PERIODICA POLYTECHNICA SER. MECH. ENG. VOL. 35, NO. 3, PP. 147–160 (1991)

# GAS TURBINES CONNECTED BEFORE HOT WATER BOILERS

A. PENNINGER, Z. FÜLÖP, B. CZINKÓCZKY, I. MÓRICZ and Á. KÓBOR

Department of Heat Engines
Technical University, H-1521 Budapest

Received: December 20, 1991.

## Abstract

Due to the efficiency of heat utilization equipment connected after gas turbines, the use of this equipment is widespread. This study examines the use potential of gas turbines connected before hot water boilers, also widely used in Hungary, to determine, in addition to electric power production, how reliable operation can be maintained with as little modification of the heat utilization equipment as possible. The recommendation is suitable for electric power production in both normal and peak operation since it retains the original burners together with their blowers.

This study has been made in relation to boilers in Hungary, of about 15 PTVM–100 units and 1 PTVM–180 unit.

This study includes the basic parameters of the gas turbine, the parameters needed to determine its energy conditions, the recommendation for the so-called canal burner placed after the gas turbine and the power engineering parameters of the boiler.

*Keywords:* gas turbines, water boilers, electric power production.

*Nomenclature*

| | | |
|---|---|---|
| $m$ | mass flow | [kg/s] |
| $B$ | fuel flow | [kg/s] |
| $P$ | electric power | [MW] |
| $\eta_e$ | eff. efficiency | |
| $\eta_b$ | gross efficiency | |
| $T$ | temperature | [°C] |
| $Q$ | thermal power | [MW] |
| $\sigma$ | specific electric power | |
| $q_G$ | specific heat consumption | |
| $q_Q$ | specific fuel heat consumption | |
| $m$ | excess air factor | |
| $\Pi$ | pressure ratio | |

*Subcripts*

| | |
|---|---|
| w | water |
| f | fuel |
| in | inlet |

## Determination of Gas Turbine Basic Parameters

The basic parameters of the gas turbine connected before the hot water boiler can be selected from the basic data of the boiler and from the examination of the parameters of gas turbines available today in this output range.

For the thermal power of $Q$ =116 MW of the hot water boiler type PTVM–100, the volume flow of air entering the boiler (at maximum load) is

$$V = 144691\,\mathrm{Nm}^3/\mathrm{h} = 40.25\,\mathrm{Nm}^3/\mathrm{s} \qquad (m = 52.04\,\mathrm{kg/s})\,.$$

The gas quantity used is

$$B = 12\,974\,\mathrm{Nm}^3/\mathrm{h} = 3.60\,\mathrm{Nm}^3 = 2.81\,\mathrm{kg/s}\,.$$

The maximum mass flow of the medium entering the boiler is

$$m_{\mathrm{in}} = 52.04 + 2.808 + 54.85\,\mathrm{kg/s}\,.$$

The mass flow of the inlet air is taken as

$$m_{\mathrm{in}} = 50\,\mathrm{kg/s}\,,$$

which is also assumed for the intake of the gas turbine. The purpose of this assumption is that the turbine intake should not be greater than the nominal one even in winter operation, therefore, the boiler furnace should not, in any circumstances, be under overpressure. The schematic diagram of the unit is shown in *Fig. 1.*

The outlet medium flow of the gas turbine enters the boiler furnace through the so-called canal burners installed at the bottom of the boiler type PTVM–100 without meeting the present burners of the boiler. Thus two types of operation can be envisaged.

The normal boiler operation to date, if for any reason the gas turbine drops out or there is no need for electric power, then the equipment can be operated under the former conditions.

### *Gas Turbine Operation*

The essential condition is that the gas turbine operates always on nominal load. Gas turbine operation is: *heat utilization in the boiler.* At that time choker No. 2 is closed — which is, however, open at the start of the gas turbine — and the flue gases from the turbine exit into the boiler furnace

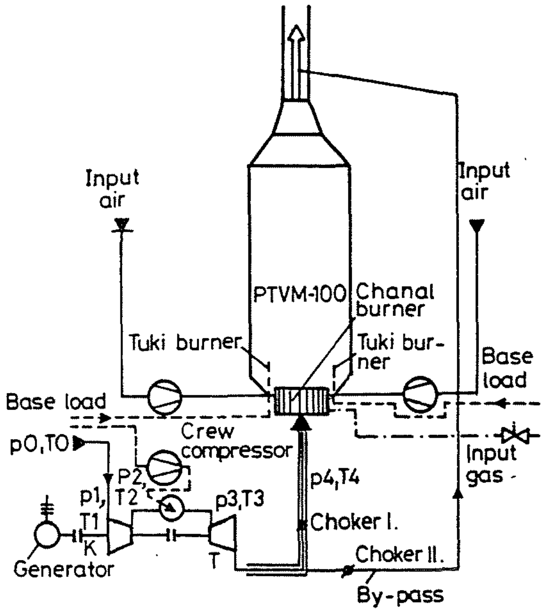


*Fig. 1.* Schematic diagram

and/or through the heat exchanger into the stack. Now the heat utilizing boiler operates at basic load. If this basic load is to be reduced for some reason, this can be done up to zero load using chokers No. 1 and 2.

Should the case occur that only electric power production is desirable or required but the boiler is not in running order or should not be operated for any reason, the turbine can be operated through the by-pass line (choker No. 2 is open, No. 1 is closed).

The loading of the boilers above the basic load can be without steps regulated by switching the canal burner on and off and/or by the inlet gas pressure. The canal burners are on the bottom of the boiler, their lighting up and operation will not disturb the thermal symmetry of the boiler in any operation mode.

## Expected Power and Parameters of the Gas Turbine

The parameters of the gas turbine have been studied, with consideration to all processes causing losses, using a computer model. For selecting the basic parameter the initial conditions were as follows:

— the turbine should be of the most simple construction, that is of single axis configuration without heat exchanger,
— the gas intake of the turbine should match the boiler, and it should operate always at nominal load when the turbine is in operation; the boiler loading above this should be made by additional firing,
— the turbine parameters such as compressor efficiency and turbine efficiency, maximum temperature on turbine blades, should be in accordance with the present state of the art.

The single axis configuration also means that the pressure ratio of the gas turbine is in the order of pressure ratio that can be achieved on a single axis (9 ... 10).

Natural gas firing was assumed for the calculations.

The gas turbine parameters, for the check calculation of the boiler, versus the gas turbine pressure ratio are shown in *Fig. 2.*

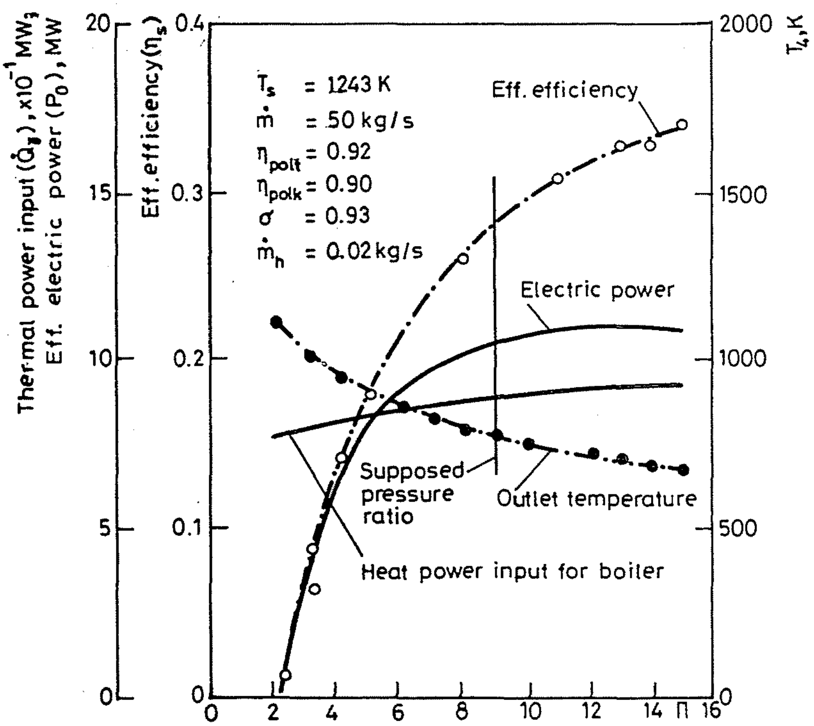


*Fig. 2.* Gas turbine parameters versus pressure ratio

At a pressure ratio of $\Pi = 9$, the power relative to the generator terminal is

$$P_e = 9.8\,\mathrm{MW},$$

which is assumed to be available in all operation modes. The minimum efficiency relative to axis power is

$$\eta_e = 0.28.$$

The temperature of the flue gases entering the boiler is

$$T_4 = 823\,\mathrm{K}.$$

The excess air factor in the gas turbine furnace is

$$m = 3.81.$$

At $\Pi = 9$ the maximum thermal power that can be released in the canal burner up to a minimum excess air of $m = 1.2$ is

$$Q_{1\max} = 87\,\mathrm{MW}.$$

*Fig. 2* shows the gas turbine parameters versus pressure ratio with consideration to the nominal parameters. At the nominal value the expected life of the turbine is about 40 000 hours, then the replacement of the turbine blades becomes necessary. The compressor efficiency and the turbine efficiency correspond to the value found in today's commercial gas turbines.

These efficiencies assume turbine housing with ceramic pads where clearance losses can be kept low. These values can be guaranteed for a relatively short period — a few thousand hours — then deterioration can be expected.

As *Fig. 2* shows, the medium temperature ($T_4$) after the gas turbine is essentially a function of $T_3$ which at lower turbine efficiency is just a little less than the temperature calculated at the 'nominal' value.

The parameters of the gas turbine determined by calculation fall close for example to those of the modern gas turbine of 8.84 MW type Mars GSC–12 000 made by Solar Turbines. This turbine has a specific heat consumption of 11 585 kJ/kWh, which corresponds to an effective efficiency of 3 600/11 585 = 0.31, which is even better than the 0.28 efficiency used in the calculation. The turbine's outlet mass flow is 38.17 kg/s, the outlet gas temperature is 460°C.

## Register Burner

The oxygen content of the medium leaving the gas turbine is high enough to burn additional gases in it. The heat generated from the burnable gas quantity can be obtained from *Fig. 3.*

From the gas turbine the medium is led through the canal burner with a self-contained burner nozzle. In the middle of the nozzle the medium from the gas turbine exits, and at its edge the natural gas exits, both with a swirl fixed intensity.

The layout of the planned installation of the burners is shown in *Fig. 3.* For the installation of the burners, naturally, the cooling pipes on the bottom of the boiler have to be relocated and the ignition and distribution line has to be run in front of the existing burners.

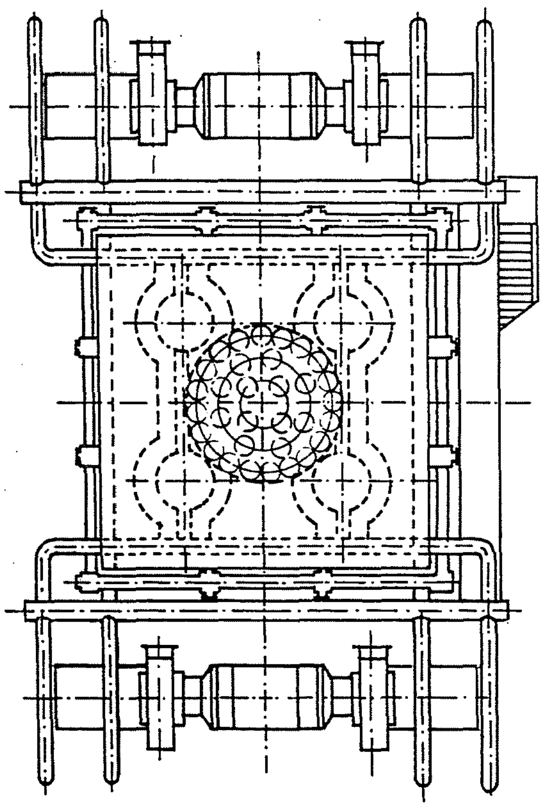

*Fig. 3.* Installation layout for canal burner for boiler type PTVM–100

The experiments with the burners conducted in the Department of Heat Engines at the Technical University of Budapest showed that the flame on the burners was stable between very wide limits, breaking off could not be created, while both the characteristics of the flame such as length and illumination capability and the design parameters of the burner could be adjusted to the requirements.

The efficiency of the burner is very good, the CO emission is 20 ppm and the hydrocarbon emission is below 5 ppm. This burner can be operated in a very wide range of air excess. The burner is not sensitive for air pressure fluctuation. It should be noted that the burner has a low flow resistance so it is very suitable for installation after the gas turbine.

The design of the burner *for a given purpose* can be made in the Department's laboratory.

## Power Engineering Calculation of Boilers Type PTVM–100 and 180 at Full Load

The power engineering calculations of the boilers were made as per [1] and [2], see *Table 1.* For boiler type PTVM–100, due to the same size of the convective heating area, the calculation of convective heat exchange for the two heating areas was made in one step, and for boiler type PTVM–180 the two convective heat exchangers were separated also in the calculation. Natural gas was used as fuel.

**Table 1**
Hot water boiler parameters at full load

| | | PTVM–100 | PTVM–180 |
|---|---|---|---|
| Boiler efficiency | % | 91.5 | 91.0 |
| Temperature of exiting flue gas | °C | 180 | 210 |
| *Water side* | | | |
| Inlet water temperature | °C | 104 | 80 |
| Outlet water temperature | °C | 150 | 130 |
| Water flow | kg/s | 594 | 1 000 |
| Useful power | MW | 116.4 | 210 |
| *Fuel side* | | | |
| Fuel flow | $m^3/s$ | 3.604 | 6.325 |
| Air flow | $m^3/s$ | 40.19 | 70.34 |
| Thermal flow taken into furnace | kW | 127 939 | 230 769 |
| Flue gas flow leaving furnace | $m^3/s$ | 43.7234 | 77.010 |

## Calculation of Boiler Operation Conditions at Base Load

During the boiler's base load operation the inlet medium parameters are as follows:

Mass flow of flue gas entering boiler: $m_f = 50\,\mathrm{kg/s}$
Temperature of flue gas entering boiler: $t_{\mathrm{fin}} = 550\,^\circ\mathrm{C}$
Water temperature entering boiler: $t_{\mathrm{win}} = 104\,^\circ\mathrm{C}$
(for boiler type PTVM–100)
Mass flow of water entering boiler: $m_w = 594\,\mathrm{kg/s}$ .

The calculation of flue gas and water leaving the boiler can be made by calculating the operation conditions under changing load as compared to the full load. To obtain these, we need the calculated values for the boiler's flue gas operation at 100% load.

*Table 2* includes the boiler's calculation for base load, that is, for that particular case when only the flue gas exiting the gas turbine and having an air excess factor of $m=3.81$ is used as fuel in the boiler. In this case the heating up of water in the furnace of boiler type PTVM can be disregarded.

**Table 2**
Gas turbine parameters connected before hot water boiler type PTVM–100

| Thermal power of additional firing | MW | 87.37 | 60 | 30 | 0 |
|---|---|---|---|---|---|
| Heat consumption by gas turbine | MW | 35 | 35 | 35 | 35 |
| Total heat consumption | MW | 122.35 | 95 | 65 | 35 |
| Electric power of gas turbine | MW | 9.8 | 9.8 | 9.8 | 9.8 |
| Heating power output | MW | 90.7 | 72.47 | 42.65 | 21.2 |
| Thermal power inlet to boiler | MW | 99.67 | 79.63 | 46.86 | 23.3 |
| Heat conservation | MW | 12.32 | 19.63 | 16.86 | 23.3 |
| Specific heat conservation | — | 0.135 | 0.217 | 0.395 | 1.10 |
| Relative heat conservation | — | 0.488 | 0.778 | 0.669 | 0.841 |
| Gross efficiency | — | 0.821 | 0.886 | 0.806 | 0.885 |
| Specific electric power | — | 0.108 | 0.135 | 0.229 | 0.462 |
| Specific heat consumption | — | 2.31 | 1.57 | 1.85 | 1.40 |
| Specific heat consumption | kJ/kWh | 8 831 | 5 646 | 6 663 | 5 069 |
| Specific fuel heat consumption | — | 0.963 | 0.820 | 0.706 | 0.00 |

## Boiler Operation Conditions above Base Load

Again the calculations were made only for boiler type PTVM–100 because from the results conclusions can be drawn for the operation conditions of boiler type PTVM–100. The results of the calculations are summarized in *Table 2*. By dividing the power taken into the boiler through the register burners into equal parts, the calculations were made for the thermal flow of

$$Q = 87.35\,\mathrm{MW},\ 60\,\mathrm{MW}\ \text{and}\ 30\,\mathrm{MW}.$$

The temperature of flue gas exiting the boiler and the water side power were determined for these three cases.

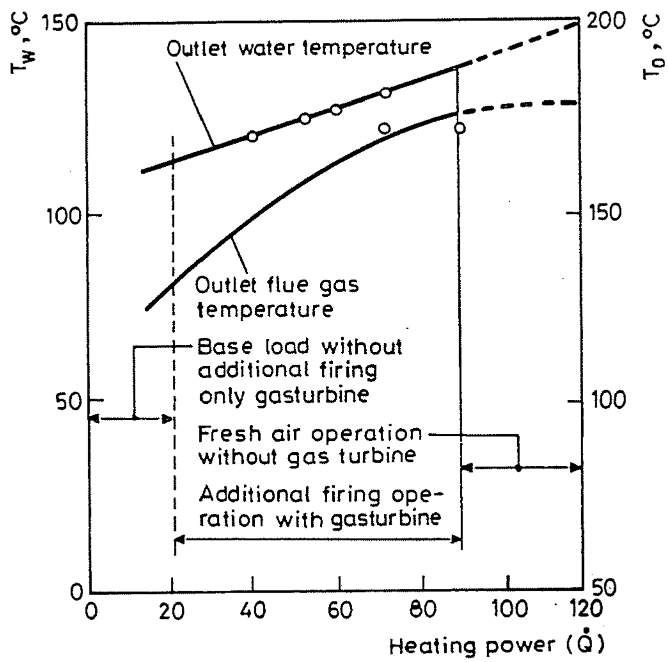


*Fig. 4*. Parameters of boiler type PTVM–100 after the gas turbine

## Gas Turbines Connected before Boiler Type PTVM–180

Two gas turbines, identical to those used for boiler type PTVM–100, can be connected before boiler type PTVM–180. This way, the electric power

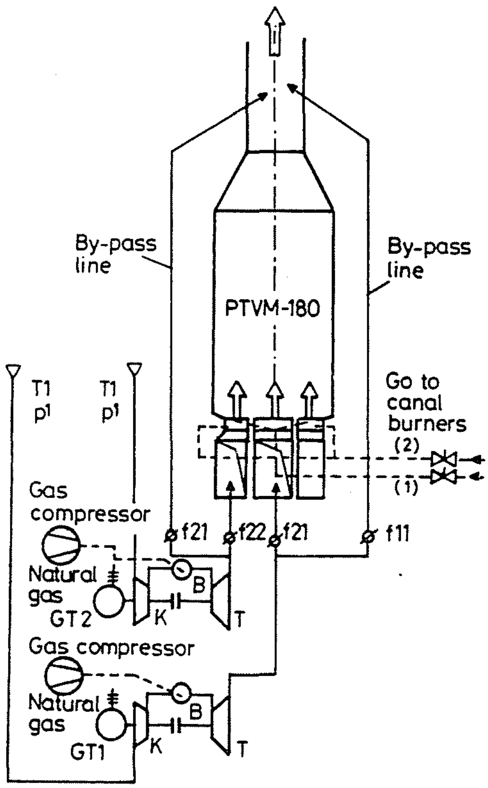


*Fig. 5.* Schematic layout of gas turbines connected before boiler type PTVM–180

in this connection is double of that of PTVM–100. According to their load, the gas turbines may operate separately or together in accordance with the parameters given above.

This connection allows boiler operation without gas turbine operation with by-passing the boiler.

At above base load operation of the hot water boiler the canal burners described for the PTVM-100 boiler will start operation allowing the stepless regulation of the boiler load.

The canal burners are located symmetrically in the boiler furnace divided into three parts. Power calculations for boiler type PTVM–180 were made for base load and nominal load operation conditions and the following results were obtained:

| Base load burners not in operation | | | Nominal load in operation |
|---|---|---|---|
| Air excess factor | | 3.81 | 1.2 |
| Air temperature leaving boiler | °C | 90.7 | 123 |
| Flue gas temp. leaving boiler | °C | 126.1 | 192.8 |
| Boiler power | MW | 44.8 | 124.94 |

It can be seen that boiler powers change about proportionally as compared to boiler PTVM–100. In the base load operation mode the flue gas temperature exiting the boiler is considerably low, the utilization of the second convective heat exchanger is bad, therefore, this operation mode should be avoided if possible.

## Evaluation of Energy Saving to be Achieved with Gas Turbines Connected before the Boiler

The heat conservation to be achieved through the utilization of the gas turbine's outlet heat is calculated by deducting the thermal power of the additional firing of the gas turbine operation from the heat consumption of the additional firing of the gas turbine. The calculation can be followed from *Table 2* and the results are illustrated in *Fig. 6*. Maximum heat conservation can be achieved at base load without additional firing, at that time the useful power output is $Q_w = 21.1$ MW, which is irrespective from the load, and by assuming a boiler efficiency of 91%, it means a conservation of 23.3 MW.

With an efficiency of $\eta = 28\%$ for the 9.8 MW gas turbine used in the calculation, it has a heat consumption of $Q_{IG} = 9.8 : 0.28 = 35$ MW. The utilizable part of this thermal power is $Q_{l\,\max} = 35 - 9.8 = 25.2$ MW, and this means the theoretical maximum. According to the calculation the degree of utilization of the outlet heat versus the heating power is 0.841.

By increasing the thermal power by additional firing, the heat conservation is reduced. The absolute heat conservation at the maximum 87 MW additional firing is only 12.32 MW and the relative value is 0.48. However, both the absolute and the relative heat conservation remain significant in the full load range.

The most favourable load is that without additional firing, and this represents 18% of the full boiler power which practically corresponds to the summer operation of the hot water boiler, for the production of domestic water. Here it is advantageous that in the summer there is a sure buffer of natural gas available.

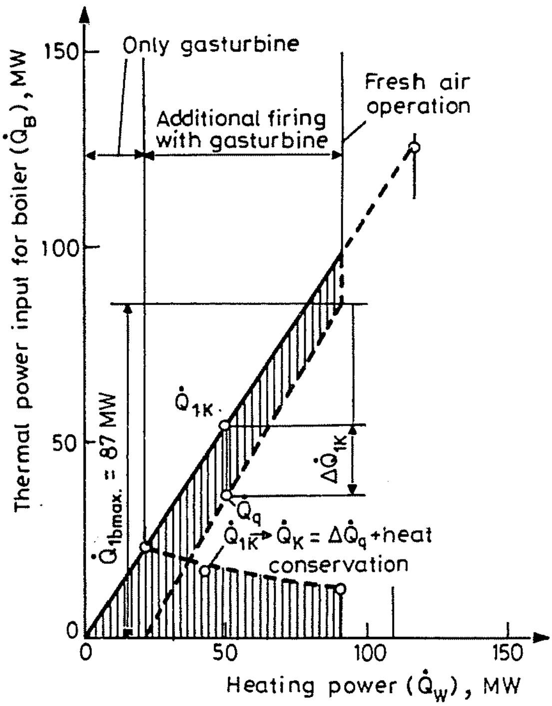


*Fig. 6.* Heat saving achieved with gas turbine before the boiler

The maximum additional firing constitutes 78% of the total thermal power of the boiler, which with the exception of the peak load lasting for a very short period, when gas turbine operation is not possible because of oxygen shortage, encompasses the full heating season.

The total efficiency (gross efficiency) of the connected system is quite high in the full additional firing range, and with increased additional firing the original $\eta_{br} = 0.88$ drops to 0.82.

However, the $\eta = P_e/Q_w$ specific electric power is relatively unfavourable and has a significant value of 0.462 only in the case of no additional firing, and drops to 0.108 with increased additional firing.

From this, the specific firing heat consumption can be calculated:

$$q_Q = \frac{1+q}{\eta_\gamma} - \frac{\sigma}{\eta_e}.$$

This index is best with no additional firing, then it has a value of 0. With increased additional firing, however, it increases up to 0.963.

The specific heat consumption of the gas turbine drops from

$$q_Q = \frac{3\,600}{0.28} = 12\,857\,\mathrm{kJ/kWh}$$

for the operation with no thermal utilization to 5 069 kJ/kWh without additional firing and even with maximum additional firing only to 8 331 kJ/kWh.

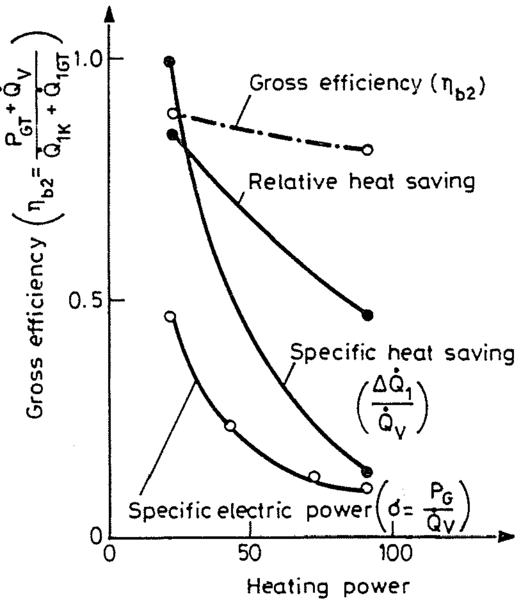

*Fig. 7.* Change of heat saving versus heating power

Based on the analysis of heat conservation and specific energetic indices, the placement of gas turbines before hot water boilers is advantageous and practical. Its application is mainly recommended in places where the use of larger combined gas turbine units is not possible due to the size of the district central heating system. In these places there is no other realistic solution for the updating of the existing hot water boilers. Other consideration is that the recommendation outlined here is based on the main existing types of equipment, the modification can be carried out by the Hungarian industry and the import ratio required is less than for the large combined gas turbine units with up-to-date parameters which is fully import equipment.

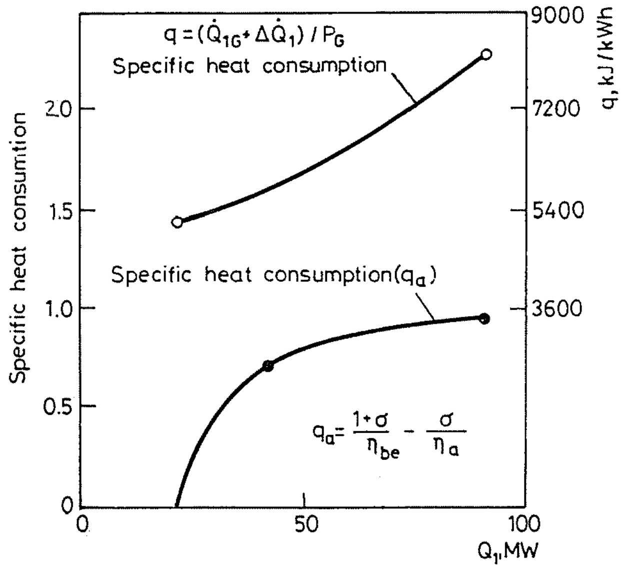


*Fig. 8.* Change of heat saving versus heating power

## References

FÜLÖP, Z.: Entwicklung einer Einrichtung zur stickstoffoxidarmen thermischen Verbrennung. *Gaswärme International,* 1991. 1/2.

FÜLÖP, Z.: Gas Turbines, Tankönyvkiadó, Budapest, (in Hungarian), 1975.

LUZSA, I. – MÓRICZ, I.–PÓTSA, E.: Construction of Steam Turbine, Tankönyvkiadó, Budapest, (in Hungarian), 1981.

LYDERSEN, A. L.: Practice of Heat and Mass Transfer. Müszaki Kiadó, Budapest, (in Hungarian), 1982.

*Address:*

Dr. Antal PENNINGER
Dr. Zoltán FÜLÖP
Dr. Botond CZINKÓCZKY
Dr. István MÓRICZ
Dr. Ágota KÓBOR
Department of Heat Engines
Technical University, H-1521 Budapest
Hungary