ON THE DESIGN AND CONSTRUCTION OF UP-TO-DATE HEATING POWER STATIONS

COMBINATION OF DISTRICT HEATING AND MEETING ELECTRIC PEAK LOADS

By

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The economic advantages of disctrict heating which up to the recent decades were rather significant, are gradually decreasing, due primarily to the fact that they resulted mainly from the savings in fuel through the low specific heat consumption of electric power generated in back-pressure power plants, which is steadily dwindling. While, namely, the specific heat consumption of electric power produced in back-pressure plants (which at present moves around 1200 kcal/kWh) cannot be reduced below 1050 to 1100 kcal/kWh, the specific heat consumption of electric power generated in condensing stations, which only one or two decades ago was in the neighbourhood of 3 to 4000 kcal/kWh, today approximates the 2000 kcal/kWh figure. Savings, accordingly, from the former 1800 to 2800 kcal/kWh have dropped to 800 kcal/kWh. Alongside this considerable fall in the achievable savings, with the very low utilization period of district heating installations under normal climatic conditions, amortisation of the capital invested is no longer possible. In addition capital costs are increasing, more particularly if the district heating network is established in built-up areas where the laying of lines is becoming more and more difficult and costly.

The daily electric load diagrams, on the other hand, show increasing peaks of load and this fact — again due to the short utilization time — imperatively calls for inexpensive peak load stations.

Under these circumstances, the idea to seek for the solution of both problems in a combined power plant which supplies district heating and meets peak loads, seems obvious. In this case the district heating power station would be of a design which is capable of supplying heat and electric power for peak loads at the same time, and a considerable portion of the first costs could be charged against the peak load supply, reducing thereby investment costs charging the district heating to such a degree that it would continue to remain economical even with considerably lower savings in fuel.

For such combined power supply, a gas turbine plant adjusted to supply heat, provides excellent facilities.

In addition to the conventional stationary gas turbine plants, also gas turbines generating gas by aeroplane jet propulsion engines can be used for the purpose to advantage. In these plants the aeroplane jet propulsion engine functions as the gas generator and the gas supplied at high pressure and high temperature is expanded in a so-called working turbine, coupled to the electric generator. Such plants are increasingly used to meet peak loads, and the experience gained with them is satisfactory in every respect. In what follows, therefore, we shall examine gas turbine plants of both types — using stationary and aircraft propulsion gear — as means for the said combined supply of energy.

The gas turbine cycle is eminently suitable for the simultaneous supply of electric and thermal energy, for two reasons. On the one hand, the temperature of the expanded fluegases leaving the turbine is still high, and their heat content can be readily utilized to warm up the circulating heating water, in heat exchangers of relatively simple structure and requiring relatively small expenditure. (In equipments which are considerably cheaper than those in which heat content of the fluegases is utilized for warming up compressed air, as is usual in the conventional gas turbine plants.) At the same time, to meet the peak loads — both thermal and electric — a simple facility is available in the fact that the high oxygen content of the expanded fluegases (moving between 15 and 19 per cent, depending on the load) can be utilized as the combustion air of an appropriately designed supplementary firing equipment.

Below we are going to describe the layout of the dual-purpose gas turbine cycle, with both aircraft propulsion engine and with conventional stationary plant.

I

Combined district heating gas turbine plant with aircraft propulsion engine

In our calculations, we relied on the data available for the Avon 1533A gas generator of the Rolls Royce Co. The heat cycle diagram is shown in Fig. 1.

The fluegases produced in the gas generator (RRGG), at high temperatures and under pressure, are passed into the working gas turbine (MGT) for expansion. When leaving the working turbine, the fluegases, still having a considerable heat content, are sent through the supplementary combustion chamber (E) to the waste-heat boiler (HK) in which they cool down and exhaust to the atmosphere through the stack. The waste-heat boiler is so arranged that the heat transfer surface destined to warm up the heating water, is accommodated in its steam space. (The supplementary combustion chamber is separately marked in the diagram; in the actual layout it must be evolved as the combustor of the waste-heat boiler.)

A plant designed according to the cycle diagram in Figure 1 can be operated under practically all conceivable working conditions. The steam produced in the boiler may be used, in its full quantity, to warm up the water of the district-heating system (T) to the temperature determined by the momentary ambient temperature, or may be passed through the condensing steam turbine (GT) in which, expanded down to condenser pressure, it generates electric power. As will be seen later, the system can be operated also in such a way that part of the steam generated in the waste-heat boiler is used to warm up the water for the district-heating system and the rest is allowed to expand through the steam turbine. Each of the three cases enables to cover incidental extra energy demand, by the combustion of additional fuel in the supplementary firing equipment.

To supply the maximum possible amount of electric power, the entire energy is passed to the steam turbine and the distric-heating is shut down completely for a certain period, which naturally depends on the storage capacity of the district heating system, respectively on the maximum permissible interruption of heating.

Accordingly, the combined system can be used as the peak load reserve of the electric power grid even in the peak heating period, and can function as a full-value peak load plant throughout the period of operation, in addition to supplying heat for the district heating system.

In the above outlined extensive range of possible working conditions, economical control of the plant can be achieved by relatively simple means.

According to the charts made available by the Rolls Royce Co., control of the gas generator is not advantageous since under conditions deviating from the optimal, the specific consumption of the plant, respectively, the quantity and quality of compressed fluegas it supplies, is greatly deteriorating. Therefore, at the beginning and at the end of heating period — viz. when the heating demand is so low that the self costs of electric power produced in the suggested system would exceed the value at which the grid would require it — the waste-heat boiler is operated solely by the supplementary heating — viz. without the gas generator and the working turbine.

When heating demand rises to a level which makes the operation of the gas turbine economical (considering naturally also the power supplied by the steam turbine), the entire equipment is put in operation and the control must ensure that the portion of steam produced in the waste-heat boiler, in excess of the quantity needed for heating, should reach the steam turbine.

For starting condition we assumed furthermore that at an ambient temperature of $-15°C$, the equipment must use the supplementary firing to meet the maximum heating load, while without the latter — by utilizing merely the heat of the fluegas leaving the working turbine — it would supply heat only up to an ambient temperature obtained as an optimum from the frequency curve of the ambient air temperatures. (This value has been taken at $-2°C$ in accordance with Hungarian practice, but it must be determined separately for each geographic location by optimalisation.) At the limit temperature of $-2°C$ — provided that the plant operates without the supplementary firing — the total quantity of steam generated in the waste-heat boiler will be condensed by warming up the heating water, and no steam is left over to drive the steam turbine, which is shut down in this period.

Another basic assumption had been a minimum temperature of 130°C in the waste-heat boiler, to prevent corrosion by fuels containing sulphur. The

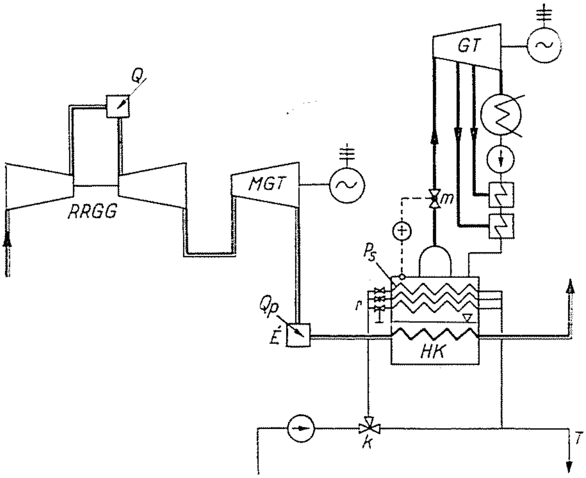

Fig. 1

minimum steam pressure in the boiler will accordingly be 2.75 at. (Later a 25 to 30°C difference of terminal temperatures was calculated assuming the temperature of exhaust fluegas to be 155 to 160°C.) At the same time, the heating surface accommodated in the steam space of the boiler, had been so dimensioned that at a steam pressure of 2.75 at. it should warm up the heating water as required for an ambient temperature of $-2°C$. (In district heating system of 130/70°C, heating water at $-2°C$ ambient temperature would be heated up from 60 to 98°C.)

The maximum steam pressure for the waste-heat boiler, so as to gain maximum efficiency from the steam turbine, must be determined by optimization, in consideration of the thermal and structural factors. Our optimum calculation carried out for the case in hand — as will be dealt with later in more detail — yielded a steam pressure of 10 at. as the optimum value of boiler pressure.

Accordingly, in the periods when the steam turbine is also in operation, a pressure of 10 at. prevails in the waste-heat boiler and a suitable regulation of the heat exchanger surfaces warming up the heating water in the boiler's steam space must, therefore, be provided. To achieve a possibly simple regulation — as the one seen in the diagram of Figure 1 — heating surfaces are divided into several parallel-connected sections, each of which can be detached separately, by means of simple stop valves (r). In Figure 4, the curve "a" indicates, in function of the momentary ambient temperature, what part (v) viz. how many sections of the total built-in water heating surface must be operated to meet the heating demand at a steam pressure of 10 at. The curve "a", which is continuous in Figure 4, is actually a staggered line with as many steps as the number of the parallel-connected surfaces and thus, the ultimate regulation of the temperature of the heating water circuit is carried out by means of the mixing valve (k) shown in the diagram.

The control of the plant, according to the actual heating and electric peak load, takes place in a simple manner. Working conditions may be divided into the following five groups:

1. Ambient temperature above —2°C

At the beginning of the heating season, when the heat demand is so low that the start-up of the gas turbine plant is not warranted, only the waste-heat boiler and the supplementary firing are operated. The specified minimum boiler pressure (2.75 at) is maintained by firing and the suitable adjustment of the water heating surface.

When heating demand attains the value determined for the respective station, the gas turbine plant is started up and the supplementary firing is shut down. At this load, heating demand does not require the full energy of the fluegases leaving the working turbine, and the pressure in the waste-heat boiler rises to $p_s = 10$ at. Attaining this value, the overflow valve (m) opens letting the surplus steam to pass into the steam turbine. At 10 at. pressure, the water heating surface must naturally be regulated according to the curve "a" of Figure 4, to adapt to the ambient temperature.

2. Ambient temperature below —2°C; no electric peak load

Since under such conditions the energy of the fluegases leaving the working turbine does no longer meet the heating demand, the supplementary firing is started up. Furthermore, the entire water heating surface is put into operation, and since the steam turbine is shut down, steam pressure required to maintain the specified heating water temperature is controlled by the supplementary firing. Steam pressure in these periods corresponds to the curve "b" of Figure 4.

3. Ambient temperature below —2°D; electric peak load arising

In such periods, a steam pressure of 10 at. is maintained in the waste-heat boiler by the supplementary firing while the size of the water heating surface in operation is controlled according to curve "a" of Figure 4, to suit the ambient temperature prevailing. Apart from the actual heating demand, the maximum output of the steam turbine in such periods is determined by the maximum capacity of the supplementary firing.

4. Ambient temperature below —2°C and maximum electric peak load

Should the grid demand electric power in such amount that even with maximum supplementary firing the load could be met by using the entire steam quantity only, heating is correspondingly reduced or, if necessary, shut down altogether, and the full steam quantity generated in the waste-heat boiler is expanded in the steam turbine. Such working conditions can be maintained until the storage capacity of the heating system permits, under the given ambient temperature. (Naturally, the station can be dimensioned also in such a way as to be capable of meeting maximum electric peak load and maximum heat load simultaneously. This solution, however, is only economical when the electric peak load considerably exceeds the value that can be met with the maximum of supplementary firing foreseen for the plant (which is required to meet the heat load without the steam turbine at a —15°C ambient temperature). Otherwise, the utilization of the storage capacity of the heating system offers economical operation under all circumstances.)

5. Ambient temperature above +12°C

Outside the heating season, the plant operates as a gas turbine power station. In the waste-heat boiler — heating being shut down completely — steam pressure is at 10 at. and the steam quantity generated is passed to the steam turbine via the overflow valve (m). Supplementary firing in such periods is operated in accordance with the required power level of the steam turbine.

It may happen, of course, that under extreme conditions, the control of the output of the gas generator may also be economical. With this facility available, the output of the plant can be controlled practically from nil to maximum, to meet both heat and electric peak demands.

As can be seen, the controlling described above enables economical operation under all practically conceivable working conditions, with the gas generator and the working turbine running at —15°C would amount to round 32 million kcal/h and the additional output to be introduced by the supplementary firing to round 12 million kcal/h. The schematic process-diagram of the working turbine is shown in Fig. 2.

Naturally, the dimensions of the supplementary firing and of the waste-heat to be introduced, can be increased at will, irrespective of the above constraints, and in the calculations herebelow, in which we wished to determine the optimum working pressure, we went into a more extensive range and varied the fluegas temperature (t_7) ahead of the waste-heat boiler as follows:

$$t_7 = 410;\ 500;\ 550;\ 600;\ 650;\ 700;\ 750;\ 800°\text{C}$$

whereby the pressure of steam generated showed the following variations:

$$p_s = 2,\ 3,\ 4,\ 5,\ 7,\ 10 \text{ and } 15 \text{ at.}$$

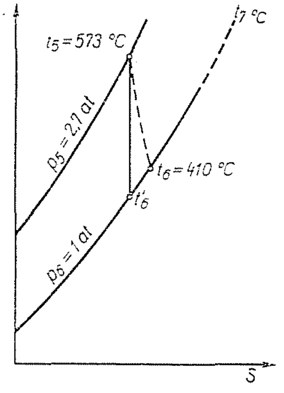

Fig. 2

To achieve a simple and inexpensive design, we have abstained from using a superheater (its insertion would be complicated anyway, since we had envisaged periods with the steam turbine standing still).

Fig. 3 shows the results of the calculation. As apparent from the figure, in the case without supplementary firing ($t_7 = 410$°C) and the entire amount of steam expanding in the steam turbine (accordingly without heating demand), the maximum output in the steam turbine would be obtained at a pressure of around $p_s = 10$ at. (on the ordinate of Chart 3 the ideal steam turbine output has been plotted, without the influence of the internal efficiency of the turbine, since the latter varies with the variations of the quantity of steam generated).

Increasing the amount of heat introduced by supplementary firing, the optimum of the curves shifts towards higher pressures. However, the flattening of the curves in the range above 10 at. and the fact that the moisture content of steam expanded from saturated state, above 10 at. live steam pressure exceeds the permissible level, justifies the selection of 10 at. steam pressure in the range which may be regarded as realistic. Particularly justifiable is this value if we bear in mind the peak load plant character of the equipment, which calls for possibly inexpensive and simple construction.

Steam turbine output can be increased at will by supplementary firing, even in the presence of heating demand. In such case the value of the incremental heat consumption of the plant (the ratio of heat introduced by supplementary firing to the addition of work obtained from the steam cycle) depends solely on the pressure prevailing in the waste-heat boiler and is independent from the reheating temperature (t_7) of the fluegases.

Namely, the thermal efficiency of the steam cycle is essentially determined by the average temperature of heat input. This temperature, in turn, depends practically on the steam pressure maintained in the waste-heat boiler.

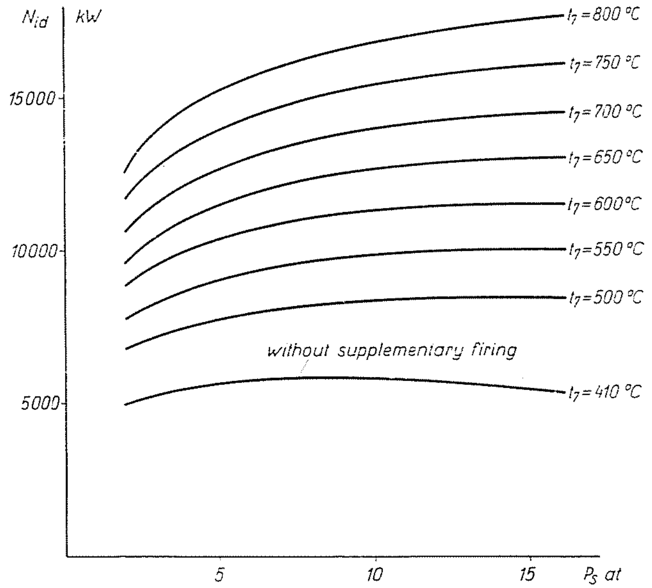

Fig. 3

With a given steam pressure, therefore, the fluegas temperature will have no influence; its value will merely indicate the magnitude of the heat quantity introduced by supplementary firing — which latter naturally determines the extra steam quantity generated. (The additional work depends naturally also on the variations in the internal efficiency which latter is a function primarily of the quantity flowing through the steam turbine.)

On the other hand, the effect of the reheating of fluegases, in case of a given boiler surface, would manifest itself only in the cooling down of the fluegases having a greater heat content, to a higher temperature, over the given heating surface. In the case in hand, namely, with the quantity of fluegas practically constant and independent from the additional heat introduced, the exit temperature and with it the exit losses will increase in the same degree as the heat introduced by supplementary firing has grown. This is so, because it is only the share of additional heat corresponding to the factor Φ of the waste-heat boiler that can be used for steam generation, while the rest $(1-\Phi)$ will be irretrievably lost.*

Accordingly, the value of the incremental heat consumption is independent in our case of the reheat temperature (t_7).

As it is obvious from the schematic chart in Figure 1, in the coupled steam cycle regenerative feedwater preheating is applied. This solution may seem unwarranted and unjustified at the first sight, since in waste-heat utilization it is generally preferable to allow the available steam quantity to expand down to condenser pressure and use the cold condensate for the further cooling down of the fluegases leaving the waste-heat boiler. This is in fact in most cases the best and most economical solution. In the case in hand, however, the regenerative feedwater preheating has to be applied for two reasons. The first is that since no economizer has been foreseen, the condensate is returned to the boiler direct and for its heating up to saturation temperature, live steam would have to be used. If, on the other hand, renegerative preheating is applied, feedwater would be preheated by bled steam — viz. steam which has done useful work already.

The second aspect is as follows: if we used economizer (for instance, in case of higher steam pressures and continuous operation of the steam turbine, viz. with continuous feed), feedwater inlet temperature may not be below 130°C, as said above, with fuels containing sulphur. This, too, calls for the use of regenerative feedwater preheating.

In our previous investigations, when prescribing the lowest exit temperature of the fluegas, we assumed the use of fuels with sulphur content. With fuels free from sulphur (for instance natural gas) this limitation is naturally obviated and the temperature of exhaust fluegases may be considerably lower, and the idea of applying an economizer may be taken into consideration. In this case, the optimization of boiler pressure, too, may bring different results. The performance of such highly complex examination, however, goes beyond the framework of this paper, since it would require the knowledge not only of the accurate numerical value of investment costs, but the trends of electrical power demand as well. Due to the same reasons, we have abstained from discussing solutions in which the full steam quantity generated in the waste-heat boiler is passed through an extraction-condensing steam turbine or a so-called "heating turbine" with varying backpressure, and the steam for heating is at all times taken from the extraction, respectively the back pressure. Finally, all these considerations are decisively influenced also by the size of the plant.

* W_1 is, namely finite; $W_2 = \infty$ in which case $- KF/W_1$
$Q = 1 - exp(-KF/W_1)$ viz: $\Phi = \Phi\ (K,\ F,\ W_1)$ function in which:
$K =$ const; $F =$ const; $W_1 =$ const; thus $\Phi =$ const.

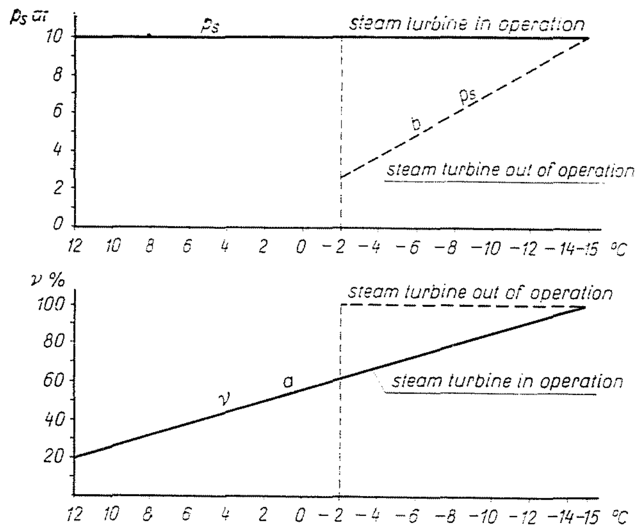

Fig. 4

II

Combined district heating power station with stationary gas turbine plant

After having discussed the conditions prevailing with the use of aircraft jet propulsion engine in the system of combined power supply, we are now going to approach the same problem with the insertion of a stationary gas turbine.

The circuit diagram referring to aircraft propulsions according to Fig. 1 can naturally be used also with stationary gas turbines, but here, there are facilities available for a more economical layout. Unlike aircraft engines, stationary gas turbines by utilizing waste-heat for steam generation, allow the reduction of the quantity of compressed cooling air and thereby of the specific compression work. The compressor, namely, in addition to combustion air, supplies also the quantity of air needed to cool down the products of combustion to the temperature permissible ahead of the gas turbine. Thus, by introducing the steam generated in the waste-heat boiler between the combustion chamber and the gas turbine, the specific amount of cooling air can be reduced and the so introduced steam, mixed with the products of combustion, expands in the gas turbine.

The heat flow diagram of the plant according to the above is shown in Fig. 5.

Fluegases leaving the open-cycle gas turbine plant consisting of the compressor (K), combustion chamber (E) and gas turbine (GT) are passed into the waste-heat boiler (FK) then exhaust to atmosphere. The waste-heat boiler produces steam (at a pressure of p_v). This steam is expanded to the back-pressure (p_e) in the back-pressure steam turbine (T). Having performed its work steam at a pressure of (p_e) can be utilized for two purposes:

a) to meet the momentary district-heating demand;

b) introduced behind the combustion chamber of the gas turbine, to increase the mass of working medium passed into the gas turbine (GT) and expanding, together with it to the atmospheric pressure, to boost the output.

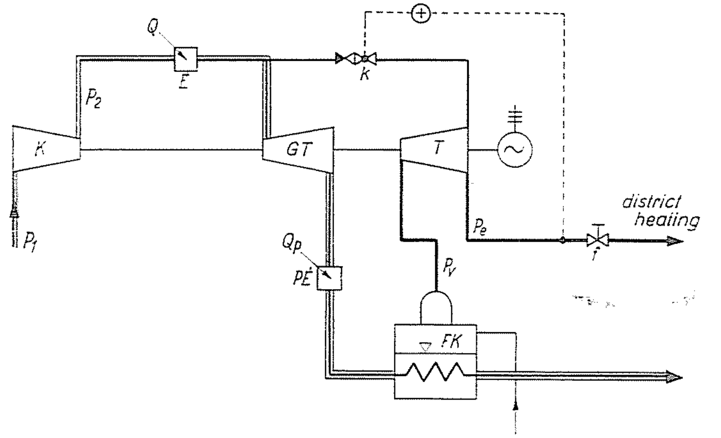

Fig. 5

This circuit is from a thermodynamical viewpoint naturally inferior to expanding the steam generated in the waste-heat boiler in a condensing steam turbine instead of introducing it into a gas turbine. The steam introduced into the gas turbine cycle, namely, must be warmed up by external heating to the temperature prevailing ahead of the gas turbine and, although the so introduced additional heat, in theory, corresponds only to the thermal equivalent of the expansion work gained from steam, nevertheless, the heat rate of the system will increase. In spite of this fact, the circuit according to Fig. 5 seems to be the most economical, due to the following reasons:

If expanded in a separate condensing turbine, steam produced in the waste-heat boiler, would generate some 30 to 40 per cent of the electric output of the gas turbine, the percentage depending on the plant parameters. The first costs of a condensing steam turbine plant of similar output (inclusive of auxiliaries, cooling tower etc.) are quite substantial. At the same time, the admixing of the steam ahead of the gas turbine would cause only a negligible influence on the investment costs. The expansion unit of a gas turbine plant would, namely, yield 300 to 400 per cent higher output than the alternator and the so introduced steam would increase the volume flowing through the turbine as little as some 5 per cent.

Although a separate steam turbine — even though not to a significant degree — is more advantageous thermically than the admixture of steam, it has its contraindications. For instance, the internal efficiency of a gas turbine absorbing a many times greater volume of gas is proportionately superior to that of a steam turbine. On the other hand, in a separate steam turbine the extraordinarily poor exploitation during the heating season, viz. in the overwhelmingly greater part of the operating period, would cause constant thermal (throttling) losses. The steam turbine would obviously be of the extraction-condensing type, inserted in such a way as to cover the heating demand off the extraction point, whereby the steam passed through the low pressure portion of the turbine (between the extraction point and the condenser) would in the overwhelming, part of the operating time serve practically only for cooling, and cause losses. This fact involves another drawback, in that, to keep this loss at a minimum, the steam flow of the turbine must be designed for a value not higher than the maximum steam rate producible without supplementary firing. In such case, accordingly, in the heating peaks the additional steam generated by the supplementary firing, is led to the heat exchangers direct, bypassing the turbine. This, in turn, means that the facility for supplementary firing cannot be utilized in the electric peak load periods for the production of additional electric power. On the other hand, additional steam produced by supplementary firing — according to the above — can readily be admixed ahead of the gas turbine, without any difficulty, since the additional gas volume is still insignificant in comparison with the gas flow of the gas turbine.

Summing up what went before and taking into consideration the electric peak load plant character of the station, a detailed calculation of economics is likely to point towards the scheme according to Figure 5, as the one which needs lower investment costs.

In the calculations we used the same initial data as in connection with the aircraft propulsion engines. Accordingly, the heat demand to be covered at an ambient temperature of -2°C is *20 million kcal/h* which the plant is capable of supplying without supplementary firing, and the outlay of the latter is such as to be able to supply round 32 million kcal/h at an ambient temperature of -15°C.

In the periods of varying ambient temperature, and according to the momentary heating and electric peak loads, operation is controlled practically in the same manner as with aircraft jet engines (see operation conditions as per groups 1 through 5).

The mechanism of control, however, adapts to the cycle used in this application. The pressure p_v in the waste-heat boiler adjusts automatically, without external interference, to the working conditions and the steam generated flows to the turbine at a pressure of p_v. The back-pressure p_e behind the turbine is maintained at its specified value by the regulator valve (k). The valve (*f*) regulating the steam used for district heating effects controls according to the momentary heating demand. Should the heating demand call for the full quantity of steam generated in the waste-heat boiler, the pressure p_e will drop and the valve (*k*) close completely. If heating does not require the full quantity of steam produced in the waste-heat boiler, the valve (*f*) will allow less steam to pass through and the back-pressure p_e will rise, whereupon the valve (*k*) opens and lets the additional steam pass toward the gas turbine.*

In such a way, steam produced in the waste-heat boiler is always utilized completely, partly for district heating and partly for expansion in the gas turbine.

What went above, was naturally merely the principal layout of the controlling; there are numerous possibilities for the practical design of the control system. The back-pressure steam turbine, for instance, may be of the extraction-back-pressure type, with the valve (*k*) coupled to the extraction line. This solution may be chosen when the steam pressure demand of heating is significantly below the rational pressure to be maintained in the combustion chamber of the gas turbine. Also the variant of the latter case is possible, in which the back-pressure behind the steam turbine is not constant, but adapts freely to the momentary heating load.

The dimensioning of the back-pressure turbine also requires optimization. The steam-flow of the turbine is obviously less than the steam rate corresponding to maximum heating load ($32 \cdot 10^6$ kcal/h at $-15°C$ ambient temperature). The line behind the turbine is therefore connected also by a bypass valve to the boiler. The optimum calculation must determine, ultimately, whether it is worthwhile at all to insert a back-pressure steam turbine into the system.

The examination of all these possibilities would go beyond the scope of this study and so would the determination of the optimum value of the back-pressure p_e in accordance with actual heating and electric peak loads. The latter would naturally determine also the pressure in the combustion chamber, and in our calculations the assumed 6 at the combustion-chamber pressure is merely the basis for an optional calculation and does not represent the optimum even for the simple case assumed in this paper.

The cycle according to Figure 5, as said, is suitable to control all conceivable working conditions, just as in the aircraft jet propulsion plant; any set of conditions can be adjusted between the two extremes. One of the two extremes is district-heating at $-15°C$ ambient air temperature when by the supplementary firing (PÉ) an additional heat quantity of Q_p is introduced behind the gas turbine and when the full steam quantity expanded through the steam turbine is used for heating. The opposite extreme is the case in which with maximum electric power requirement in summer or in winter, with the

* The lower limit of the back-pressure is naturally determined by the fuel introduced into the combustion chamber (E) resp. the supplementary combustion chamber (PÉ).

heating shut down, the full steam quantity produced by the heat Q_p introduced by the supplementary firing, after expanding in the steam turbine (or, part of it bypassing the steam turbine) is further expanded in the gas turbine down to exhaust pressure.

The cycle outlined has another advantage over the one according to Fig. 1 — provided that a back-pressure steam turbine is used: if heating steam is expanded through the back-pressure steam turbine, electric power can be generated as a by-product. Its drawback at the same time is, that the steam expanding through the gas turbine is lost in its entirety and must be made up by fresh feedwater, continuously. The seriousness of this disadvantage is mitigated by the fact that the low pressure waste-heat boiler does not require a costly water pretreatment process on the one hand and that, on the other, it can be evolved in a structural layout which is easy and fast to clean. (If a condensing steam turbine would be applied, the same amount of water would be lost by evaporation in the cooling towers.)

In the schematic diagram of Figure 5, we have not discussed the problem of the preheating and deaeration of the condensate arising in the district heating system, and of the boiler make up. There are numerous facilities for its economical solution: by steam taken from the turbine, or by supplying the waste-heat boiler with a suitable feedwater preheater economizer. To the latter facility we have not extended not only because it is beyond the scope of the present study but also because in this respect the sulphur content of the fuel is even more significant than in the case discussed above, due to the considerably higher optimum value of steam pressure (p_e) to be maintained in the waste-heat boiler.

The feedwater heater (to function at the same time as a deacrator) can be evolved also as a secondary cycle of lower pressure, which is a particularly advantageous solution with fuels having sulphur content.*

A significant factor of the optimum evolution of the heat flow diagram according to Fig. 5 is the determination of the optimum value of the pressure (p_e) in the waste-heat boiler. The effects manifesting here have been calculated for an optional case.

The following starting data had been assumed for the calculations (Fig. 6).

P_1	1 at.
P_2	6 at.
t_3	700°C
Internal gas turbine efficiency	$\eta_g = 86\%$
Internal compressor efficiency	$\eta_k = 84\%$
p_e	6 at.

* This cycle has been used by Sulzer in the ENSA power plant in Neuenburg (Switzerland).

Internal steam turbine efficiency $\eta_t = 80\%$
Δt 30°C

The district heating system being of the 130/70 type.

When determining the optimum value of the pressure in the waste-heat boiler (p_v), the degree of superheating (Δt_t) was also varied.

Calculations have shown that operation without superheating is always the most economical; this means that it is a paying proposition to produce the possible largest amount of steam with the heat given up by the fluegases entering the waste-heat boiler.

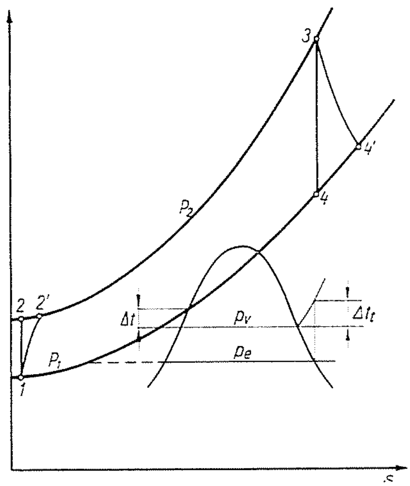

Fig. 6

The conditions and their variations for the case when there is no district heating demand and the full steam quantity generated in the boiler is expanded through the gas turbine, is illustrated by Fig. 7. As seen from the figure, maximum output was obtained in operation without superheating ($\Delta t_t = 0$) while by increasing the superheating ($\Delta t_t = 50$ and $\Delta t_t = 100$°C) the output dropped. The optimum value of pressure in the waste-heat boiler, in such operation, was around 10 at.

The variations of optimum steam pressure for the cases in which part of the steam is spent for district heating, were studied at all times without superheating. The so obtained set of curves is shown in Fig. 8. As it is apparent, optimum steam pressure rises as the share of steam used for heating increases. At zero heating demand this value is around 10 at., while with 100 per cent steam going for heating, the curve standing for the optimum steam pressure begins to flatten above 35 at. For the determination of the actual optimum, therefore, the yearly variations of steam spent for heating must be taken into consideration. In consideration of the temperature frequency curve (Fig. 9), correspond-

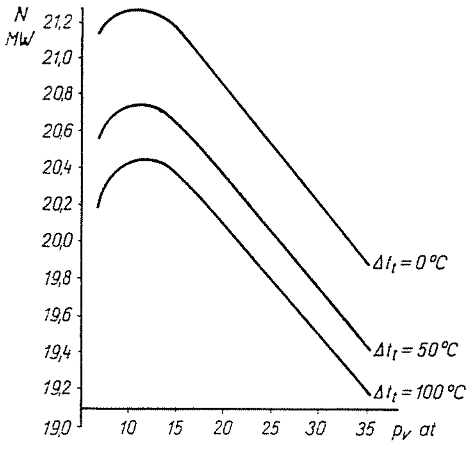

Fig. 7

ing to the climatic conditions in Hungary, the amount of electric power obtainable during the year can be plotted in function of the optimum pressure maintained in the waste-heat boiler. This function is graphically illustrated by the curve "L" in Fig. 8. It shows that considering a round-the-year operation of the heating power station, the optimum boiler pressure p_v is 17.5 at. (the numerical values of Figs 7 and 8 refer to operation without supplementary firing).

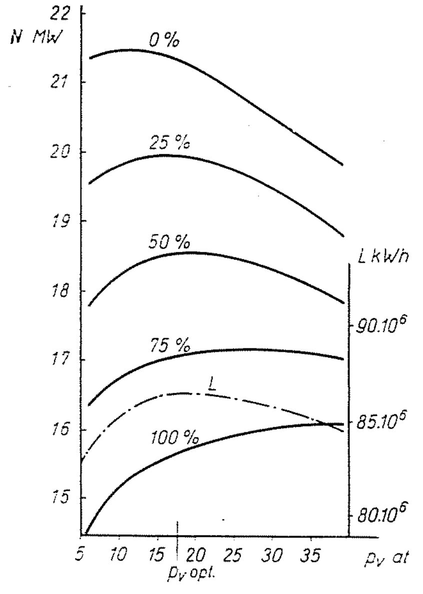

Fig. 8

The above calculations were intended only to point to the main features of the system proposed. Calculations must always rely on precise data, relating to all those values which are required for the performance of a detailed calculation of economics. In addition to the knowledge of the trends of heating and electric peak loads, composition of the fuel, first costs of the equipment, costs incurred with maintenance and personnel, annuality rate etc. etc. must also be known.

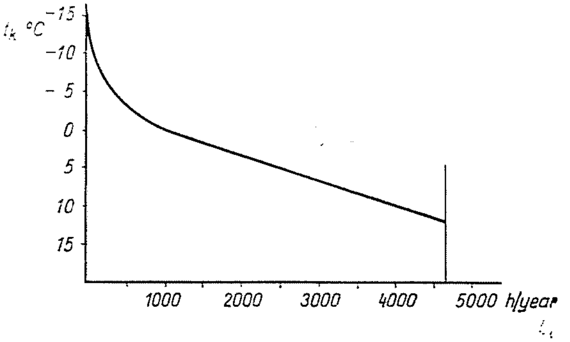

Fig. 9

All these calculations are needed for detailed design. The subjects touched upon in this brief study were intended to point out facilities for the economical application of district heating combined with the supply of electric power for peak loads and to prove that it holds out good promise for the future.

Summary

Specific heat consumption of public utility power stations is steadily decreasing, so that economy of district heating plants, based essentially on the production of cheap back pressure power, is ever less satisfactory. Nevertheless, quite favourable district heating stations can be established as peak stations of a public service network, a possibility due to the gas turbine cycle process. Based on the detailed thermodynamical analysis of two different cycles — using aircraft jet propulsion engine as gas generator and a stationary gas turbine — thermal and other advantages of the proposed system are presented.

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