air, has to represent the heat removal capability of air with an ambient temperature of up to $44^{\circ}C$. At the station elevation of 877m above sea level the barometric pressure is 91.3kPa. The total fan capacity of a unit is represented by the equivalent of 7 870 kg/s of water.

Further steps to include more sophisticated dry cooled condenser models are included in the last part of this work.

1. INTRODUCTION

The demand for electricity in South Africa has been increasing over the past 30 years at the average rate of some 8% p.a., and the present installed capacity of Escom (Electricity Supply Commission of South Africa) is 22 000MW. To fulfill the energy needs of the country till the end of the century, Escom is undertaking a series of projects to construct large power stations, mostly of the nominal capacity of 3 600MWe (6x600) each. Total plant under construction and on order is now 21 000MW. The availability of low grade coal in areas with very little or no water has led Escom to embark on a bold development to reduce the water consumption of individual power stations from the order of 2.5 to 0.5 liters/kWh. This is going to be achieved by South Africa becoming the world leader in dry cooling capacity, as an all dry direct condensation cooling system will be employed at the Matimba station now under construction near one of South Africa's large coal deposits[1]. This dry system is similar to that used at the 330 MWe Wyodak plant in Wyoming (U.S.A.), and is illustrated in Fig. 1[2]. Another type of dry cooling being used is the indirect dry cooling system. This method is already used at two of Escom's 6x200 MWe Grootvlei power station units and is to be used for the new 6x600MWe (nominal) Kendal power station. A few similar types of dry cooling systems are already installed around the world but are of a smaller scale. An alternative dry-cooling system is under development by EPRI, which will use a phase-change loop with ammonia as the heat transfer medium[1] similar to the system proposed in Germany by MAN some years before.

This work describes the modelling effort done by Escom related to the Matimba power station, and particularly the efforts to model its directly dry cooled condenser. The proprietary computer program PEPSE[3] running on Escom's CDC Cyber 175 computer is the main tool used to achieve this purpose. Some desk-top computer programs have been used to fit turbine and condenser plant vendors' data to a format suitable to be used in PEPSE.

The major objective of the work is the optimization of the cycle details and studying the effect of the various parameters on the cycle performance. Also, this work is aimed at preparing tools to aid decision making regarding future power stations in general.

2. THE TURBINE CYCLE

The details of the Matimba turbine cycle, and the way it is modelled using PEPSE are described in full in an internal Escom report[4]. The model includes some 77 components and 101 streams, and the parts concerning the condenser cooling are discussed in the following section. Results for a performance mode calculation are presented in Fig. 2, which are within 0.15% from the vendor's heat rate. The turbines are modelled using the G.E. procedure for fossil turbines (GER-2007C). As the turbine expansion differs from the expansion according to the G.E. procedures (the turbine is of German design) it was necessary to determine the correction factors (EFMULT and SHAPER) which adjust the G.E. expansion to the required expansion. These factors were initially established using the "controls" option of PEPSE.

The high pressure (HP) turbine is modelled using the General Turbine component (type 08) since the Matimba system does not include a governing stage. For the intermediate pressure (IP) and low pressure (LP) turbines it was found that using the PEPSE "controls" option the "SHAPER" which dictates the slope of the expansion line at any point in the enthalpy-entropy (Mollier) diagram was the same for every stage of each section. For adjustment of this to the vendor data a separate computer program was written on a desk-top computer to analyse each stage separately. The program represents the mathematical model described at section 8.1.1 of Vol. II of the PEPSE Manual[3]. The expansion analysed this way gave for every stage a different value for the "SHAPER". Applying those factors in a PEPSE run seems to have improved the result.

Another point that came across was an apparent restriction of PEPSE that if the efficiency of the first LP stage was better than that of the last LP stage it could occur that it was impossible to find a "SHAPER" value which would give the desired entropy at the end of the first LP stage.

For exhaust loss calculation the Brown-Boveri method of total exhaust loss is applied. This is so because in the vendor steam expansion line, the exhaust loss "tail" of the expansion curve is on a constant pressure line. The design pressure at the end of the turbine expansion line is 22 kPa. This higher exhaust pressure corresponds to the higher heat sink temperature which is due to the fact that in an all dry system the limiting temperature is the ambient dry-bulb temperature - typically 15-20°C higher than the wet bulb temperature on a hot day [2].

3. THE CONDENSER COOLING

The Matimba air cooled condenser is located alongside the turbine hall and at an elevation of 45m above ground level. Steam is ducted directly from the turbine exhaust to the condenser and condensate drained back to the condensate extraction pumps. The condenser consists of a series of A-frame units having banks of finned tubes on the sides with fans at the base of the A-frames. Steam is condensed in the tubes by the flow of air over the outer finned surface. The total area occupied by the condensers of six units is 500m by 70m. There are 288 fans 9m in diameter with a total capacity of over 600 000 t/h.

At present the dry condenser is modelled in the performance mode using the Standard Condenser component (type 10) with the cooling fluid source adapted to the specific requirements of the model. The cooling water, which represents the actual cooling air, has to represent heat removal capability of air with an average ambient temperature of up to 40°C. At the station elevation of 877m above sea level the barometric pressure is 91.3 kPa. The total fan capacity for the direct condenser for each of the six turbo-generators is represented by the equivalent flow of 7 870 kg/s of water, which takes into account the ratio of the relevant specific heats.

Further steps to include more sophisticated dry cooled condenser models are under way. The intention is to modify the condenser model, so that PEPSE will find that end point of the expansion line whereby the actual heat rejection necessary to condense the steam from that point along the constant pressure line, is consistent with the calculated heat rejection of the condenser. For this the condenser has to be analysed in the design mode. The way this problem is being tackled is as follows: From the condenser manufacturer there are curves available which will be entered as a two dimensional matrix by using "schedules" option cards. These data are interpolated in PEPSE as bivariate functions which represent the heat rejection as a function of back-pressure and ambient temperature. The number of "schedules" these data will be translated into depends on the required accuracy and each of them will represent a range of ambient temperatures. A further sophistication would be inclusion of the effect on heat transfer of the mode of operation of each fan (full or zero speed modes).

4. SUMMARY

The PEPSE modelling of the all dry direct condensation cooling to be employed at Escom's Matimba power station has been outlined. The turbine system in general and the condenser cooling section in particular, have been defined. The German design of the turbines has necessitated the modelling of the HP section using the General Turbine component (type 08) and the use of correction factors (EFMULT and SHAPER).

The present condenser model using the Standard Condenser component (type 10), and more sophisticated dry cooled condenser models were described. The work in progress is incorporating detailed design mode analysis to replace existing performance mode components.

REFERENCES

[1] T. Moore, Cooling Without Water, EPRI Journal, May 1983, p. 19.

[2] J.A. Bartz and J.S. Maulbetsch, Are Dry-Cooled Power Plants a Feasible Alternative, Mech. Engineering, October 1981, p. 34

[3] W.C. Kettenacker et.al., PEPSE Manual, Rev. 8, Energy Inc., May 1982.

[4] A.F. Madlener, PEPSE Computer Model of Matimba Power Station, Escom New Works Department Report, September 1983.

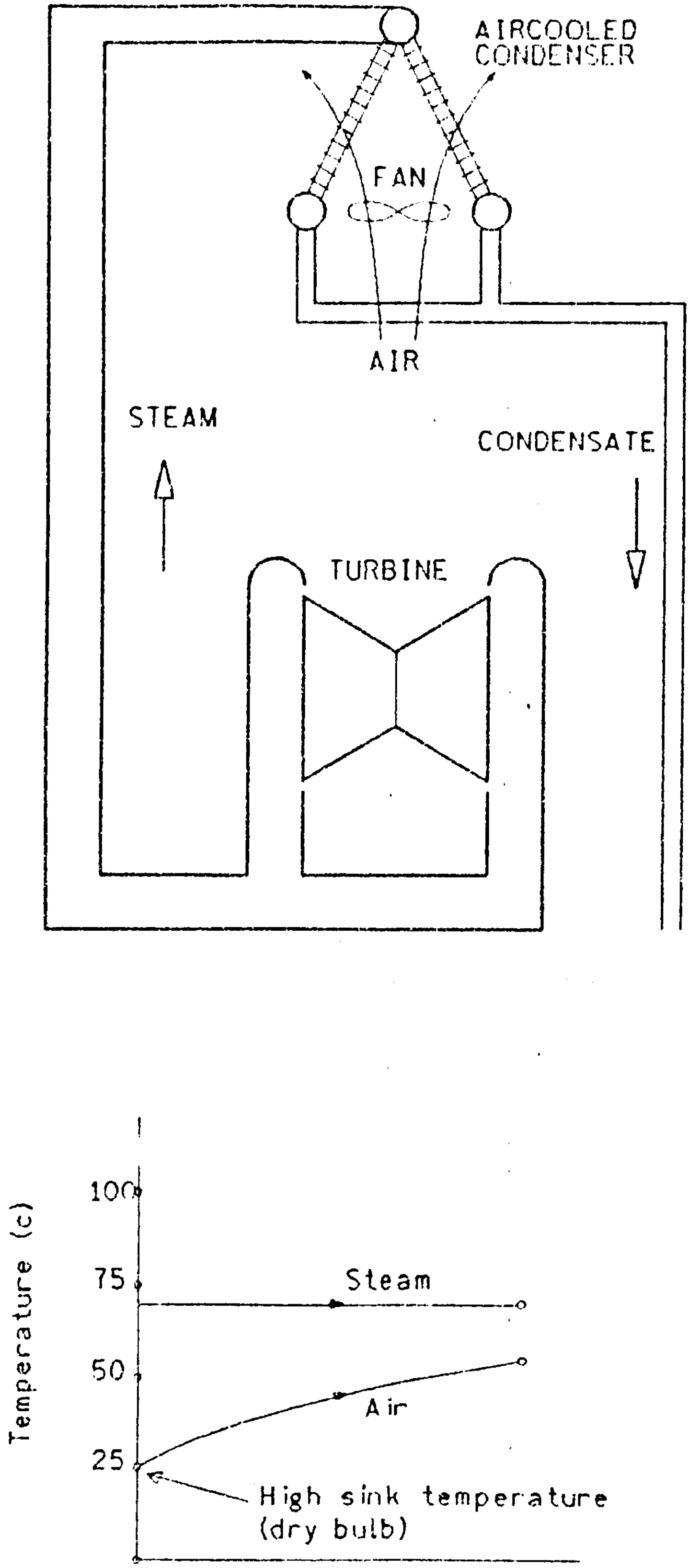

Fig. 1: A Schematic of the Direct Steam Condensation type of cooling system for a power plant together with a corresponding plot of fluid temperature variations in the heat exchanger.

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                              FIRST LAW OF THERMODYNAMICS PERFORMANCE - SYSTEM

HEAT SUPPLIED -
      BOILER HEAT TO WORKING FLUID, W                                1.30441E+09
      1ST BOILER REHEAT, W                                           2.71540E+06
      2ND BOILER REHEAT, W                                           0.
      UNSPECIFIED HEAT EXCHANGERS, W                                 0.
            GROSS HEAT SUPPLIED, W                                                 1.57524E+09
      MAKEUP HEAT SUMMATION, W                                       0.
      LETDOWN HEAT SUMMATION, W                                     -1.40111E+07
      COMPONENT VESSEL LOSSES, W                                    -1.22252E+06
      PIPE HEAT AND ELEVATION LOSSES, W                             -5.54543E+04
      PUMP INEFFICIENCY LOSSES, W                                   -1.70564E+06
      PUMP GLANDS AND SEALS LOSSES, W                               -3.44604E+05
      GENERATOR HYDROGEN AND OIL COOLER HEAT, W                      0.
            NET HEAT SUPPLIED, W                                                   1.55971E+09
      HEAT SUPPLIED AS ELECTRICAL PUMPING POWER, W                   1.90857E+07
            NET POWER SUPPLIED, W                                                  1.57779E+09

POWER OUT -
      NET TURBINE WHEEL POWER, W                                     6.75786E+08
      GENERATOR POWER WITHOUT LOSSES, MWE                                675.783
      GENERATOR MECHANICAL LOSSES, MWE                                    -2.600
      GENERATOR ELECTRICAL LOSSES, MWE                                    -7.597
            GROSS GENERATOR POWER, MWE                                                 665.586
      IDENTIFIED ELECTRICAL LOAD OF HEAT SOURCE CYCLE, MWE                 0.000
      ELECTRICAL PUMPING POWER OF TURBINE CYCLE, MWE                     -19.086
      AUXILIARY HOUSE LOAD NOT INCLUDED ABOVE, MWE                       -21.300
            NET GENERATOR POWER, MWE                                                   625.201

NET HEAT BALANCE ON WORKING FLUID -
      CIRCULATORY WATER SYSTEM LOAD, W                               9.03401E+08
      NET POWER SUPPLIED LESS NET TURBINE WHEEL POWER, W             9.02006E+08

CYCLE PERFORMANCE
                                                                                   THERMAL EFF.     HEAT RATE
                                                                                      ( - )        ( KJ/KW-HR)

      GROSS ACTUAL TURBINE CYCLE (GROSS HEAT SUPPLIED / GROSS GENERATOR POWER)        .42226          8526.

      NET ACTUAL TURBINE CYCLE (MOD NET POWER SUPPLIED / NET GENERATOR POWER)         .39537          9105.

      NET COMBINED HEAT SOURCE AND TURBINE CYCLE                                      .39537          9105.

      TURBINE CYCLE STEAM RATE,  KG/KW-HR                                3.0269

      NOTE:  MOD NET POWER SUPPLIED = NET POWER SUPPLIED - COMPONENT VESSEL LOSSES
                                    - PIPE HEAT AND ELEVATION LOSSES - PUMP GLANDS AND SEALS LOSSES
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Fig. 2: Results for a performance mode calculation for Matimba.