Some aspects of the design of cooling plant installations

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SYNOPSIS

This paper describes in broad outline the development of a computer programme for determining the effects of changes in various factors in the condenser water circuit on the operation of a cooling plant. The programme is used to obtain design figures which result in minimum total power consumption. This was done for a range of different upcast air quantities, and the implications of low air quantities for heat rejection are discussed.

SINOPSIS

Hierdie verhandeling gee in breëtrekke die ontwikkeling van 'n rekenaarprogram vir die bepaling van moontlike invloed wat verandering in verskeie faktore op die gekondenseerde watervloei in die werking van 'n verkoelingsaanleg mag hê.

Die program word aangewend om ontwerpfaktore te verkry om sodoende kragverbruik te minimaliseer.

'n Reeks van verskeie hoeveelhede uittreklug was ondersoek, en die invloed van hitteverwerping met lae lugvloeihoeveelhede word bespreek.

INTRODUCTION

The designer of an underground cooling plant installation is faced with a very large number of alternatives. Apart from the determination of the required cooling duty, which is in itself a major task, decisions have to be made concerning such items as pipe sizes, chilled water flow rates, heat exchanger sizes, siting and selection of the cooling plant, condenser water flow rates, cooling tower design, and many others. This paper describes work which has been done towards the optimization of factors in the condenser water circuit. It is assumed that the duty of the cooling plant is fixed, and that a given amount of upcast air is available for heat rejection. In the condenser water circuit optimum values for such factors as water flow rate, cooling tower cross-sectional area, and cooling tower height have to be determined, and must be chosen so as to result in a minimum total cost for the installation. The costs of a cooling plant installation comprise both capital and running costs, and changes in the variables will in general affect both these costs.

The operation of cooling plants is complicated and the estimation of the effects of various changes is not always easy. For example, a change in water flow rate effects the performance of both the condenser and the cooling tower and hence results in changes in the condensing temperature. The air pressure loss through the cooling tower and the power consumption of the condenser water pumps are also affected. Changes in the condensing temperature will in turn affect the rate at which heat has to be transferred to the tower.

The paper describes in broad outline the development of a computer programme to calculate the effects of changes in the various parameters on the performance of the cooling plant, and shows how this programme can be used to determine optimum values when only the power requirements have to be minimized.

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THE MODEL

The heat rejection circuit of the conventional underground cooling plant is comparatively straightforward. Water is pumped through the condenser, where it receives heat from the condensing refrigerant. The water then passes to a cooling tower where its heat is transferred to the upcast air stream. From the cooling tower the water returns through the pumps to the condenser.

The operation of the computer model was as follows. For given values of such variables as the water flow rate, tower height, air velocity in the tower etc the computer calculated the condensing temperature for an assumed rate of heat transfer in the condenser and cooling tower. Once the condensing temperature was known it was possible to calculate the power required by the compressor and hence a new value for the heat transfer rate was obtained. If this differed from the originally assumed value the calculation was repeated until the two values agreed. The computer then calculated the power required by the condenser water pumps and by the fan to pass the upcast air through the tower.

The method of calculating the performances of the various system components is given below.

Compressor Power Requirements

The main factors affecting the power required by a cooling plant compressor are the condensing and evaporating temperatures, which determine the pressure range over which the compressor has to operate, the duty, which determines the flow rate of refrigerant in the circuit, and the compressor efficiency.

Compressor power requirements were determined by analysing the performance on an ideal cooling cycle using the pressure enthalpy diagram. It was assumed that no subcooling of the condensed liquid took place in the condenser, that no superheating of the vapour occurred in the evaporator, and that there were no pressure losses in the circuit. Performance figures were determined for

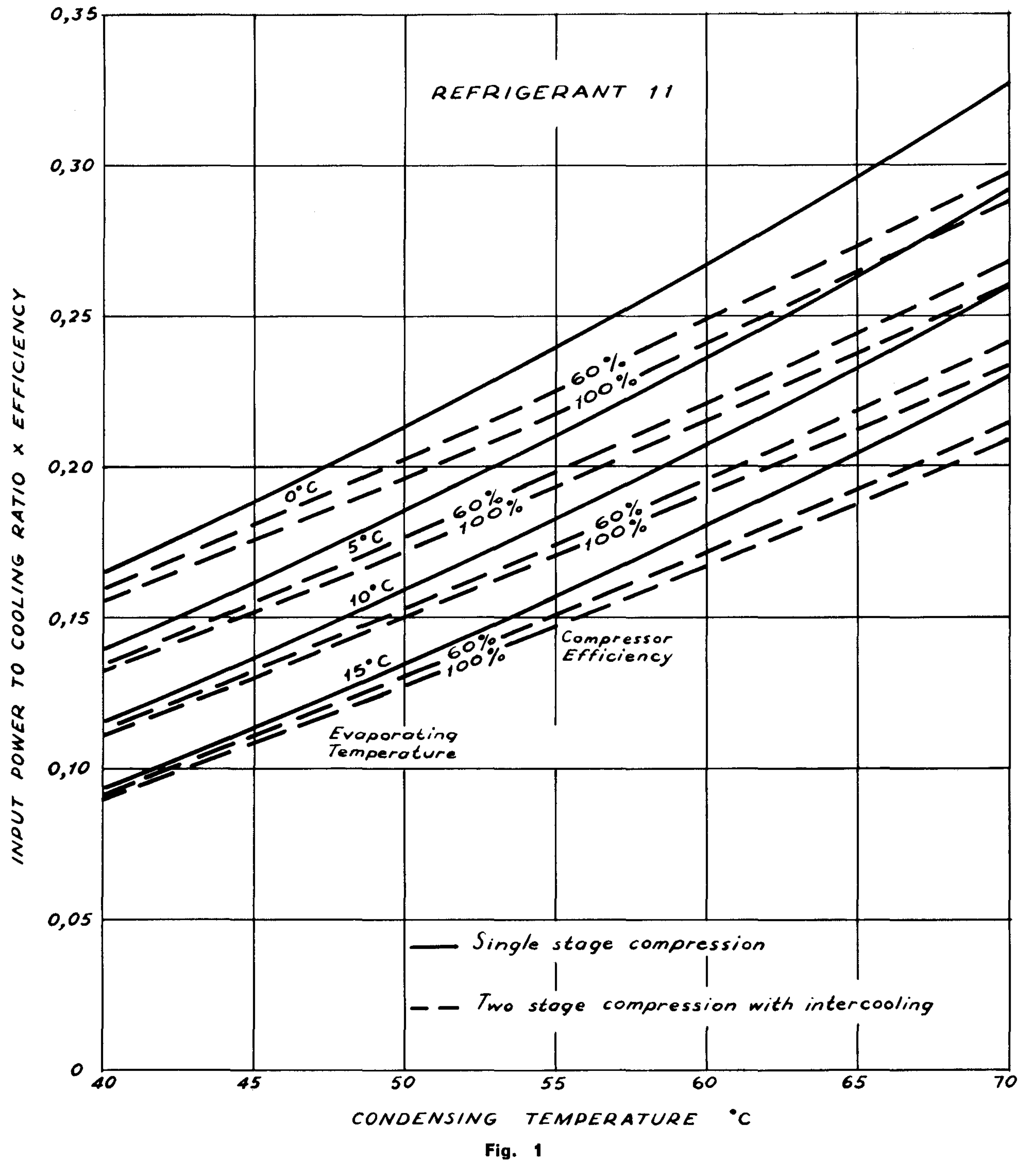


Fig. 1

a single-stage machine using both refrigerants 11 and 12, and for a two-stage machine with intercooler using refrigerant 11.

In the analysis of the operation of a single-stage machine the compressor efficiency only affects the state of the refrigerant leaving the compressor, and can be allowed for by dividing the input power to cooling ratio for the ideal compressor by the actual compressor efficiency. The same does not apply in a two-stage machine with intercooler, for here the efficiency of the first compressor stage affects the state of the refrigerant entering the second stage. Fig. 1 shows the results obtained for refrigerant 11. The benefits of intercooling, and the small effect which compressor efficiency has upon the product of the input power to cooling ratio and the compressor efficiency can be seen.

Fig. 2 compares the performance of refrigerant 11 in both single-and two-stage machines with the performance of a single-stage machine using refrigerant 12. The power to cooling ratio for the Carnot cycle is also shown. It is obvious that from this particular point of view refrigerant 11 is a more suitable refrigerant than

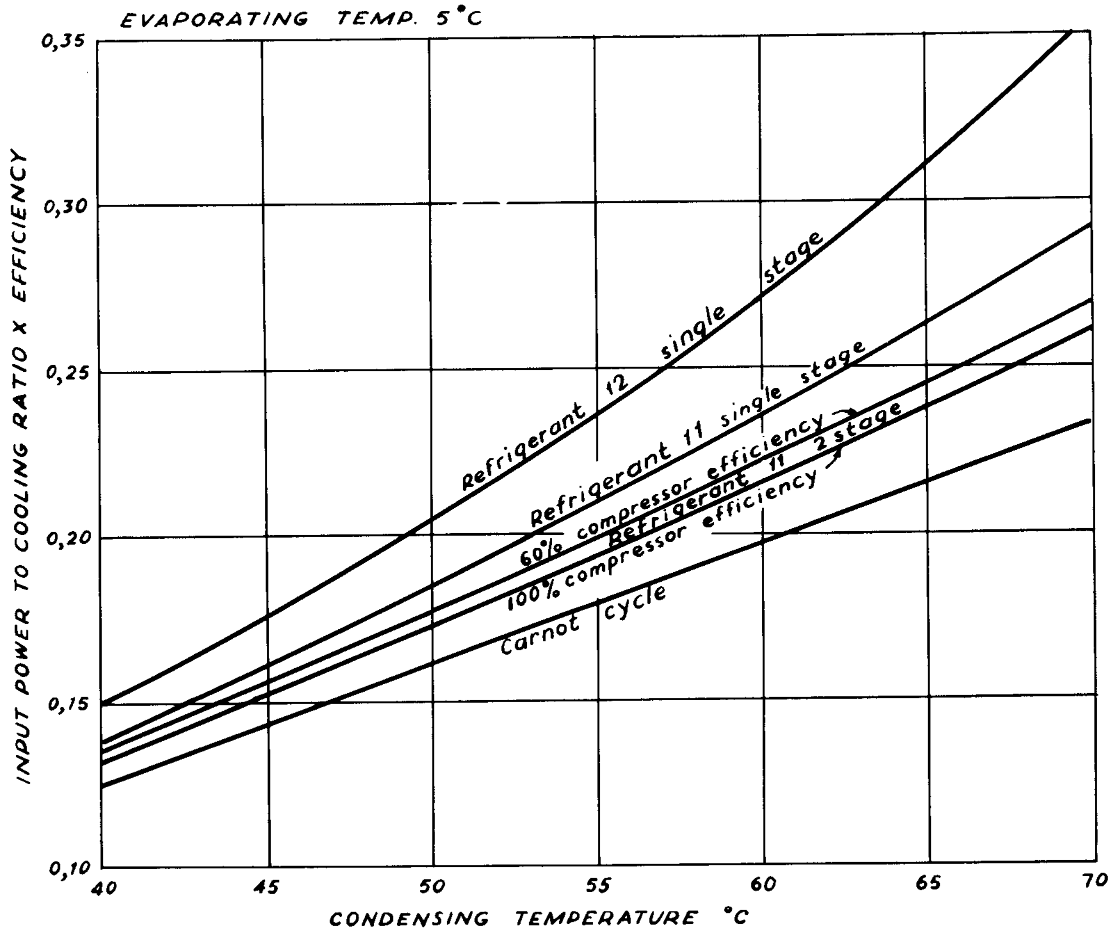


Fig. 2

refrigerant 12, particularly at high condensing temperatures.

For use in the final computer programme a simple quadratic equation was fitted to the calculated results for a two-stage machine with intercooling using refrigerant 11. A compressor efficiency of 80% was assumed and the evaporating temperature was taken as 5 °C.

Heat Transfer in the Condensor

Heat transfer in the condenser of the cooling plant can be determined from the following three equations:

$$q = U_i A_i \triangle t_1$$

$$\triangle t_1 = \frac{t_o - t_i}{\ln \frac{tc - ti}{tc - to}}$$

$$\frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{1}{h_f A_i} + \frac{x}{kAm} + \frac{1}{h_o A_o}$$

Once the overall heat transfer coefficient U_i has been calculated from the third equation, the condensing temperature t_c can be obtained in terms of the inlet and outlet water temperatures t_i and t_o by manipulation of the first two equations.

The inside heat transfer coefficient h_i depends upon the water velocity in the tubes, the mean water temperature, and the inside diameter of the tubes. McAdams[1] has given the following equation to calculate h_i.

$$h_i = \frac{5\,680\,(1+0{,}015\, t_m)\, v^{0,8}}{d^{0,2}}$$

The water velocity in the tubes depends upon the water flow rate, the total number of tubes, the tube diameter and the number of passes used in the condenser, and can easily be calculated once these factors are known.

The fouling factor $(\frac{1}{h_f})$ makes allowance for the build up of scale and dirt on the inside surface of the tubes. There has been very little published work on the build of scale in condensers operating underground, but the following figures would seem to be reasonable

Clean	0,0001 m² °C/W
Dirty	0,0002 ”
Very Dirty	0,0005 ”

The term dealing with conduction through the tubes is straightforward and does in fact have very little effect upon the overall heat transfer coefficient, but has nevertheless been included. The value used for the outside heat transfer coefficient was taken as 1 400 W/m² °C, and was assumed to be constant.

The one variable for which no value has as yet been assigned is the number of water passes in the condenser. The effect of increasing the number of passes is to increase the water velocity in the tubes. This results in an improvement in the overall heat transfer coefficient (due to an increase in the inside heat transfer coefficient) which leads in turn to a reduction in the condensing temperature, and consequently a reduction in the compressor power required. The pressure loss across the condenser depends upon both the number of passes and the water velocity, and has the following form

$P = CNv^a$

Thus increasing the number of passes, while leading to a reduction in compressor power consumption, causes an increase in pump power consumption and the optimum number of passes occurs when the sum of compressor power and that portion of the pump power required to pass water through the condenser is a minimum. The computer programme was arranged so as to select the number of passes which led to the minimum power consumption.

The numerical values chosen for the condenser are given below and were taken from manufacturer's specifications for a particular cooling plant, and manufacturer's test data was used to obtain values for the pressure loss coefficients C and a, and, the outside heat transfer coefficient h_o. With these sizes approximately four hundred condenser tubes are required per 1 000 kW of cooling.

Tube internal dia	$d = 13{,}9$ mm
surface area per tube:	
inside	$A_i = 0{,}216$ m^2
mean	$Am = 0{,}231$ m^2
outside	$A_o = 0{,}784$ m^2
tube wall thickness	$x = 1$ mm
tube conductivity (90/10 cupronickel)	$k = 44$ W/m °C
outside heat transfer coefficient	$h_o = 1\,400$ W/m^2 °C
pressure loss per pass	$P = 60\ v^{1,7}$ mbar

Cooling Tower Performance

Two papers have been published giving details of methods of calculating cooling tower performance when such variables as air and water flow rates, air velocity and tower height are known.

The first of these papers was by Whillier[2], and here information is initially required on the amount of heat to be transferred in the tower, the quantity and temperature of the air entering the tower, the air velocity in the tower and the water flow rate. Once these are known an iterative procedure is used to determine the temperature of the water leaving the tower. An exact method was used to calculate the reference water to air ratio. This method assumed that make-up water was added to the system at the temperature of the water leaving the tower and that the air leaving the tower was saturated. A fixed tower height of 17 m was used and it was assumed that three screens were installed in the tower.

The second paper describes tests on a cooling tower at City Deep[3], and here values of the air quantity and velocity through the tower, the water flow rate and amount of heat to be transferred in the tower, and the height of the tower, lead directly to the difference between the enthalpy of saturated air at the temperature of the water leaving the tower and the enthalpy of the air entering the tower. Once this enthalpy difference is known it is possible to calculate the temperature of the water leaving the tower. Unfortunately the series of tests used to obtain the performance equation did not include tests at low air flow rates and high water flow rates, and under these conditions the equation leads to the entering water temperature being lower than the wet bulb temperature of the air leaving the tower. This is obviously impossible, and when it occurred it was assumed that the water entered the tower at the same temperature as the air leaves the tower. This assumption is most probably incorrect and will result in some underestimation of the water temperatures.

In all cases the water pressure at the spray nozzles was assumed to be 0,5 bar.

Air Pressure Loss through the Tower

There is very little published information on the pressure loss occurring across cooling towers, but this is an important consideration, as the power used by the fans in passing air through a tower is not insignificant. The method used here follows a contribution by van der Lingen[4] to a paper by Lambrechts.

It was assumed that the total pressure drop across the tower has two components. The first component is the pressure drop occurring when no water flows in the tower, while the second component is the pressure drop due to the presence of water in the tower. The following equation was used

$$P = 0{,}01\,\left(RQ^2 + \frac{HLg}{(v_T - v)}\right) \quad v_T \text{ was taken as } 8{,}6 \text{ m/s}$$

Whillier[2] has published several sets of measurements of pressure loss in cooling towers, and the above equation was used to analyse the data. The procedure used was first to calculate the resistance of the dry tower by substitution of values in the equation, and then to use this resistance in predicting the total pressure loss. The results are given in Fig. 3, where it can be seen that this method provides a reasonable estimate in some cases

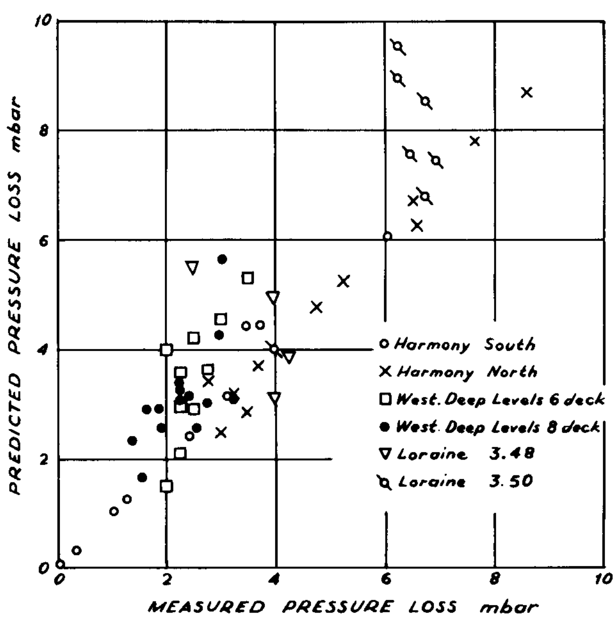


Fig. 3

only. This is one aspect of cooling tower operation which should receive more attention, particularly as the pressure loss across a tower could be expected to yield some information on the distribution of air and water within the tower.

In the final computer programme the pressure loss component for the dry tower (RQ^2) was ignored.

GENERAL

The final computer programmes involved putting together the various component parts mentioned above, together with a set of standard psychrometric equations. In these equations the vapour pressure was calculated from the equation given by Goff and Gratch[5], which was mentioned in a recent paper by Salamon[6].

Two programmes were prepared, one using the tower performance equation proposed by Whillier and the second using the tower performance equation developed from the City Deep tower.

The following input was required

Condenser

Number of tubes, fouling factor and water flow rate

Upcast Air

Quantity, wet and dry bulb temperatures, barometric pressure

Cooling Tower

Air velocity, height (constant for the Whillier equations, but variable for the City Deep equations), screen factor (Whillier equations only.)

Efficiencies

Compressor efficiency, overall pump efficiency, overall fan efficiency, and compressor motor efficiency.

In using the two programmes most of the above factors were taken to be constant, the only variables being air quantity, air velocity, water flow rate and height (City Deep equations only.)

The factors taken as constant were given the following values:

Condenser

Number of tubes	400
fouling factor	0,0003 m² °C/W

Upcast Air

Wet bulb temperature	33 °C
Dry bulb temperature	38 °C
Barometric pressure	1 200 mbar

Cooling Tower (Whillier equations only)

Height	17 m
Screen factor	1,02

Efficiencies

Compressor	80%
Pump	70%
Fan	75%
Compressor motor	95%

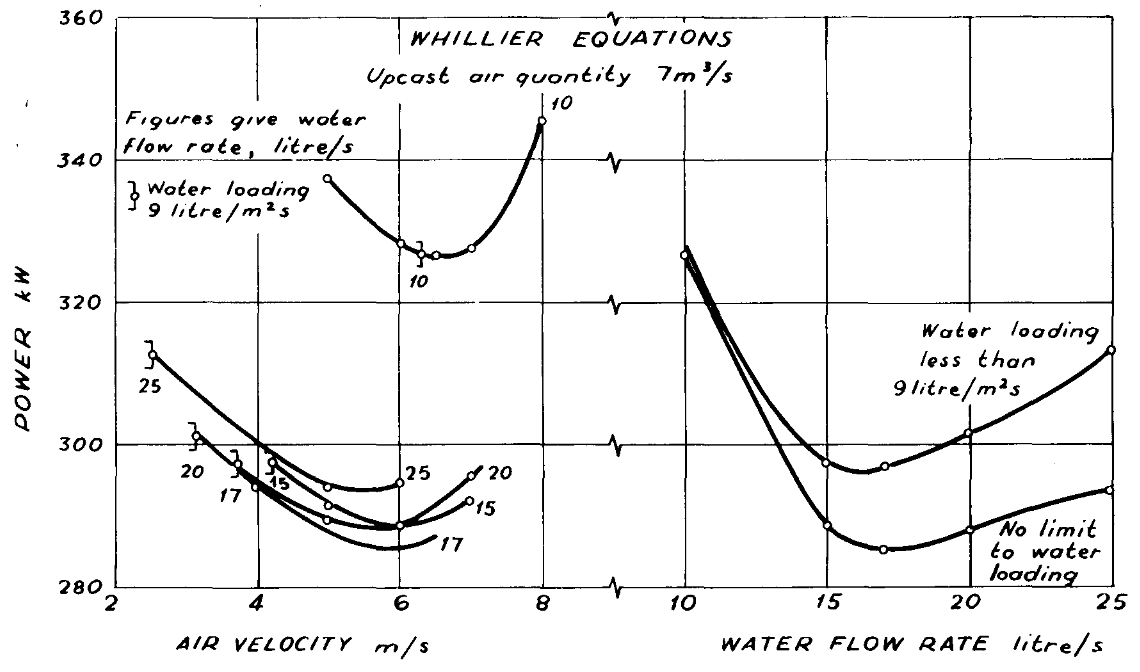
A cooling duty of 1 000 kW was used.

DETERMINATION OF OPTIMUM CONDITIONS

The two programmes were used to determine optimum values of the water flow rate, air velocity and height (City Deep equations only) for several assumed values of upcast air quantities. The optimization was done only in terms of the total power consumed and no account was taken of variations in capital costs. The method used was basically one of trial and error. Fig. 4 gives an example of the graphs which were plotted from the computer results and it is obvious how the optimum conditions were determined.

Fig. 5 shows the final results obtained from the programme using the Whillier equations. The top graph shows the variation in total power requirements with upcast air quantity, while the bottom graph shows how the optimum values of air velocity, water flow rate and condensing temperature vary with air quantity. This bottom graph shows that the optimum values of water flow rate and air velocity vary considerably as the quantity of air available for heat rejection varies. The optimum water flow rate varies from 15 l/s at the lowest air quantity to 50 l/s at the highest air quantity. The lowest air quantity of 5 m³/s is very low indeed, and is perhaps $\frac{1}{3}$ to $\frac{1}{2}$ of the lowest air quantity presently used underground for heat rejection, while the highest air quantity of 80 m³/s is very high. The optimum air velocity increases as the air quantity drops, varying from 3,5 m/s at 80 m³/s to 6 m/s at 5 m³/s. These last figures (an air velocity of 6 m/s when the air quantity is 5 m³/s and the water flow rate 15 l/s) lead to a water loading of 18 l/m²s, which is twice the maximum recommended by Whillier. This perhaps indicates a field for further investigation into cooling tower performance, for the performance equations do indicate better performance at high water loadings. It is interesting to see that the cooling tower for an air quantity of 5 m³/s would be just over 2,2 m in diameter, if the water loading is not limited, when the plant duty is 5 000 kW.

As would be expected the results show that the condensing temperature increases as the air quantity is reduced, but it is interesting to see that even with an air quantity of 5 m³/s and with the water loading limited to 9 l/m²s, the condensing temperature is just over 65 °C, which is surprisingly low. As far as total power consumption is concerned the reduction of the upcast air quantity from 80 to 5 m³/s causes the total power requirement to increase by only 40% if the water loading is limited, and to increase by about 30% if high water loadings are used. This somewhat surprising result is due to the extreme non-linearity of the vapour


WHILLIER EQUATIONS
Upcast air quantity 7m³/s
Figures give water flow rate, litre/s
Water loading 9 litre/m²s
Water loading less than 9 litre/m²s
No limit to water loading
POWER kW
AIR VELOCITY m/s
WATER FLOW RATE litre/s

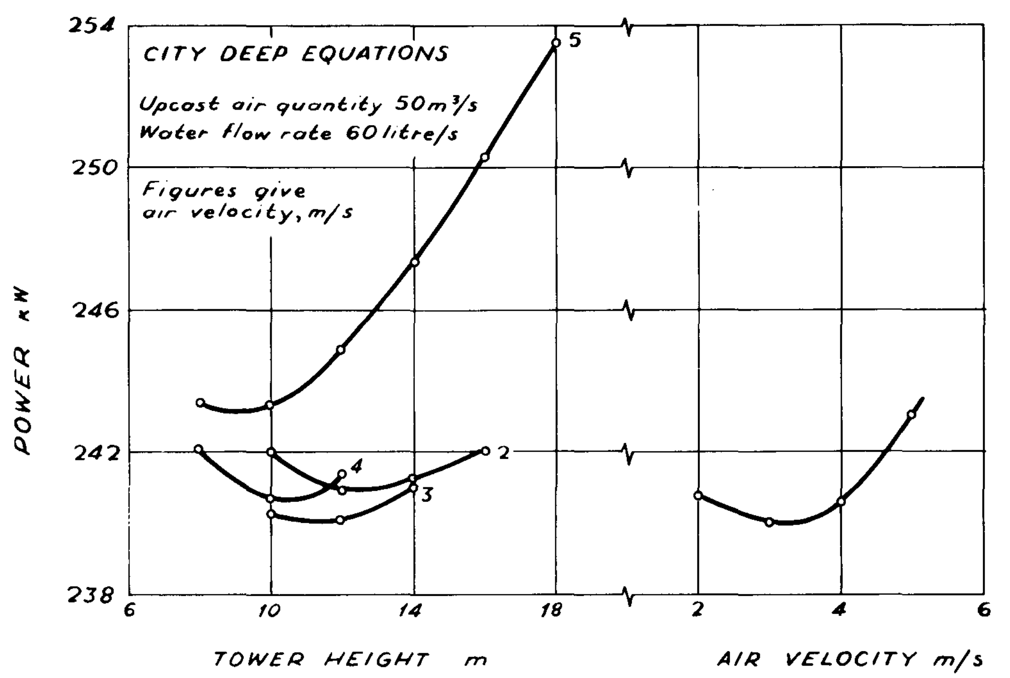
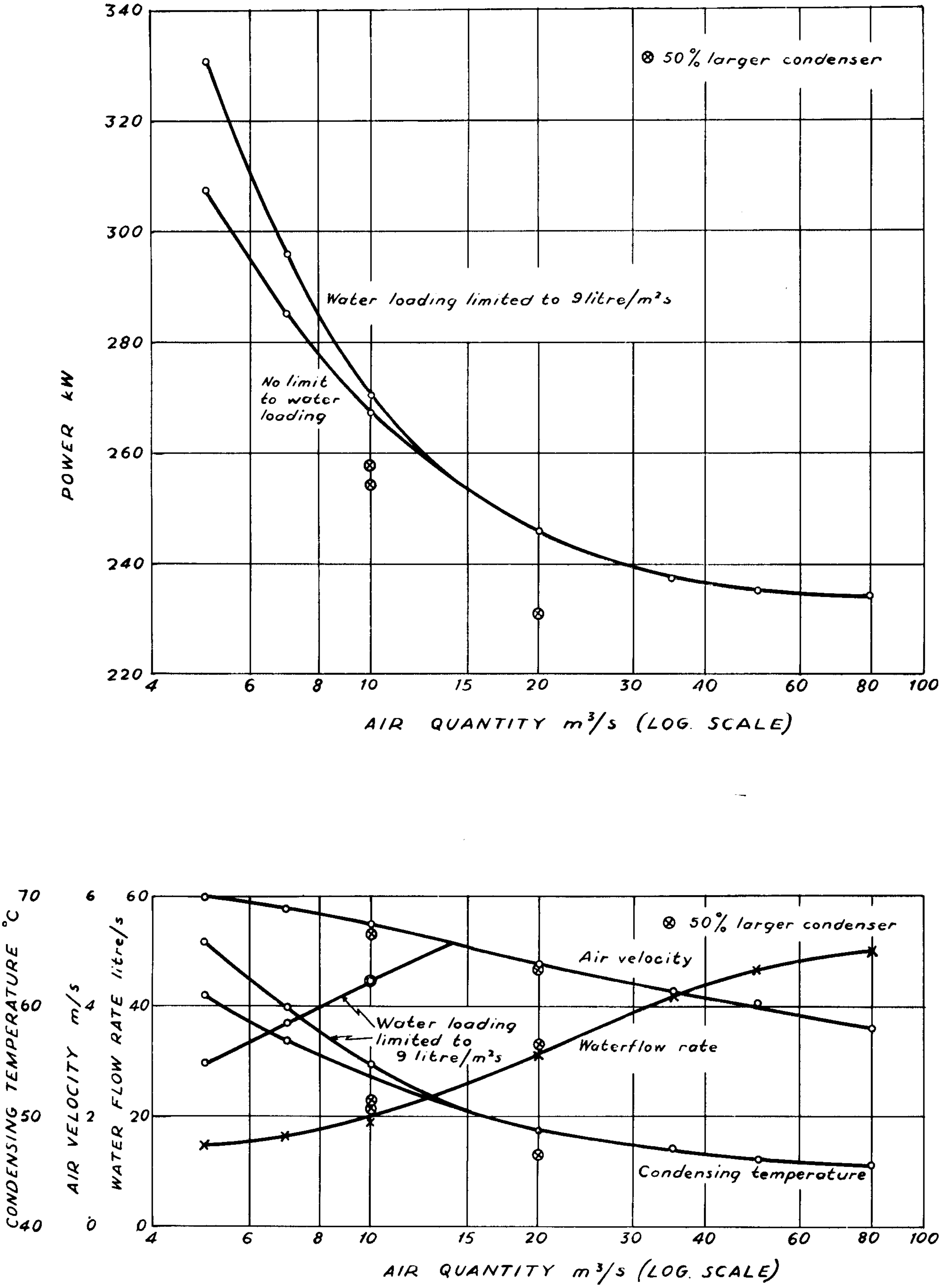

CITY DEEP EQUATIONS
Upcast air quantity 50m³/s
Water flow rate 60 litre/s
Figures give air velocity, m/s
POWER kW
TOWER HEIGHT m
AIR VELOCITY m/s

Fig. 4

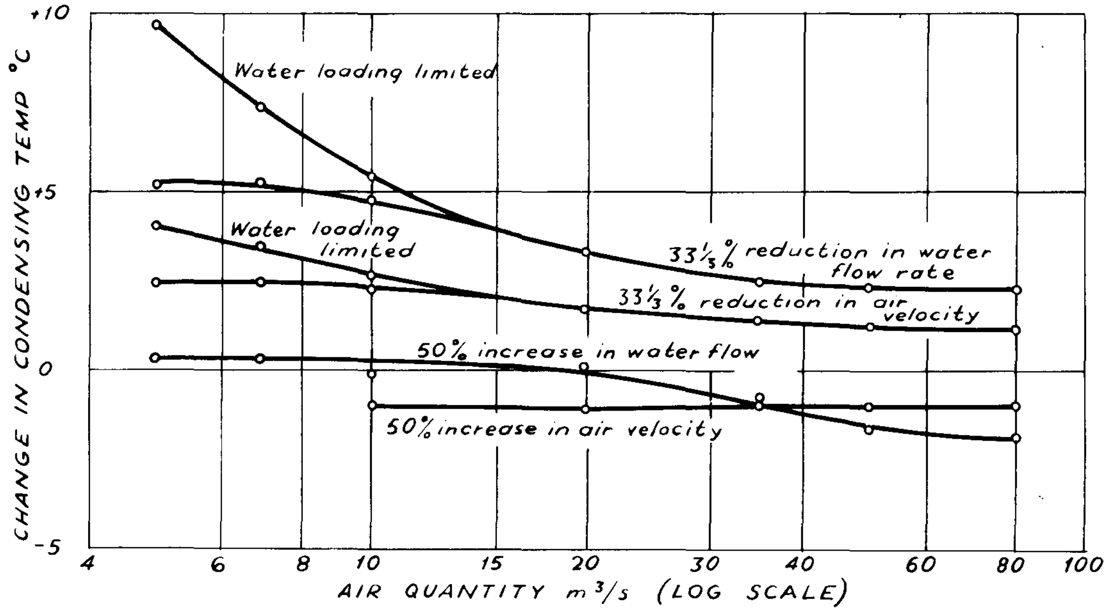


⊗ 50% larger condenser
Water loading limited to 9 litre/m²s
No limit to water loading
POWER kW
340
320
300
280
260
240
220
4
6
8
10
15
20
30
40
60
80
100
AIR QUANTITY m³/s (LOG. SCALE)
⊗ 50% larger condenser
Air velocity
Water loading limited to 9 litre/m²s
Waterflow rate
Condensing temperature
CONDENSING TEMPERATURE °C
AIR VELOCITY m/s
WATER FLOW RATE litre/s
70
60
50
40
6
4
2
0
60
40
20
0
AIR QUANTITY m³/s (LOG. SCALE)

Fig. 5

pressure-temperature relationship for water. The extent of this non-linearity is perhaps not fully appreciated. Table I gives the increase in sigma heat of saturated air for temperature changes of 1 °C at various temperatures.

It is worth looking at the effect on cooling plant operation of departure from optimum conditions. Fig. 6 shows the results of a series of calculations where either the water flow rate or the air velocity were increased by 50% or reduced by 33⅓% without changing any of the other values. The first graph shows the effect of these changes on the condensing temperature. It can be seen that it is very difficult to bring about any subtantial decrease in the condensing temperature at any one air quantity. Increases in either water flow rate or air velocity only produced temperature reductions of about 1 °C or 2 °C; indeed at the lower air quantities an increase in water flow rate actually causes a slight increase in the condensing temperature. The condensing temperature is more sensitive to reductions in air velocity and water flow rate, particularly to a reduction in the water flow rate, which can cause a substantial increase in the condensing temperature. On the other hand power consumption, which is shown in the second graph, is most affected by an increase in the air velocity or reduction in the water flow rate. The very sharp increase in power consumption which occurs when the air velocity is increased is a result of the form of the equation



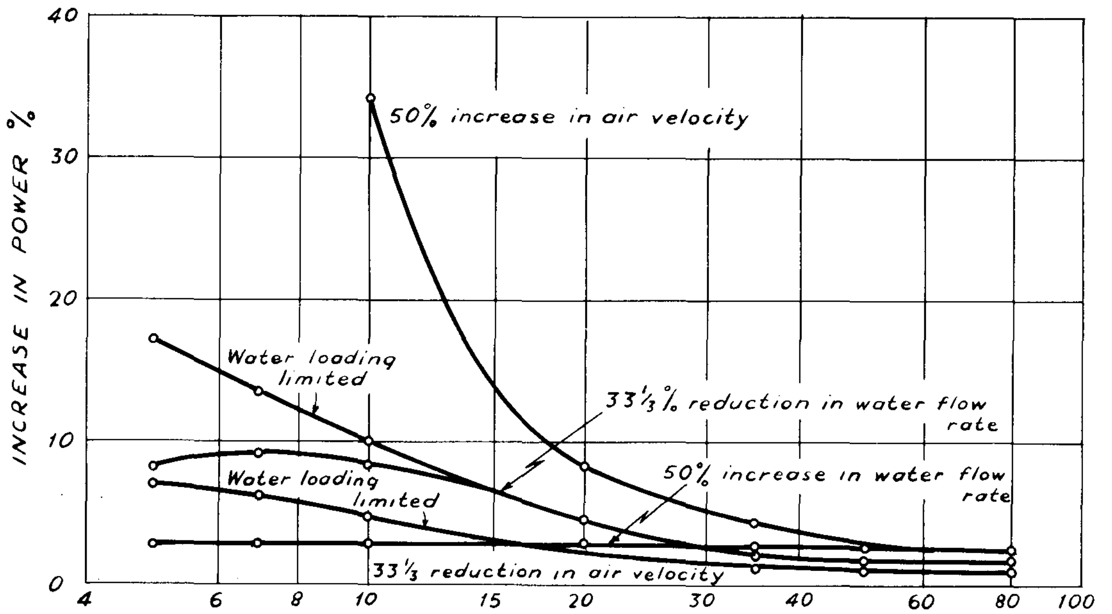


Fig. 6

TABLE I

COPPER OXIDE SOLUBILITIES IN LIME- AND SODA-BOROSILICATE SLAGS

Slag Composition			Liquidus temperature °C ± 10 °C	Copper oxide solubility as percentage of total copper oxide content					
				1 150 °C			1 250 °C		
CaO %	B_2O_3 %	SiO_2 %		5 % CuO	10 % CuO	15 % CuO	5 % CuO	10 % CuO	15 % CuO
56,2	19,5	24,4	1 130	100	70	50	100	90	60
41.2	36,0	22,8	1 020	100	90	60	100	100	100
34,8	35,7	29,5	980	100	90	60	100	100	100
Na_2O %									
10	60	30	670	100	100	100	100	100	100
10	54	36	700	100	100	80	100	100	80
20	48	32	620	100	80	55	100	90	60
40	36	24	820	100	60	40	100	75	50
10	45	45	820	90	80	50	100	90	60
10	35,6	54,4	870	80	80	40	100	70	40

chosen to calculate the pressure loss in the cooling tower. As the air velocity approaches the critical velocity of 8,6 m/s so the pressure drop, and hence the fan power tends to infinity.

Fig. 5 also shows the effect of using a larger condenser (600 tubes instead of 400 tubes.) This change results in a reduction in both power consumed and condensing temperature, but does not change significantly the optimum values of water flow rate or air velocity.

Fig. 7 and 8 show the results of similar calculations with the programme using the City Deep cooling tower performance equations. The power consumption curves show a very similar trend to that obtained from the Whillier equations, except that the increase in power consumption as the upcast quantity is reduced is not nearly as marked. This is no doubt due to the shortcomings of the City Deep equation mentioned earlier. Optimum values differ considerably from those obtained from the Whillier equations, the air velocity being lower and the water flow rate higher. Optimum tower heights vary from 11 m at an air quantity of 50 m^3/s to 15 to 20 m at an air quantity of 17 m^3/s. The effects of departures from optimum conditions are shown in Fig. 8. Once again the results are similar to those obtained from the Whillier equation, except that, as the optimum air velocity is now lower than before, an increase in air velocity does not produce the steep rise in power consumption obtained previously. Variation in the height of the tower does not cause a particularly large change in either condensing temperature or power consumption. . . . .

The optimum water velocity in the condenser tubes was found to vary considerably. This occurred because there can obviously only be integral numbers of passes.

The optimum velocity varied from 1,4m/s to 2,4 m/s, with a mean value of about 1,8 m/s.

DISCUSSION

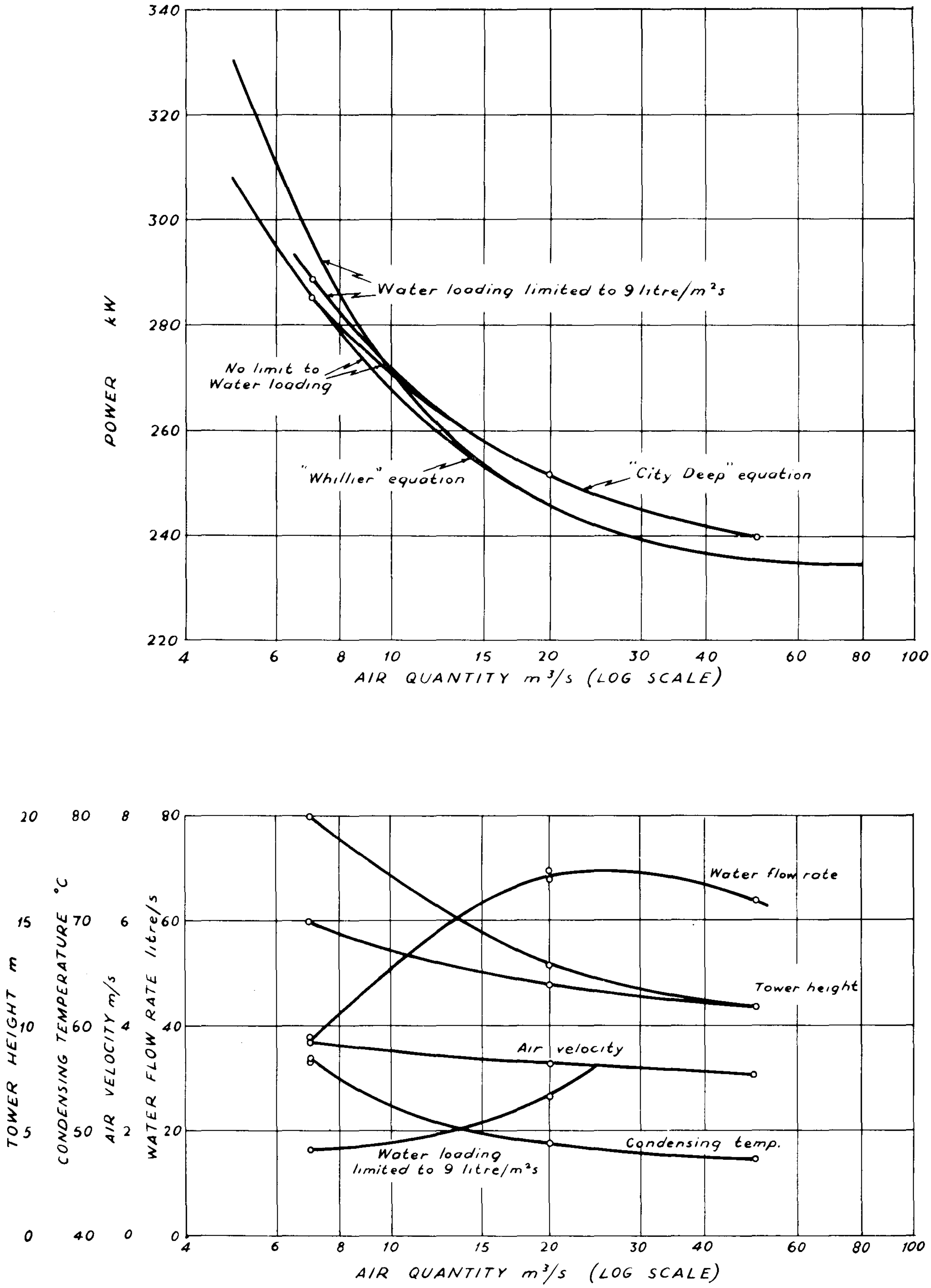
While a considerable portion of this paper has been devoted to the presentation and discussion of the results of the optimization studies it is not claimed that these results can be generally applied. There are several reasons for this.

The first is that in carrying out the optimization the only criterion applied was that the power consumption should have a minimum value. No account was taken of the effect of variations in the various parameters on capital costs. Seeing that in most cases the optima were not very critical, i.e. fairly large deviations from optimum conditions do not greatly increase the power consumption, it can be expected that even fairly small variations in capital cost will affect the optimum conditions obtained.

A second reason is that the work was confined to one particular type of cooling plant, a two-stage machine with intercooling using refrigerant 11 with an evaporating temperature of 5 °C and a compressor efficiency of 80 %. Any change, for example in compressor efficiency or in the refrigerant used, will change the slope of the graph of compressor power against condensing temperature and will thus again affect the optimum conditions.

The third, and perhaps most important reason, is that the reliability of this type of model study depends to a very large extent upon the reliability of the information used in formulating the model. In particular it would seem that the method of predicting the air pressure loss across the tower could do with further systematic investigation.

All work on the computer simulation of various problems should not be thought of as providing a final answer to the particular problem, but rather as providing a useful tool to aid further study of the problem.



POWER kW
340
320
300
280
260
240
220
Water loading limited to 9 litre/m²s
No limit to Water loading
"Whillier" equation
"City Deep" equation
4
6
8
10
15
20
30
40
60
80
100
AIR QUANTITY m³/s (LOG SCALE)
TOWER HEIGHT m
CONDENSING TEMPERATURE °C
AIR VELOCITY m/s
WATER FLOW RATE litre/s
20 80 8 80
15 70 6 60
10 60 4 40
5 50 2 20
0 40 0 0
Water flow rate
Tower height
Air velocity
Condensing temp.
Water loading limited to 9 litre/m²s
4
6
8
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15
20
30
40
60
80
100
AIR QUANTITY m³/s (LOG SCALE)

Fig. 7

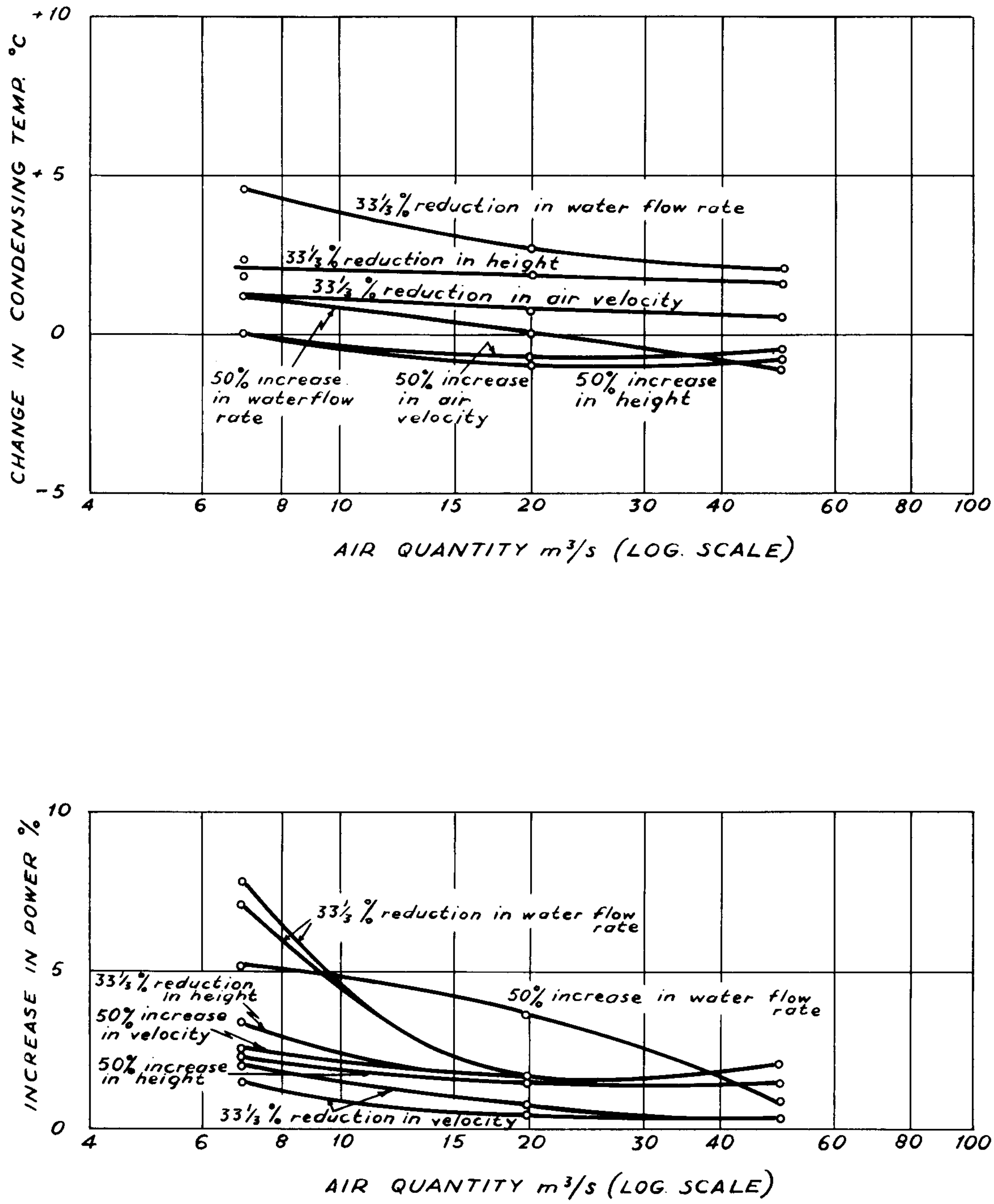

Fig. 8

This work is no exception, and it is felt that it has demonstrated the feasibility of applying this type of approach to the solution of one particular cooling plant problem and pointed the way to further applications.

To demonstrate how the computer simulation of this problem can be used as a tool, mention need only be made of the following:

(i) the recognition of the importance of information on the air pressure drop across cooling towers.

(ii) seeing that the results of the optimization studies indicate that at low upcast air quantities operation of the cooling tower at very high water loadings leads to lower overall power consumption, it would seem that further work is required on the effect of high water loading on cooling tower performance.

(iii) operation of underground cooling plants with small quantities of air available for heat rejection does not appear to lead to excessively high condensing temperatures. It would seem to be quite possible to operate a cooling plant with an upcast air quantity of about 7 m³/s per 1 000 kW of cooling.

Mention was made earlier of the further application of computer methods to the study of other aspects of cooling plant operation. The design of chilled water reticulation systems is one application, while another would be the prediction of the performance of existing cooling plants when subjected to various conditions.

SYMBOLS AND UNITS USED

A_i	inside tube area (m^2)
Am	mean tube area (m^2)
A_o	outside tube area (m^2)
a	constant
c	constant
d	inside tube diameter (mm)
g	acceleration due to gravity (9,81 m/s^2)
H	tower height (m)
h_f	reciprocal of the fouling factor (W/m^2 °C)
hi	inside heat transfer coefficient (W/m^2 °C)
h_o	outside heat transfer coefficient (W/m °C)
k	tube conductivity (W/m °C)
L	water loading ($1/m^2$s)
N	number of condenser passes
P	pressure loss (mbar)
Q	air quantity (m^3/s)
q	heat transfer rate (W)
R	resistance (Ns^2/m^8)
t_c	condensing temperature (°C)
t_i	inlet water temperature (°C)
tm	mean water temperature (°C)
t_o	delivery water temperature (°C)
U_i	overall heat transfer coefficient (W/m^2 °C)
v	water, air velocity (m/s)
v_T	terminal velocity (m/s)
x	tube wall thickness (m)
$\triangle t_1$	logarithmic mean temperature difference (°C)

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