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24 October 1980

# USSR Report

## ENERGY

(FOUO 21/80)

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JPRS L/9365

24 October 1980

# USSR REPORT

## ENERGY

(FOUO 21/80)

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ELECTRIC POWER

UDC [621.311.25:621.039]:621.313.322-81

KHARKOV 'ELEKTROTYAZHMASH' PLANT BUILDS TURBOGENERATORS FOR AES

Moscow ELEKTRICHESKIYE STANTSII in Russian No 8, Aug 80 pp 5-8

[Article by G. I. Grigorash, director of Kharkov "Elektrotyazhmash" plant imeni V. I. Lenin; passages enclosed in slantlines printed in boldface]

[Text] In the 11th and 12th five-year plans both the number and capacities of nuclear electric power stations in the USSR will increase. They will require highly reliable, economic turbogenerators of unit power ranging from 200 to 1,000 MW, and excitation systems for them. The manufacture of new, powerful, four-pole turbogenerators is to be started.

To this end the Kharkov "Elektrotyazhmash" plant imeni V. I. Lenin has undertaken a reconstruction involving the installation of new, specially designed machine tools and heavy-duty hoisting equipment, modernization of the acceleration-balancing unit, and construction of a new test stand. The production base will make it possible to build turbogenerators of up to 2,000 MW of both 2-pole and 4-pole design.

The plant's research institute and technologists have also done much to study, develop and introduce progressive designs. The result has been the designing of powerful turbogenerators for AES with high technical and economic indicators.

The "Elektrotyazhmash" plant is a major manufacturer of heavy-duty electric power equipment. Since 1959 it has been specializing in building 200-, 300-, and 500-MW turbogenerators. The extensive experience in designing and building turbogenerators gained at the plant makes it possible to successfully tackle the tasks of building turbogenerators for AES. Principal among these tasks are: assuring high reliability, low cost and convenience in operation; assuring good operation under prolonged high loads; and better repair opportunities.

The electrical equipment for AES manufactured by the "Elektrotyazhmash" plant includes 200- and 500-MW turbogenerators at 3,000 rpm, 500- and 1,000-MW turbogenerators at 1,500 rpm, and excitation systems for them. The basic specifications of this equipment are presented in the table.

| Indicator | Turbogenerator TGV-200-2M | TGV-500 | TGV-500-4 | TGV-1,000-4 (design) |
|---|---|---|---|---|
| Rated power: | | | | |
| Active, MW | 200 | 500 | 500 | 1,000 |
| Apparent, MVA | 235 | 588 | 588 | 1,111 |
| Revolutions per minute | 3,000 | 3,000 | 1,500 | 1,500 |
| Rated voltage, kV | 15.75 | 20 | 20 | 24 |
| Rated power coefficient | 0.85 | 0.85 | 0.85 | 0.9 |
| Prolonged permissible peak-load power, MW/MVA | 220/259 | 550/611 | 550/611 | 1,100/1,220 |
| Temperature at rated operating conditions, °C: | | | | |
| Stator winding (according to resistance thermometers) | 60 | 63 | 61 | 67 |
| Rotor winding (according to resistance) | 74.5 | 71 | 68 | 58 |
| Stator core steel (resistance thermometers) | 77.5 | 74.5 | 70.0 | 73.5 |
| Efficiency, percent | 98.6 | 98.83 | 98.8 | 98.89 |
| Proportionate materials input, kg/kVA | 0.99 | 0.61 | 0.84 | 0.54 |

Let us examine in greater detail the new technological solutions incorporated in the designs of turbogenerators and exciters for AES aimed at assuring high utilization and reliability, as well as convenience in operation.

/TGV-200-2M turbogenerator/, power 200-220 MW, 3,000 rpm, with direct water-cooled stator winding and hydrogen-cooled rotor winding and stator core. The schematic design of the TGV-200-2M turbogenerator is presented in Figure 1. It can be installed at either a thermal electric station or an AES. Notably, this generator is used at an AES in conjunction with a BN-600 reactor.

The use of water cooling for the stator winding made it possible to obtain considerable peak-load reserves, thanks to which the TGV-200-2M turbogenerator can operate for long periods at a load of 220 MW at the rated power coefficient. Furthermore, water cooling assured even temperature throughout the body of the stator, which in turn reduced temperature strain and deformation and substantially enhanced the reliability





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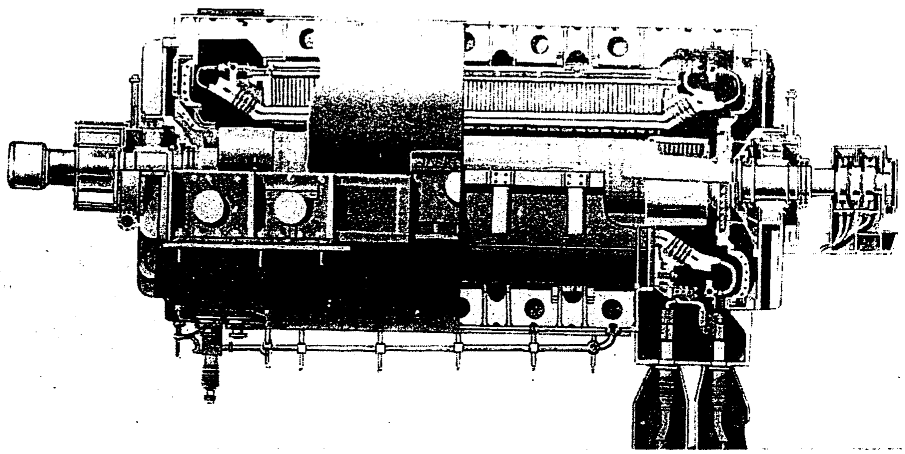

Figure 1. TGV-200-2M turbogenerator, power 200-220MW, 3,000 rpm (schematic design)

of the turbogenerator. The system of mounting the end elements of the stator winding designed by the "Elektrotyazhmash" plant and the design of a number of other generator elements assure its high repairability.

At present there are more than 25 TGV-200-2M turbogenerators in operation; they are convenient in service and highly reliable. The TGV-200-2M turbogenerator has been assigned the State Quality emblem.

The main excitation system for the TGV-200-2M turbogenerator at AES currently used is a thyristor autoexcitation system, and as a reserve a motor-generator excitation system which assure reliable feeding of the rotor winding and its back-up. The plant research institute subsequently carried out a greal deal of development and research in the creation of brushless excitation systems; as a result of this work, the BTV-300 brushless exciter was completed in 1977-1978. Its use at the Zaporozhye, Zmiyev, and Shatura GRES demonstrated its high reliability and effectiveness.

The transition to brushless excitation for TGV-200-2M turbogenerators in AES will make it possible to do away with the slip ring and brush holding unit, which requires special attention in machine-room conditions, thereby considerably improving operation conditions.

/TGV-500 turbogenerator/, power 500 MW, 3,000 rpm, has a water cooled rotor and stator windings and hydrogen cooled stator core. The

TGV-500 turbogenerator is shown mounted on the factory stand in Figure 2. The design of a turbogenerator of such high power with a water-cooled rotor was realized by the "Elektrotyazhmash" plant and its research institute for the first time in the world.

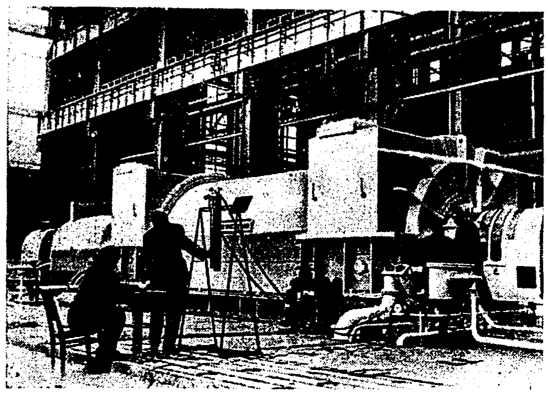

Figure 2. TGV-500 turbogenerator, power 500 MW, 3,000 rpm, on a factory stand

The use of direct water cooling of the rotor winding made it possible to build a 500-MW turbogenerator with the lowest proportionate materials input, high efficiency, and high winding and stator core heating reserve. However, in operating conditions, especially at the Troitskiy and Reftinskiy GRES, a large period of tuning up was required to increase reliability. Cases of operational failure have been due in the first place to the fact that not all solutions adopted in designing and elaborating the manufacturing techniques of the generator withstood the test in operating conditions. The problem of building such turbogenerators proved much more difficult than expected.

A series of design and technological measures were carried out to assure the high reliability of TGV-500 turbogenerators, more studies were carried out with models and in real-life conditions, objective operation-by-operation control methods were introduced, and production standards were improved.

At present a reliable system of independent thyristor excitation incorporating an auxiliary STV-12B turboexciter is employed to excite the TGV-500 turbogenerator. A motor-generator convertor unit is employed as a backup excitation source.

To eliminate the slip ring and brush holding unit the "Elektrotyazhmash" Research Institute began, in 1979, work on the manufacturing plans for a brushless exciter for the TGV-500 turbogenerator, manufacture of which may be launched in the next few years. The first-such exciter will be built for the Troitskiy GRES.

/TGV-500-4 Turbogenerator, power 500 MW, and TGV-1,000-4 Turbogenerator, power 1,000 MW, 1,500 rpm/. In most countries nuclear electric power production is based on slow-neutron, water-moderated reactors which produce steam at relatively low parameters (pressure about 80 kgfs/s$^2$, temperature about 300°C).

For such steam parameters and high-power units it is best to employ turbines with rotation speeds of 1,500 rpm and current frequency of 50 Hz, necessitating the building of 4-pole turbogenerators with rotor diameter and mass greater than for 2-pole turbogenerators.

In view of the importance of the problem, which required the solution of a number of specific questions, especially those associated with the manufacture of custom-made rotors weighing up to 160 tons, in 1965, pilot plants were designated for the manufacture of low-speed turbines and turbogenerators for AES, and provisions were made for necessary reconstruction of those plants.

Building the turbines was assigned to the design bureau of the KhTGZ [Kharkov Turbogenerator Plant] imeni Kirov, and the turbogenerators were assigned to the Kharkov "Elektrotyazhmash" plant. Consideration was given to the proximity of both plants in one city for purposes of collaboration.

In 1977, the "Elektrotyazhmash" plant built the Soviet Union's first 2 TGV-500-4 turbogenerators for the Novovoronezhskiy AES (Figure 3).

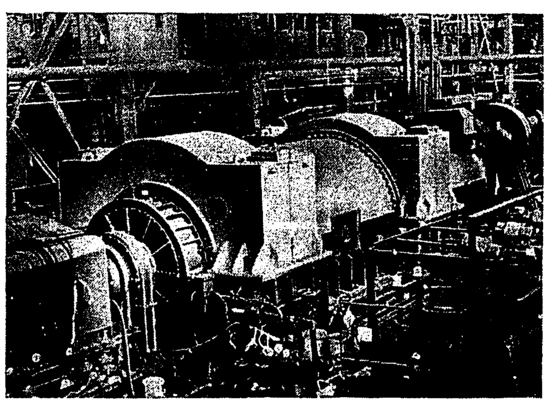

Figure 3. TGV-500-4 turbogenerator, power 500 MW, 1,500 rpm, on a factory stand

Comprehensive tests and studies of the turbogenerators on the factory stand showed that they possess considerable power reserves and can take considerable loads in excess of those permitted under operating specifications and state standards.

The turbogenerators have direct water cooling of the rotor and stator windings. The use of such principles of cooling windings makes it possible to build 4-pole machines of unit power up to 2,000 MW. Furthermore, technological continuity makes it possible to utilize accumulated experience in design and technology.

The TGV-500-4 turbogenerators employ an up-to-date brushless excitation system.

Test results for the BTV-500-4 brushless exciter show that it meets technical stipulations and specifications. A schematic design of the brushless exciter is presented in Figure 4 [photo not reproduced].

At present the plant's research institute has drawn up the manufacturing documents for the 1,000-MW, 1,500-rpm TGV-1,000-4 turbogenerator with the BTV-1,000-4 brushless exciter. The installation dimensions of the turbogenerator with the exciter have been unified with the dimensions of the TVV-1,000-4 turbogenerator developed by the Research Institute of the Leningrad Production Association "Elektrosila."

The use of water cooling for the rotor winding, improvement of the design during work on it, and more precise definition of the required operational parameters of the 1,000-MW, 1,500-rpm turbogenerator made it possible to develop a design with a rotor of virtually the same dimensions and weight as the one employed in the TGV-500-4 turbogenerator (Figure 5). Moreover, it was possible to employ for the TGV-1,000-4 turbogenerator a number of tested units from the TGV-500-4 turbogenerator without changing them (for example, packings, bearings, and others).

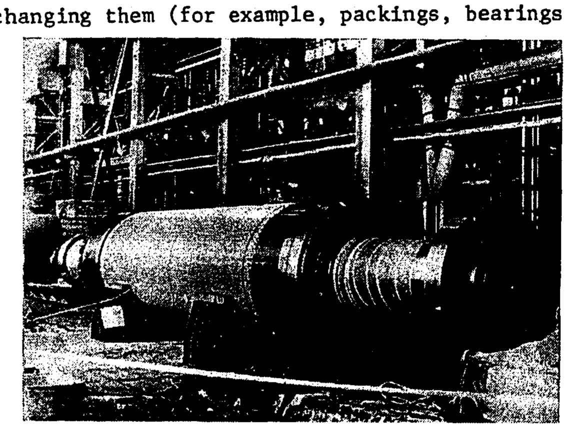

Figure 5. Rotor of the TGV-500-4 turbogenerator, weighing 150 tons, in a shop

The use of water cooling of the rotor winding for the 1,000-MW 4-pole turbogenerator makes it possible to reduce the generator's weight by 60-80 tons and raise its prolonged permissible peak-load capacity and efficiency as compared with a turbogenerator with a hydrogen-cooled rotor winding.



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## HEAT TESTING GTA-18 GAS TURBINE PLANT WITH RD-ZM-500 JET ENGINE

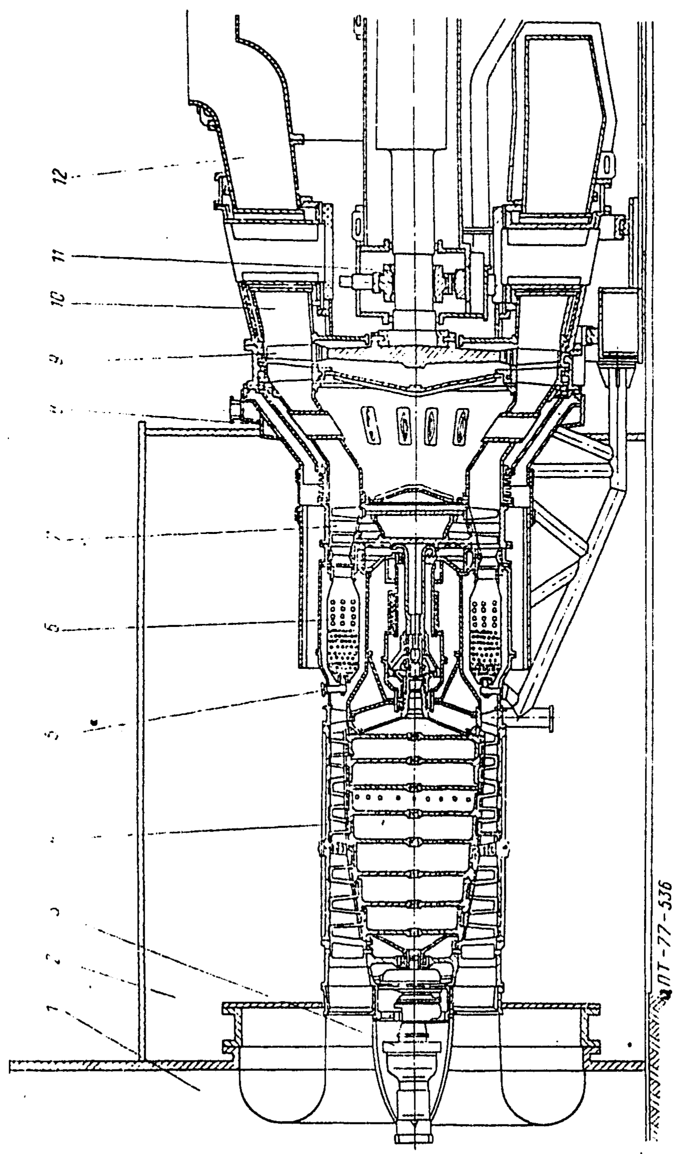
Moscow TEPLOENERGETIKA in Russian No 8, Aug 80 pp 23-28

[Article by Engineer V.G. Polivanov; Cand. of Technical Sciences G.G. Ol'khovskiy, L.V. Povolotskiy, M.P. Kaplan; Engineers L.A. Chernomordik, A.O. Bumarskov, I.N. Skvirskiy, P.I. Korzh; Cand. of Technical Sciences A.G. Tumanovskiy, PO KhTZ-VTI-Soyuztekhenergo [Khar'kov Turbine Plant All-union Thermotechnical Institute imeni F.E. Dzerzhinskiy-Soyuztekhenergo Production Association]]

[Text] Gas turbine units developed using aircraft engines as the base are widespread in power engineering and industry abroad. The leading motor-building firms Rolls-Royce (England) and United Technologies (Pratt and Whitney-USA) have formed special development and production divisions and have already produced thousands of such GTU [gas turbine units] with a total output of about 40 million kW. Experience in earth-bound operations has shown that the use of advanced aviation technology and scientific and technical developments and the advantages of large-series production, as well as methods adopted in aviation for the finishing work and for insuring the reliability of gas turbine engines permit high technico-economic and operations indicators while preserving the advantages associated with the small dimensions and weight of GTU and the possibility of very rapid start-up to full load (within 1.5-3 minutes). The output of foreign GTU with a single aviation engine now reaches 30-35 MW, and their efficiency is 32-34 percent.

Installations rated at between 1.6-3 MW with various modifications of the AI-20 turboprop engine, as well as installations rated at 4 and 12 MW designed around marine engines which have been engineered in the same way as have aviation engines, have found limited application in domestic power engineering.

The GTA-18 plant, a schematic of the design of which is shown in Figure 1, is the first domestic unit with an adequately high output (15-20 MW) with an aviation turbojet engine.



Key:

1. Air intake chamber
2. Box for turbojet engine
3. Turbostator
4. Compressor
5. Injectors
6. Combustion chambers
7. Turbojet engine turbine
8. Transition diffuser from turbojet engine to power turbine
9. Power turbine
10. Exhaust diffuser
11. Power turbine bearing
12. Exhaust outlet

Figure 1. Design diagram of the GTU (figure 1 continued on next page)

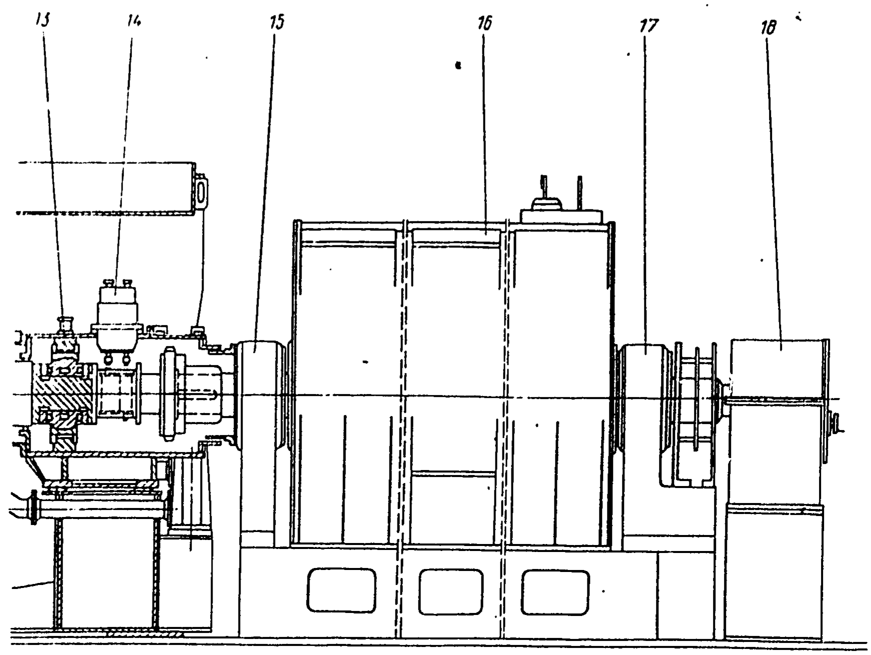



Figure 1. Design diagram of the GTU (continued)

Key: 13. Bearing-thrust bearing
14. Emergency governor unit
15. Electric generator bearing
16. Electric generator
17. Electric generator bearing
18. Control unit

a) PT-77-536

The unit was designed and produced by the Khar'kov Turbine Plant, and it was installed for testing under industrial conditions at the Khar'kov-energo TETs No 3. After adjustments and testing, normal operation of the plant with designed parameters was possible.

A detailed study of the individual aircraft engine components was not specified when preparing the heat tests for the GTA-18 gas turbine unit. Precise measurements of fuel consumption (using a precalibrated nozzle with a quarter-circle cross section) and electrical load (using a precision class 0.2 three-phase current watt meter which was duplicated by an active power meter) were conducted to determine the efficiency of the GTU. The temperature of the gas after the turbojet engine turbine (before the power turbine) was measured by 8 standard thermocouples placed uniformly around the perimeter in the inlet diffuser (3 of them were subsequently replaced by 4-point terminals), and the temperature of the gases beyond the power turbine was measured by 8 Chromel-Kopel open junction thermocouples installed on the horizontal section of the gas conduit from the GTU to the chimney.

Designing an economical power turbine and its mating with the turbojet was the main engineering problem in developing the plant because the level of gas velocities at the turbojet engine turbine outlet is very high. As a result, when preparing the tests, particular attention was paid to organization of internal measurements in the flow-through section.

Static pressure samples were taken from the walls to measure the pressures after the aircraft engine, before the power turbine, after the power turbine and at the outlet from the diffuser after the power turbine. Four holes were made in each section near the root and on the circumference [Russian--u kornya i periferii]. The pressures were let out from each hole by an impulse tube to outside assemblies where the tubes were united by collectors (the root and circumference separately in each section) to differential manometers filled with mercury or water. The static pressure was also sampled from the walls before the compressor and in the exhaust outlet section via 4 holes joined by ring collectors. Two terminals with 4 samplers situated in the centers of rings of equal area on each terminal were installed after the power turbine, at the inlet into the diffuser for direct measurement of the absolute pressure. These terminals were also used for taking samples of the combustion products and for determining chemical underburn.

All exterior indicators of the GTU were calculated based on the results of the measurements. Airflow rate was determined in the cycle from the heat distribution in the GTU. The absolute pressures and temperatures were computed according to results of the measurements using gas dynamics functions; the radial inequality of the velocities was also taken into consideration in the section after the turbojet engine.

The heat tests were performed using liquid fuel (aviation kerosene and diesel fuel) at outside temperatures of 20-27 and -2°C with loads up to 19 MW, as well as when using natural gas at outside temperatures of +5 and -10°C with loads up to 18.5 MW.

The basic test results characterizing the GTU's indicators are plotted in Figure 2 as a function of the reduced angular velocity of the jet engine and in Figure 3 as a function of the reduced output of the GTU.

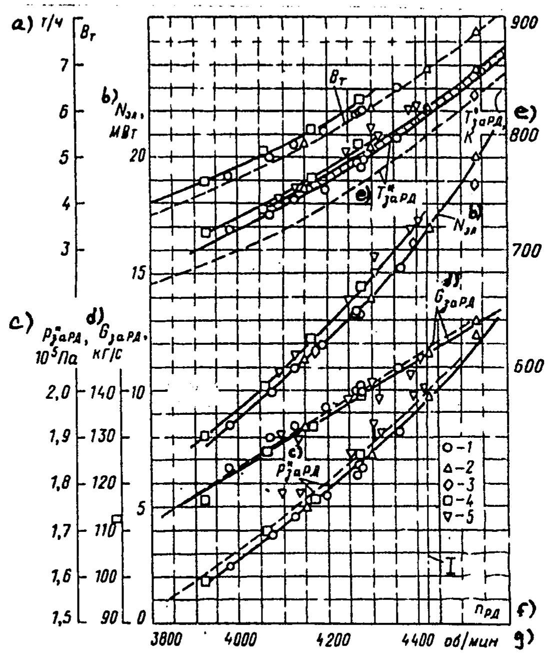



Figure 2. Dependence of the parameters for the GTA-18 on angular velocity of the turbojet engine (reduced to International Standard Atmosphere; $T_{ext}$ = 288 K, B = 1.013 x $10^5$ Pa). $B_T$ - fuel consumption; $T_{after}$ - absolute temperature after the turbojet engine (calculated from the output distribution); $N_{el}$ - electrical load (output); $G_{after}$ - gas consumption after turbojet engine; $p_{after}$ - absolute pressure after turbojet engine; calculated values for the following parameters are shown by dotted lines: I - rated duty of turbojet engine; 1 - June, 1977, liquid fuel, turbojet engine No 1; 2 - December, 1977, the same; 3 - January, 1978, natural gas, turbojet engine no 1; 4 - June, 1978, liquid fuel; 5 - October, 1978, natural gas, turbojet engine No 2

Key:
a) $B_T$, ton/hour
b) $N_{el}$, MW
c) $p^*_{after}$, $10^5$ Pa
d) $G_{after}$, kg/sec
e) $T^*_{after}$, K
f) $n_{TJE}$
g) rpm

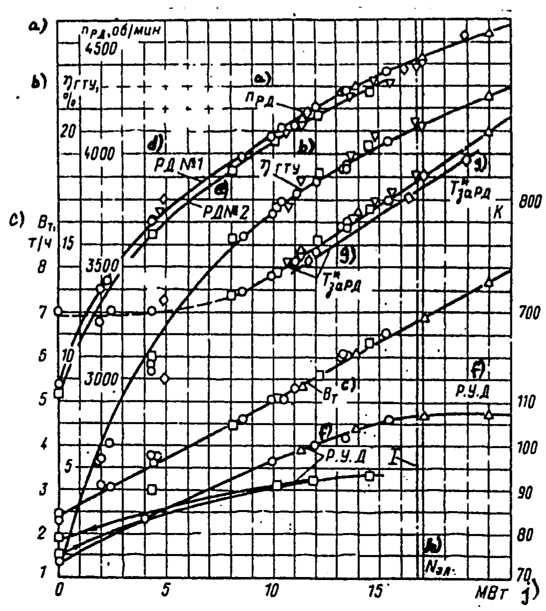



Figure 3. Dependence of the parameters and indicators of the GTA-18 on load (reduced to International Standard Atmosphere).

c.h.p. - engine control lever position; $\eta_{GTU}$ - overall GTU fuel efficiency based on heat consumption; cf. Fig 2 for remaining symbols

Key:
- a) $n_{TJE}$, rpm
- b) $\eta_{GTU}$, percent
- c) $B_T$, ton/hr
- d) Jet engine No 1
- e) Jet engine No 2
- f) c.h.p.
- g) $T^*_{after}$, K
- h) $N_{el}$
- i) MW

The absolute temperature values depicted in figures 2 and 3 were obtained from the power turbine's output distribution. The temperatures, which were measured at various points around the circumference of the flow-through part after the jet engine differ by 60-100 K (greater differences when the loads are smaller). The average measured temperature was 10K higher and the maximum temperature 50 K higher than equilibrium temperature (Figure 4).

It may be seen from figures 2 and 3 that at the rated, reduced angular velocity of the turbojet engine, the values for the output and the efficiency of the GTU under standard conditions are equal to 16.8 MW and 20.3 percent, respectively.

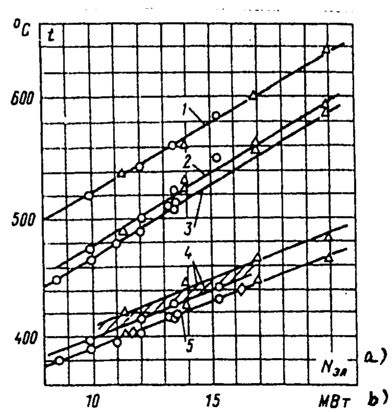



Figure 4. Characteristic gas temperatures when operating on liquid fuel (reduced to International Standard Atmosphere)

1 - maximum temperature after turbojet engine according to standard thermocouples; 2 - average temperature after turbojet engine measured by standard thermocouples; 3 - average temperature after the turbojet engine calculated from the output distribution; 4 - average temperature after the power turbine measured by standard thermocouples; 5 - the same, measured by research-type thermocouples

Key:
a) $N_{el}$
b) MW

(Note--see Figure 2 for explanation of the o-▲-◊ notation)

The dependencies of these indicators on external temperature and barometric pressure at $n_{TJE} = n_{rated} = 4425$ rpm are presented in Figure 5. Data corresponding to standard specifications at delivery are also plotted there. The tests showed that the actual output and efficiency of the GTU at the reference outside temperature (for the GTA-18) of +5°C (278 K) are 18.6 MW and 21.1 percent. The output is 1.9 MW (11 percent) greater than the guaranteed output. All of these indicators hold for pressure losses achieved in the experiments which are equal to conditions close to rated conditions, i.e. 0.2 kPa in the intake passage and 1.5 kPa in the exhaust passage.

In one of the stages during testing of the plant, the jet engine, being a gas generator for the power turbine, was replaced by an analogous engine (no 2). Test results for the unit with turbojet engine no 2 are also plotted in figures 2 and 3. As it follows from the graphs, the GTU indicators after changing the turbojet engine did not change

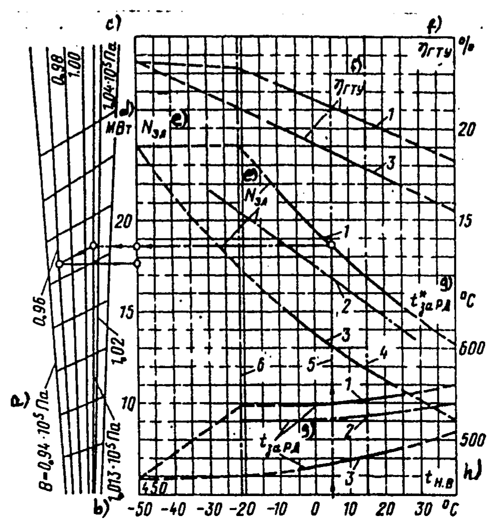



Figure 5. Dependencies of the GTA-18 indicators on external conditions

1, 2 - rated duty ($n_{TJE}$ = 4425 rpm) (1 - actual; 2 - calculated); 3 - 0.8 of rated duty; 4 - International Standard Atmosphere; 5 - calculated temperature of the outside air; 6 - maximum (calculated) output of the GTU. For example: $T_{ext}$ = +5°C, B = 0.96 x $10^5$ Pa (720 mm Hg); from the diagram we determine $N_{el}$ = 18.6 MW on the base line with B = 1.013 x $10^5$ Pa and $N_{el}$ = 17.6 MW at B = 0.96 x $10^5$ Pa.

Key:
- a) B = 0.94 x $10^5$ Pa
- b) 1.013 x $10^5$ Pa
- c) 1.04 x $10^5$ Pa
- d) MW
- e) $N_{el}$
- f) $\eta_{GTU}$, percent
- g) $T^*_{after}$, °C
- h) $t_{ext}$

for practical purposes, although the characteristics of the turbojet engines proper were not entirely identical. One may presume, for example, that the through-put capacity of the turbine in turbojet engine No 2 is somewhat greater (by about 2 percent), as a result of which the gas temperature and fuel consumption were somewhat higher and the airflow rate was the same as in turbojet engine No 1, given the identical angular velocity of the turbojet engines.

The values for the output and efficiency of the GTU which were given above and in Figure 5 were obtained from turbojet engine No 1. The plant with turbojet engine No 2 develops 0.6 MW more output at a correspondingly higher gas temperature at the exhaust (by 5-10°C) at the rated angular velocity.

We were unable to detect any increase in the output of the GTU during operation using natural gas. The GTU's efficiency values, calculated on the basis of fuel consumption, were lower, particularly under partial loads, as a result of the less complete combustion of the natural gas (cf. below); the efficiency values for the GTU which were calculated on the basis of heat consumption were practically identical.

The data presented in Figure 3 also illustrate the characteristics of the GTU under partial loads. It may be seen from the figure that under standard external conditions heat consumption when running without load is 32 percent of the rated value. The GTU's efficiency at half-load is 15.1 percent; it is 24 percent (relative) lower than the rated duty ($\eta_{50} = 0.76\eta_{rated}$). When loads are less than 4 MW, the engine operates with open bypass strip and discharge of a significant portion of the air compressed in the first stages of the compressor into the atmosphere (into the box). Closed bypass strips result in an increase in output up to 8-8.5 MW. The temperature at the outlet from the power turbine changes little as a result of the increase in consumption and pressure of the gases after the engine which occurs at this time. It begins to increase noticeably at $N_{el}$ [electrical load] > 8-9 MW.

Since we were unable to install absolute pressure terminals after the engine, in the section with the greatest level of velocities (up to 290 meter/sec in the experiments) out of design considerations, we calculated the values for absolute pressure in this section based on the average static pressure (between the root and circumference), the temperature and the velocity, with corrections for the velocity epure (compiled on the basis of data taken for an analogous stage) and the flow vortex which is present. After analysis, both of these corrections were taken as constants for operating conditions with a load of more than 10 MW; their total equals 11.5 percent of the dynamic pressure determined on the basis of the average velocity using gas dynamics functions. The conditionality associated with such a method for determining absolute pressure must be taken into consideration when evaluating the test results. In particular, comparing the calculated and the experimental values for the absolute pressures and the indicators for the diffuser passage from the engine to the power turbine, one may presume that this conditionality resulted in approximately a 2 percent reduction in the absolute pressure after the engine (cf. Figure 2).

The divergence ratio in the power turbine $\varepsilon^*$ under conditions close to rated load is 1.75-1.85; the available temperature drop is 125 kJ/kg, exhaust velocity is 140-150 m/sec ("losses" with an exhaust velocity $\frac{Ac^2}{2} = 11.3$ kJ/kg). The average values for the efficiency of the power turbine, calculated on the basis of absolute pressures after the turbojet engine and in the exhaust outlet section, are 84-85 percent, whereas they are 89 percent based on absolute pressure before the power turbine and in the exhaust outlet section. When calculated on the basis of static pressure in the exhaust outlet, these same efficiency ratings are 2-3 percent lower. Stage efficiency, calculated on the basis of absolute pressure at the intake and the outlet from the vane assembly, equals 93 percent. This efficiency drops off by 1 percent when the load is reduced from 15-20 to 8 MW, and there is a growth in $u/c_0$ from 0.5 to 0.7; the drop in efficiencies which takes into consideration losses in the exhaust passage as a function of operating conditions is somewhat greater (it is 2-3 percent). The throughput capacity of the turbine, given the designed divergence ratio, turned out to be about 1 percent greater than according to the design. The total turbine consumption increases noticeably with an increase in the divergence ratio up to the rated value and above it. Various dependencies of the throughput capacity of the turbine and nozzle on the divergence ratio may be one of the reasons that the test results differ from the engine's calculated throttle characteristics (Figure 2).

The characteristics of the passage from the engine to the power turbine (the efficiency of the diffuser $\eta_d$ and the coefficient of absolute pressure losses $\xi_d$) are also independent from the engine's operating conditions for practical purposes.

The average values for the efficiency and the coefficient of absolute pressure losses in the passage, as calculated according to the pressure measured after the engine, are equal to 84 and 17 percent.

Lower values ($\eta_d$ = 60 percent and $\xi_d$ = 28 percent) are obtained when calculations are made based on absolute pressure taken from the engine's performance rating. Even they indicate that the curvilinear passage from engine to power turbine, in which the gas velocities are reduced from 280-300 to 140-150 m/sec, have been adequately developed aerodynamically. The exhaust diffuser characteristics and those of the entire exhaust passage, including the outlet, are independent from $u/c_0$ for practical purposes under working conditions (with loads greater than 9 MW). The average diffuser efficiency is 55-60 percent, and that of the diffuser plus outlet is 42 percent; average coefficients of absolute pressure losses in these sections are 29 and 41 percent, respectively.

The completeness of liquid fuel combustion under operating conditions reaches 99 percent; when running without load it is 97.5 percent. The GTU exhaust was absolutely clean at loads less than 12-13 MW; at maximum loads, it was slightly colored but always remained transparent. The smudging ratio, which characterizes the content of soot particles in combustion products, was determined according to darkening of the filters through which a standardized sample of the gases was passed (0 - clean surface, 100 percent - absolutely black filter surface). It was 5 percent when running without load and about 40 percent under a load of 20 MW. The concentration of soot particles corresponding to the latter figure was about 50 $mg/m^3$.

In experiments using natural gas, a reduction in the completeness of combustion to 85 percent was observed under small loads.

The flues in the RD-ZM-500 engine's combustion chamber are made with subsequent introduction of air into the combustion zone. Highly boosted fuel combustion in chambers of this type is accompanied by formation of relatively small amounts of nitrous oxides. Their concentrations in spent gases at rated outputs of the GTU are 0.0025-0.003 percent. They are 0.0005 percent less when natural gas is used.

The parameters were also measured under start-up conditions. The GTU passes from engine idle conditions to having the electric generator running with no load at a moderate level of gas temperatures before and after the power turbine (not more than 450 and 400°C, respectively). The static pressure in idle $p_{before\ PT}$ is 3 kPa (gage), the heat loss $\Delta i_s$ in the turbine is about 8.4 kJ/kg. When running without a load, $p_{before\ PT}$ = 14-14.5 kPa (gage) and $\Delta i_s \approx 31$ kJ/kg, respectively. The values for $u/c_0$ under these conditions is significantly greater than the calculated values (more than twice as great).

Time metering of the start-up conditions showed that the engine's exit into idle continues from 80-120 seconds under various conditions. Maximum values for the gas temperatures after the engine, determined during this period by standard thermocouples, is 450-500°C. The power turbine begins to move when the angular velocity of the engine reaches 1,300 rpm. If start-up is not boosted after the engine goes into idle, the power turbine proceeds to stabilization of angular velocity at a level of 1500-1530 rpm in 2 minutes. This time was somewhat greater during certain start-up attempts. Starting up during the rated time (6 minutes to full load) met with no technical difficulties. Moreover, this time may apparently be reduced further by some 2-3 minutes.

Given standard external conditions (outside temperature = +15°C) and reference conditions ($t_{ext}$ = +5°C), the output of the GTA-18 is 16.8 and 18.6 MW, and the efficiency is 20.3 and 20.1 percent, respectively. Plant output is 1.9 MW (11 percent) greater than according to design.

The increase in output and efficiency of the GTU was the result of a certain increase in temperature after the jet engine, and, primarily, of a higher efficiency for the entire power turbine passage, than that with which it was designed.

Start-up of the GTA-18 occurs with a moderate level of temperatures ahead of the power turbine (no more than 450°C according to readings from standard thermocouples). It is possible to start up the unit and reach full load during the rated time of 6 minutes.

The methodology which was used during testing permitted us to determine reliably not only the external characteristics of the GTU, but the internal characteristics of the power turbine passage as well.



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# COMPARISON OF THE TECHNICAL-ECONOMIC INDICATORS FOR 3000, 1500 RPM 1000MW STEAM TURBINES FOR AES POWER UNITS

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[Article by Doctors of Technical Sciences N. M. Markov and L. P. Safonov]

[Text] The growth of unit ratings and the increase in the volumes of production for powerful turbine units for AES require the selection of their economically optimal design variants.

During development of steam turbines for AES, we are striving for standardization of low-pressure cylinders (LPC) on the basis of LPC from thermoelectric power stations (TES) turbines which have been well tested in operation. However, the throughput of the LPC used in TES turbines rated at 300-800 MW permits a unit rating of no more than 700-800 MW to be achieved for AES turbines. The number of cylinders in the turbine reaches 5 (a HPC [high-pressure cylinder] and 4 LPC), which is considered the limit for a single-shaft unit, both in the USSR and abroad.

AES turbine units with VVER [water-moderated water-cooled electric power reactor] and RBMK [expansion unknown] type reactors operate at substantially lower available heat losses (by a factor of 1.6-1.7) than do TES turbine units. Therefore, given identical output, the turbine units for an AES are designed for an appropriately greater throughput. For the limitation on the number of cylinders which has been adopted for the present, this is possible only by increasing the throughput and the unit output of the exhaust, which is determined by the ratio

$$N = \frac{1000 F_T M_{c_{2a}} p_2}{q},$$

where $F_T$ -- the exhaust area, $m^2$; $M_{c_{2a}}$ -- the Mach number of the delivery component for the steam exit velocity from the last stage; $p_2$ -- exhaust pressure, $kgf/cm^2$; q -- specific steam consumption in the condensor, kg/kW·hr.

Since the values of $M_{c2a}$ are close to the maximum values ($M_{c2a} < 0.8$) for the exhaust of modern turbines and q is practically constant given the desired initial steam parameters (3.1-3.3 kg/kW·hr), for the low values of $p_c$ adopted in the Soviet Union in accordance with technical requirements (0.04-0.055 kgf/cm$^2$), the problem of developing AES turbounits rated at 1,000 MW and more was irrevocably associated with the development of a new exhaust with increased face area.

Proceeding from the ratio which was given, it is possible to find the minimum required face area of individual LPC exhausts for AES turbine plants rated at 1,000 MW. Their values are presented in Table 1 as a function of the final pressure which has been adopted and the number of LPC.

Table 1

| Number | | Minimum area of individual exhausts | |
|---|---|---|---|
| | | Terminal pressure, kgf/cm$^2$ | |
| LPC | Individual Exhausts | 0.04 | 0.055 |
| 2 | 4 | 23-26 | 17.5-19 |
| 3 | 6 | 16-17.5 | 11.5-12.5 |
| 4 | 8 | 11.5-13.0 | 8.8- 9.5 |

Two basic solutions are possible to increase the face area of the exhausts: increase the length of the vanes by using valloys with a higher specific strength or convert to a reduced angular rotor velocity--25 rps (1,500 rpm). The second solution is simpler because the face area in this instance (given identical stresses in the runner) can be increased by a factor of 4. Realization of the first solution is connected with a multi-year search for metal scientists and designers, metallurgists and engineers, and it was exceedingly difficult to predict this process. About 10 years were spent on the development of the design and manufacturing technology for a new exhaust with 1,200 mm vanes made from a titanium alloy.

Considering the conditions which were set forth and proceeding from the promise for the development of nuclear power engineering, it was decided to create a 1,000 MW turbine with an angular velocity of 25 rps was at the end of the 1960's.

During these years, a production base for producing such units was created at the "Khar'kov Turbine Plant" Production Association, projects for 1,000 MW turbines were worked up and their production was prepared for. A 500 MW turboplant for the Novovoronezhskaya AES, which is an experimental production model, an analogue for development of the primary units for 1,000 MW turbines, has been installed and is in the start-up stage. Manufacture of a pilot 5-cylinder 1,000 MW plant with lateral condensers is presently being concluded at the "Khar'kov Turbine Plant" Production Association for the Yuzhno-Ukrainskaya AES. Technical documentation for a low-speed turbine without an intermediate pressure cylinder (IPC), with condensers being located under the shaft, has been developed in collaboration with the NPO TsKTI [Scientific Production Association of the Central Scientific-Research and Planning and Design Boiler and Turbine Institute imeni I.I. Polzanov] in 4-cylinder (1 HPC + 3 LPC) and 3-cylinder (1 HPC + 2 LPC) versions. The first variant is intended for high-vacuum conditions ($p_c$ = 0.04-0.05 kgf/cm$^2$), whereas the second is for lesser vacuum ($p_c$ = 0.055-0.065 kgf/cm$^2$).

A new LPC with increased face exhaust area (11.3 m$^2$) has been developed during the past decade at the "Leningrad Metal Plant" Production Association for the K-1200-240 turbine at the Kostromskaya GRES. In so doing, the first of the above-mentioned decisions was realized--a titanium alloy with a specific strength $\sigma_T/\rho$ approximately twice as great as that of ordinary vane steels ($\sigma_T \approx 80$ kgf/mm$^2$ with $p = 4.5 \times 10^3$ kgf/m$^3$) was used as the material for the rotor vanes in the last stages of the LPC, the length of which (at angular rotor velocity of 50 rps) is 1,200 mm.

It is possible to develop a turbine plant for a 1,000 MW AES with angular rotor velocity of 40 rps based on the LPC with increased throughput. The technical documentation for this type of turbine in 5-cylinder format was developed and defended by the Scientific and Technical Council of the Ministry of Power and Electrification and the Ministry of Power Machine Building. The through-flow section of this turbine's LPC has undergone meticulous final aerodynamic and vibrational work on large-scale test stands at the NPO TsKTI.[1]

Thus the USSR's power machine building industry has presently prepared for production 2 types of 1,000 MW turbines for AES with an angular rotor velocity of 25 and 50 rps. The basic specifications for these turbines are presented in Table 2.

At the present time, most of the turbines from abroad which are rated at 1,000 MW and higher for AES are being designed and manufactured to be operated at low speed. Such turbines are produced by the firms General Electric and Westinghouse Electric (USA), KWU (FRG), Alsthom (France) and Brown Boveri (Switzerland) in particular. However, a number of firms have in recent years developed and are planning production of high-speed turbines rated at up to 1,000 MW for AES (Brown Boveri, Stal-Laval, General Electric, KWU).

Table 2. Primary Specifications for Low-Speed and High-Speed K-1000-60 Turbines

| Characteristics | Type of Turbine | | |
|---|---|---|---|
| | K-1000-60/1500 | | K-100-60/3000 |
| Enterprise developing turbine | "Khar'kov Turbine Plant" Prod.Assn. | | "Leningrad Metal Plant" Prod.Assn. |
| Length of rotor blades in last stage, mm | 1450 | | 1200 |
| Number of LPC | 3 | 2 | 4 |
| Exhaust area, $m^2$ | 113.4 | 75.6 | 90.4 |
| Unit Steam Load of Exhaust, $ton/m^2\cdot hr$ | 28 | 44 | 35 |
| Rated pressure in condenser, $kgf/cm^2$ | 0.04 | 0.055 | 0.04 |
| Design (guaranteed) unit heat consumption (gross), kcal/kW·hr | 2480 | 2560 | 2500 |
| Weight of turbine, ton | 2992 | 2241 | 2171 |
| Length of turbine, m: without generator | 49.4 | 37.4 | 49.7 |
| with generator | 73.0 | 61.0 | 74.0 |

In spite of the fact that an adequately large number of works[2-4] et al has been devoted to the basis for selection of angular velocity for high-powered turbine plants (on the order of 1,000 MW), there is not as yet adequate specificity to this question. Even given an identical approach to turbine design, data from various studies digress significantly in evaluation of thermal efficiency (the advantage of low-speed turbines is determined by a magnitude of 0.5-3 percent[2-4]), and of reliability, particularly that based on dynamic strength and erosion reliability, weight-size and layout characteristics, etc., and, consequently, based on combined technical-economic indicators. The task becomes even more complicated if it is necessary to compare the turbine plants from several producer enterprises which have been designed according to different methods, with maximum consideration of the engineering and production capabilities of a specific plant, and employing engineering decisions which have been officially approved by them.

The specific nature of developing equipment for AES, fluctuations in the prices for fuel and metals, as well as the relatively wide range of operating conditions for 1,000 MW turbines in the USSR require a comparison of the various designs, not only on the basis of combined adjusted costs, but also on the basis of such important individual indicators as reliability, maneuverability and metal consumption.

This type of study was performed in 1978 at the NPO TsKTI. It differs from previously conducted studies by virtue of the fact that it is based on developed technical documentation for the K-1000-60/1500 turbine from the "Khar'kov Turbine Plant" PA and the K-1000-60/3000 from the "Leningrad Metal Plant" PA. These projects differ both in design methods and engineering decisions.

Under these conditions, the methods adopted as industrial standards were used to make a correct comparison (heat calculations, design of the low potential section and the heat configurations, calculations of temperature fields and low-cycle fatigue [Rus.--'malotsiklovaya ustalost''], analysis of erosional damage to the vanes and design of the vanes for the last stages).

1. Comparison of Turbines Based on Heath Efficiency.

Throughput section--We compared the efficiency of the throughput sections of the turbines by calculating the basic loss components according to the methodology in[5] for the following assemblies: in the nozzles and rotor vanes ($\Delta h_r + \Delta h_v$), with leakages through the nozzle and rotor vane seals and from wire connections ($\Delta h_l + \Delta h_w$), losses from humidity ($\Delta h_h$) and exhaust losses. The difference in individual loss components for the two types of turbines is presented in Table 3.

Table 3.

| Turbine | K-1000-60/1500 | K-1000-60/3000 |
|---|---|---|
| $H_{ad}$, kcal/kg | 275.05 | 284.0 |
| $\Delta h_r + \Delta h_v$, kcal/ka | 13.63 | 14.42 |
| $\Delta h_l + \Delta h_w$, kcal/kg | 7.38 | 9.55 |
| $\Delta h_h$, kcal/kg | 15.77 | 16.42 |
| $(\Delta h_r + \Delta h_v)/H_{ad}$ | 0.049 | 0.051 |
| $(\Delta h_l + \Delta h_w)/H_{ad}$ | 0.026 | 0.033 |
| $\Delta h_h/H_{ad}$ | 0.057 | 0.058 |
| $\Sigma\Delta h/H_{ad}$ | 0.133 | 0.142 |

It may be seen from the table that losses in the vane units are practically identical in both turbines, however, losses with leakages and from humidity are greater in the high-speed turbine. The total difference in energy losses is about 0.9 percent, in favor of the low-speed variant. Taking the computation method's degree of estimation into consideration, it may be considered to be equal to 1 percent.

Calculations of the changes in the efficiency of the last stages of the turbines being compared showed that, given total volume consumptions $nGv_2 = (24-29) \times 10^3$ $m^3/sec$(1) in the rated duty zone, the efficiency of the slow-speed variant is noticeably higher. As may be seen from Figure 1, the last stage of this variant shifts into a state of power consumption for low values of $nGv_2$. The approximate conincidence of the efficiencies of the stages being compared under close to rated conditions is explained by the off-axis emergence of the flow in the low-speed variant, a feature associated with attempts of the designers to insure more stable operation of the stage under partial loads. The studies of the variants made by the NPO TsKTI showed that it is only possible to increase the efficiency of this stage substantially (by approximately 2 percent) at rated duty by changing the angles at which the vane unit is installed.

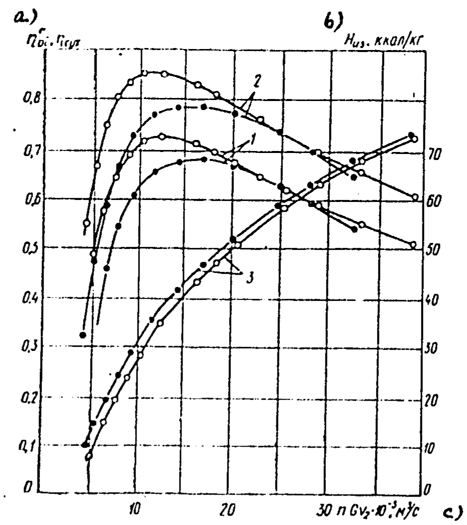


Figure 1. Change in efficiency and available heat loss in the last stages as a function of total volumetric rate through the turbine

-o-o Khar'kov Turbine Plant

-•-• Leningrad Metal Plant

1- $\eta''_{0i}$ stage efficiency; 2- $\eta_{dry}$ stage efficiency; 3- available (heat) loss

Key:

a) $\eta''_{0i}$, $\eta_{dry}$ b) $N_{is}$, kcal/kg c) $nGv_2 \times 10^{-3}$ [sic!], $m^3/sec$

Note: Point c) is given as "$nGv_2 \times 10^3$" in the text.

(1) n--the number of LPC in the turbine.

Heat configurations--When comparing the efficiency of the heat configurations, we pursued the goal of revealing the difference in efficiency between the variants when the standardized heat configuration proposed by the NPO TsKTI(1) is used in them and, consequently, of evaluating the effect of the specific design format of the turbine.

The selection of values for the separation pressure and the diverse number of low-pressure heaters are primary factors influencing the difference in efficiency of heat configurations. Calculations made on the basis of the standard methodology[6] are close to those most widespread throughout the world. Here losses from humidity were determined according to formula[7]:

$$\zeta_h = 2 \frac{u}{v_{is}} [0.9\, y_0 + 0.35\, (y_2 - y_0)],$$

where $u$ -- tip speed (of vane) on the average diameter; $v_{is}$ -- velocity corresponding to isoentropic (heat) loss of the stage; $y_0$, $y_2$ -- humidity before and after the stage.

It was determined as a result that the optimum value for separation pressure for a high-speed turbine is 8-9 kgf/cm$^2$, and 9-10 kgf/cm$^2$ for low-speed turbines (without an OPC), i.e. they are practically the same.

At the same time, the selected values of separation pressure for the types of turbines under examination(2) are 6 and kgf/cm$^2$, respectively.

The dependence of the thermal efficiency of the turbine unit on the separation pressure is presented in Figure 2. The dependence shows that the separation pressure is taken to be somewhat different from the thermodynamic optimum for both types of turbines; supplementary losses in the low-speed turbine are 0.15 percent and 0.6 percent in the high-speed, as a result of deviations from the optimum separation pressure value.

According to calculations which have been made, installation of a supplementary LPC in the high-speed turbine regeneration system (5 LPC) will produce an advantage of 0.45 percent in comparison with the low-speed (4 LPC). Therefore, the total efficiency of the heat configurations for

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1. Single-stage SPP [separator-steam superheaters] are used in this configuration, with condensate from the heating steam being pumped from the SPP into the high-pressure water supply line, pressure head is excluded and a deaerator working on a pressure of 10-12 kgf/cm$^2$ and a vacuum group of mixing heaters are used.

2. A separation pressure of 12 kgf/cm$^2$ was selected as optimal in the low-speed turbine for the variant with IPC.

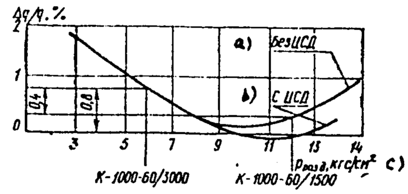

Figure 2. Change in specific heat flow from the separation pressure.

Key: a) without IPC
b) with IPC
c) $p_{sep}$, $kgf/cm^2$

the high-speed and low-speed turbine units, making maximum use of standardized units (according to proposals from the NPO TsKTI), particularly mixing vacuum heaters), is practically identical. Consequently, the difference in the thermal efficiency of low-speed and high-speed turbines is brought about only because of a certain difference in the efficiency of their throughput sections.

2. Static and Dynamic Strength of Basic Turbine Components. Comparison of Maneuverability and Reliability Indicators

An analysis of strength calculations for both the high-speed and low-speed turbine shows that stress in their primary components and the accepted safety factors are within the range of values officially approved for steam turbine construction in practice. The stresses in housing components are approximately identical, and they are less by a factor of 1.3-2 in the low-speed turbine rotors than in those of the high-speed turbine. (One must, however, bear in mind that the significantly larger dimensions of the low-speed turbine components may result in a decrease in service characteristics of the materials--the scaling factor. Moreover the possibility of using integral disc LPC rotors in high-speed turbines provides a greater safety factor.) In both cases, the safety factors are in total conformity with standardized engineering documentation.

A large set of scientific research and experimental designing was done to insure vibrational reliability of the rotor blades in the last stages of low-speed and high-speed turbines. These vanes were used in the K-500-60/1500 and K-1200-240-3 turbines and were studied in detail by the NPO TsKTI in collaboration with the plants, using large-scale models. They were rebuilt for satisfactory vibration characteristics as a result of this work.

The safety factor for the shafting during unexpected short circuiting is one of the most important reliability indicators of a turbine plant. The low pressure rotor journal closest to the generator is the most highly stressed section of the shafting under these conditions. A comparison of the maximum torques and stresses in the low pressure rotor performed according to the NPO TsKTI methodology showed that the shafting of the low-speed plant has significantly greater safety margins during unexpected short circuiting, although in both cases the value of the safety coefficient is greater than that permitted by technical reference materials (K - 1), being K ≈ 2.1 for a low-speed plant and K ≈ 1.3 for a high-speed plant.

Two factors which determine the physical possibilities with regard to the turbine were examined when analyzing the maneuverability indicators: their dynamic properties with specifically designed control systems and the accumulation of damage under identical working conditions.

The dynamic properties of the low-speed and high-speed turbines and their control systems were calculated by solving simplified differential equations of motion. Analysis of the calculations showed that the K-1000-60/1500 and K-1000-60/3000 turbines, as objects of automatic control, are almost equally responsive, this feature being better than in TES turbines of identical output. This circumstance is explained by the lesser time constant for the separator-steam superheaters (1.5-2 sec) in comparison with industrial superheaters for the TES units (6-9 sec).

In both cases, overshooting rotor speeds when there is a sudden drop in load is approximately equal for practical purposes, being 5-6 percent. This is significantly less than the acceptable value at which the automatic governor is activated.

As is the case for TES turbines, the cyclical temperature and force effects resulting in fluctuations in the level of maximum stresses and the development of low-cycle fatigue processes in the structural material are the main factors restricting the maneuverability of the turbines under examination. (We do not have in mind the other equipment for the unit.) We made calculations for the temperature fields and the stresses for the housings and rotors of HPC as well as for LPC rotors, based on standard industrial materials.[8]

Since the requirements for steam turbine maneuverability for AES have not been finally formulated and the corresponding sections of engineering requirements for the turbines under examination are significantly different, the more strict requirements formulated for the K-1000-60/3000 turbine were taken as the basis for the comparisons. The stringency of these requirements is characterized by this example in particular: The start-up time for the turbine (from the first impulse of the rotor until full load is reached) after standing idle for 60 hours is 1.5 hours total (it is usually no less than 6-8 hours for turbines of similar output operating on organic fuels).

In spite of the greater high-pressure rotor diameter in the low-speed unit and its slower cool-down rate, the temperature drops and stresses in it are practically the same as in the high-speed variant. This fact is explained by the adoption of a drum design for the K-100-60/1500 turbine's high-pressure rotor, due to which the equivalent wall thickness of its rotor is substantially less than in the integral disc rotor of the K-1000-60/3000 turbine's HPC.

In both variants, in spite of the high start-up speeds, the thermal stresses in the rotors and housings are small, and their range does not exceed 40 $kgf/cm^2$. However, during power drops, the susceptibility of the material to damage increases substantially. Thus, the susceptibility of high-pressure rotors to damage at 3000 rpm ("start-up--8 hours cool-down--stop") does not exceed 17 percent. When start-up is joined with a drop and then an increase in output, the susceptibility to damage increases to 54 and 30 percent for low-speed and high-speed turbines, respectively. The permissible number of "power drop--power increase" cycles (100-30-100 percent) is extremely large, being 46,000 and 41,000 respectively.

Thus the maneuvering characteristics of the turbines under comparison are nearly identical and at a high level.

We calculated the erosion reliability indicators according to methodology developed at the NPO TsKTI. Calculations based on this procedure correspond with known experimental data. Here the characteristic or erosion wear rate was taken as a measure of erosion danger:

$$e = \frac{dh}{dt} = kgv^n,$$

where -- average wear depth; t -- time; g --consumption of dangerous erosive moisture [Rus. 'raskhod erozionno-opasnoy vlagi'] per unit of eroded surface; v -- relative velocity of drop impact on surface of rotor vane on the curvature of the intake edge of the profile; k, n -- experimental constants depending on vane material.

The calculations showed that significantly smaller velocities of moisture impacting with the inlet edge of the rotor vanes occur in the low-speed turbine. This circumstance determines the small relative danger of erosion of their vanes (significantly less, for example, than on the rotor vanes in the last stages of K-300-240 LMZ turbines where no dangerous erosional wear is observed). During the course of the estimated service life, erosional danger for the blading is greater in the high-speed 1000 MW turbine, in spite of more favorable indicators determining the quantity of roughly-dispersed [krupnodispersnaya] moisture. On the whole, one may speak of the qualitatively higher erosion reliability of blading in the low-speed turbine plant. Therefore questions concerning protection of the vanes in the last stages of high-speed turbines from erosion require more attention than for low-speed turbines.

### 3. Weight-Size Indicators and Comparison of Reduced Costs

The weight indicators for the K-1000-60 turbines at 1500 and 3000 rpm are shown in Table 4. A comparison of the low-speed and high-speed turbines intended for operation in a high vacuum ($p_c = 0.04$ kgf/cm$^2$) with 3 (at 1500 rpm) and 4 (at 3000 rpm) LPC (variants I and III) shows that the high-speed turbine is 820 tons (or 27 percent) lighter. This difference is approxinmately preserved when the weights of the separator-steam superheater, condenser and generator are taken into consideration, but the percentage relationship diminishes to 12.5 percent.

Table 4. Comparative Weight Characteristics of 1000 MW Turbines for AES, tons

| Parts of the Unit | K-1000-60/1500 | | K-1000-60/3000 |
|---|---|---|---|
| | HPC + 3 LPC | HPC + 2 LPC | 2 LPC+HPC+ 2 LPC |
| | $P_c = 0.04$ kgf/cm$^2$ | $P_c = 0.055$ kgf/cm$^2$ | $P_c = 0.04/0.055$ kgf/cm$^2$ |
| Turbine | 2990 | 2240 | 2170 |
| Separator-steam super-heater (SPP), single stage | 185x2 = 370 | 185x2 = 370 | 160x4 = 640 |
| Generator with exciter | 723 | 723 | 615 |
| Condenser | 2170 | 2030 | 2050 |
| Turbine generator, total | 6255 | 5364 | 5476 |

It is advisable to reduce the number of LPC to 2 at AES with average annual temperatures of 22°C and higher for the cooling water (at a rated pressure in the condenser $p_c \geqq 0.055$ kgf/cm$^2$) in low-speed turbine plants. This provides a weight reduction of approximately 750 tons in the low-speed turbine (in comparison with the variant with 3 LPC). In these terms, a critical study of a high-speed plant with 3 LPC is also of interest.

The pilot variants of the turbines are approximately equal in size. In its 5-cylinder format, the high-speed plant is 74 meters long and 17.4 meters wide, the low-speed 4-cylinder plant is 73 meters long and 19.6 meters wide.

Calculations of the technical and economic efficiency[9,10] have shown that, independent of the water supply system, using low-speed turbines in units at AES with VVER-1000 reactors provides a savings in the total adjusted costs. However, its value is relatively small, R 600-800 thousand per year. Considering that the indicated values are about 1 percent of the adjusted costs at a 1,000 MW AES, one may consider the turbine variants under examination to be equally efficient for practical purposes.

It must be noted that their differing degree of reliability, which is very difficult to predict, may have a substantial impact on the actual reliability indicators. Thus, the possible reduction in the accident rate indicator by 1 percent may provide savings in the adjusted costs in the amount of R 260-240 thousand per year.

Conclusions

1. It is necessary to produce the K-1000-60/1500 and K-1000-60/3000 turbines at the same time in the near future. In this case, the low-speed turbines with 3 LPC are preferable for AES with an average annual water temperature up to 20°C, whereas the high-speed turbines and the K-1000-60/1500 plants with 2 LPC are advisable for AES with temperatures above 20°C.

Considering the practically identical efficiency of the specified turbines and the preparedness of the "Khar'kov Turbine Plant" PA to produce the low-speed turbines and of the "Leningrad Metal Plant" PA to produce the high-speed turbines, it is advisable to use the production facilities of each of the plants with their present specialization.

2. In connection with the tendency of the average annual temperatures of the cooling water to increase, it is important to undertake production of the low-speed K-1000-60/1500 turbines with 2 LPC, installing the pilot prototype of such a plant in the first power units of the Zaporozhskaya or Khmel'nitskaya AES with the aim of reducing metal costs and labor intensity of manufacture and reducing installation and servicing operations. The savings in metal is about 750 tons for a single turbine (as compared to the variant with 3 LPC), and the adjusted cost, given average annual cooling water temperatures of 22°C and higher, will be approximately identical. During operation of the AES turbines under partial loads, this modification will provide supplementary advantages.

Introducing turbine units with a reduced number of LPC into nuclear power engineering will also insure an increase in reliability, maintainability and reductions in building costs and transport expenses.

3. The experience of domestic and world power machine building attests to the tendency toward increasing the cut-off output at which point turbines running at 1500 and 3000 rpm are equally efficient. The need to concentrate efforts on the critical analysis of a high-speed variant of the K-1000-60 (68)/3000 variant follows from this experience. This variant may be left as the sole type of 1000 MW turbine in the XII

Five-Year Plan after testing the main engineering decisions and accumulation of operating experience with the series designs for high-speed K-1000-60 (68)/3000 turbines at electric power stations. In this case, the low-speed plants will remain preferable for turbines with large unit outputs (1500, 2000 MW).

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## ENERGY CONSERVATION

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# BASIC PROBLEMS IN ENHANCING EFFICIENCY AND RELIABILITY OF HEAT SUPPLY TO THE NATIONAL ECONOMY

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[Article by V. P. Korytnikov, VNIPIenergoprom]

[Text] One of the tremendous national economic problems is the development of a long-term goal program of efficient development of heat supply for the national economy, which now consumes about a third of the fuel in the land.

Much attention has been focused on this problem in previous decades. More than 50 years ago our country set a course for centralization of heat supply which was based on district heating. The relative significance of centralized heat supply is now about 55 percent of the overall heat consumption in the USSR including 34 percent of it from TETs. The relative significance of TETs heat supply in cities is 42 percent.

In 1980, centralized sources will meet about 70 percent of the urban demand (based on industry) for heat, and the fuel for this will only constitute about 63 percent of the total fuel consumed for heat supply for cities. In the current year alone, 60,000,000 to 70,000,000 tons of conventional fuel will be saved due to centralized heat supply from TETs and large heat-generating boiler plants.

In the studies carried out by institutes of the USSR Ministry of Energy and the USSR Academy of Sciences, it was proven that the share of heat production from centralized sources for cities and industrial enterprises could come to 80 or 85 percent, not a mere 70 percent. In recalculating for the level of energy demand in the 10th Five-Year Plan, this would provide additional fuel economy and release hundreds of thousands of maintenance personnel.

Thus further centralization of heat supply from TETs and large boiler plants should form the basis of energy policy and for the long-term prospect in development of heat supply for the national economy. An inseparable aspect of centralization is also the reliability of heat supply. We must achieve a position where the rate of development of centralized heat supply does not lag behind the rate of growth of the concentration of heat loads, as happened in the 10th Five-Year Plan, but outstrips it.

Special technical and economic calculations showed the following:

In regions of Siberia and the Far East, where coal will henceforth be the primary fuel for thermal energy, TETs are absolutely efficient with thermal loads of more than 580-930 MW, and large boiler plants are suitable for smaller loads. Centralization of heat supply based on organic fuel can achieve a level of 85 to 90 percent.

In regions of Central Asia where the climate is relatively mild and the primary fuel will henceforth be natural gas, TETS should be built for thermal loads of 930 to 1160 MW; and for smaller loads—large, medium and even small highly-mechanized boiler plants as well as heat pumps and solar energy devices. An efficient level of centralization here could reach 80 to 85 percent.

The situation is different in the European part of the country. The remoteness of fuel bases from the sites of primary fuel consumption and the advisability of using petroelum in the national economy, primarily as motor fuel and raw material for the chemical industry, do not make it possible to predict a significant increase in the use of organic fuel in the European regions of the country for heat production purposes. This all predetermines the effectiveness of using nuclear energy in these regions, not only to produce electrical energy but also for the needs of heat supply.

The enormous sluggishness of energy management, related to planning and construction of new power plants, will not support the wide utilization of nuclear energy for heat supply for another 8-10 years. Furthermore, the basis of heat supply in the European part of the USSR in the 1980s will remain organic fuel sources. In view of what has been said, one of the major problems for the 1980s in heat supply of the Euorepan regions of the country is the increased efficiency of traditional organic fuel sources in addition to the involvement of nuclear energy.

This problem can only be resolved by increasing the economy of operating TETs and further development of district heat in conjunction with optimum scales of construction of large boiler plants.

The following data for 1978 make it possible to assess the reserves for enhancing economy of operating TETs of the USSR Ministry of Energy.

The average annual relative consumption of conventional fuel to produce one kilowatt-hour of electrical energy for all TETS was 265 grams; 44 TETs (24 percent) consumed less than 220 grams and 88 TETs (45 percent) consumed less than 250 grams.

Analsyis of the operating efficiency of high pressure TETs (13 and 9 MPa) which produced 223.4 TW-hour (91 percent of the output of all TETs of USSR Minenergo) in 1978 show that when turbine steam recovery is recharged to technically feasible values, the relative consumption of conventional fuel can be decreased by about 18 grams of conventional fuel per kilowatt-hour. In this case the mean relative consumption of fuel in this group of TETs could come to 246-247 grams of conv. fuel/kW-hr), while the total fuel economy due to district heating from Minenergo TETS in 1979 could be about 37,000,000 tons of conv. fuel, instead of 33,000,000 tons (11 percent increase).

As concerns the question of further increase in capacities for TETs, we feel there exists an unjustified point of view according to which, with the noted scales of construction of nuclear condensation electric power plants (AKES) in the European part of the country it is inefficient to consume scarce organic fuel to produce electric energy at TETs and the entire increase in thermal loads should be handled by boiler plants.

In counterbalance to this point of view we can cite the following notions based on technical and economic calculations and comparisons:

• the noted volumes of capacities in AKES, GES and GAES, and completion of construction of the Ekibastuz-Tsentr direct current electric power transmission line will not meet the increasing needs for electrical energy in the Euroepan part of the USSR at the 1990 level;

• with a concentration of thermal loads of 700 to 815 MW, TETs is more economical than a separate power supply circuit, either in the KES + boiler plant version or the AKES + organic fuel boiler plant version.

• under actual organizational conditions of heat supply development, if TETs are not constructed there will inevitably be small and tiny boiler plants constructed having a relative consumption of conventional fuel of 48 to 58 kg/GW versus 41 to 42 kg/GW with the version combining TETs with large boiler plants;

• with wise, economically sound combination of TETs construction with large boiler plants and change in the technical profile of new TETs, the additional consumption of fuel for production of electricity will either be negligibly small as compared to the overall inevitable delivery of fuel to European Russia, or may be completely excluded. In the latter case, the introduction of new capacities in TETs (in addition to the planned one in accordance with an optimum structure of electrical power production) is economically correctly implemented with simultaneous displacement of generation of electricity using obsolete condensation electric power plants (KES) which use low-economy turbine units of 200 MW or lower capacity (in the near future this will also concern 300 MW power plants). This policy will ensure a considerable reduction of mean relative consumption of fuel for generation of electricity in coming years, will raise the capacity reserve in power systems and will prepare a timely renewal of the pool of energy capacities of thermal power plants.

New TETs should be constructed as base-maneuvrable, i.e., with flexible modalities, able to operate in the basic part of the schedule of electrical loads as well as in the semi-peak portion. By solving the problem in this manner, we can theoretically avoid construction of special semi-peak load KES, whose relative consumption of conventional fuel is 120 to 150 g/KW-hr higher than in the version of a TETs operating in the maneuvrable mode.

Thus in the European part of the country in the coming decade it is necessary to construct base-maneuvrable TETs with a total capacity of 8 to 10 GW and implement clearer, more harmonious planning of interrelated development of TETs, boiler plants and thermal networks.

The concentration of thermal loads becomes so great that favorable conditions are created for the development of standardized systems of centralized heat supply of industrial-urban agglomerates based on large suburban TETs and nuclear TETs (ATETs).

Nuclear district heating will become a natural extension and evolution of district heating based on organic fuel TETs.

The cautious attitude toward fuel resources should not lessen even with a transition to heat supply from nuclear energy installations.

In the table are cited comparative expenditures and a characterization is given of diverse versions of centralized heat supply in terms of consumption of fuel (in conventional calculation) from the following sources:

- ATETs using two VVER-1000 reactors with four TK-500-60 turbines (peak heat sources using organic fuel);
- nuclear "boiler"--an atomic heat supply plant (AST) with four heating reactors of 500 MW each (peak sources of heat using organic fuel);
- TETs using organic fuel including five type T-250-240 turbines using coal and peak water-heating boilers;
- organic fuel boiler plants (OK) of unit capacity 350-580 MW.

The energy balance of versions can be achieved using AES with VVER-1000 reactors.

| Expenditures | ATETs | AST+AES | TETs+AES | OK+AES |
|---|---|---|---|---|
| Effective expenditures (0.12K + I), % | 100 | 124 | 120 | 129 |
| Fuel consumption (organic+nuclear), percent | 100 | 120 | 87 | 113 |

As the table shows, it is most efficient to use fuel in TETs. But TETs are less than ATETs in terms of expenditures and, what is particularly important in this period, nuclear district heating displaces organic fuel from the fuel balance by reducing the need to transport it from the eastern regions of the country.

At the present time work is completed on technical and economic justification of the advisability of construction of ATETs for Odessa which confirms the validity of the chosen direction.

Work on justifying installation of ATETs in other cities and agglomerates must be maximally accelerated so that even in the coming decade some six to ten GW can be put into operation. This will serve as a good base for working out all scientific and technical problems of development of nuclear district heating in the 1990s.

A nuclear heat supply plant (AST) is much inferior to ATETs in terms of efficiency (see table). In characterizing AST, reference is often made to the possibility of situating them near consumers with relative simplicity and low cost of the reactor. But, as critical analysis has shown, for some facilities the extent of transit heat lines to peak heat sources and consumers in the AST version as compared to the ATETs version will be nearly identical if the latter are equipped with type VK reactors.

In the author's opinion, it would advantageous to speed up elaboration and creation of one all-purpose type of reactor for nuclear plants of communal and general as well as industrial heat supply (ASPT) which could supply consumers with heat in the form of steam and hot water in nearly any proportion. A nuclear heat source of this type will not compete with ATETs, but will organically supplement it by meeting the need for thermal energy of cities and industrial centers where other sources, including ATETs, can not be constructed for some reason.

A great deal of research has already been done to create a reactor for ASPT with participation of VNIPIenergoprom.

Calculations show that a properly balanced development of ATETs, AST, and organic fuel boilers makes it possible to solve the problem of heat supply for consumers beginning in the 1990s without increasing delivery of organic fuel to European Russia with a simultaneous reduction in the use of liquid fuel for heat supply. The use of nuclear sources for production of thermal energy will roughly double the possibilities of involving nuclear fuel for energy supply.

In summarizing the above, we can assert that in European Russia the construction of organic fuel TETs is justified until such time as the possibilities of constructing base TETs are exhausted according to the conditions of the balance of introduction of new capacities in AKES, GES, GAES and TES, and displacement by base-maneuvrable TETs of low-economy KES; and until the prime growth in thermal loads can be taken on by ATETs in combination with ASPT (and partially AES by organization of vapor take-off for heat supply of adjacent consumers).

This concept forms the basis of the project of the Program of development of centralized heat supply in the European part of the country for the 11th and 12th Five-Year Plans developed by VNIPIenergoprom with participation of other institutes of USSR Minenergo USSR and the USSR Academy of Sciences, and approved by the NTS of the USSR Minenergo.

Any concepts of development of the energy balance of our country must consider the problem of conserving and effectively using fuel and energy which is becoming comparable to the problem of expanding exploration, because simultaneous expenditures for conserving energy are much lower or comparable with expenditures for an increase in production of an equivalent amount of energy.

Substantive possibilities for increasing the effectiveness of use and savings of natural fuel and energy resources exist in stages of production, transportation and end use of fuel and energy.

Operating practice of VNIPIenergoprom shows that these problems can be optimally solved with a comprehensive study and accounting not only of energy supply, but also energy utilization of large industrial centers and residential consumers. The comprehensive approach in a regional cross section will make it possible to cover a range of problems such as creation of highly efficient interrelated enery and technological schemes of industrial enterprises, efficient utilization of secondary energy resources and low-potential waste of heat, increased reliability of energy supply of consumers and improved modes of energy consumption, provision of a choice of an optimum structure of equipment for energy supply sources and successive improvement of energy-consuming equipment, purification of effluent and utilization of production waste, etc.

Practice of comprehensive consideration of problems of energy supply and energy consumption of industrial centers is the first step in transition to analysis of energy management on the scale of industrial and urban agglomerates, and in the future, for regions of the country.

Large reserves in economy of thermal energy exist in thermal management of the residential and domestic sector of cities. Examinations conducted in 1979 by VNIPIenergoprom in cooperation with the industrial association "Soyuztekhenergo" and other organizations of the technical level and state of preparedness for reliable and operation which is economical in terms of consumption of energy resources of systems of centralized heat supply in several regions of Moscow, Leningrad, Kiev, Minsk, Baku on the whole showed that the personnel of heat supply systems did a great deal of work on preparing for trouble-free, high quality operation of heat sources and thermal networks. All inspected regions were systematically carrying out specified measures to save fuel (0.5 to 0.7 percent of the overall consumption of fuel according to heat source) and generally have satisfactory preparedness for production and transportation of thermal energy in the desired amounts.

Analysis of the results of inspections permitted the establishment of the basic shortcomings typical of nearly all systems which reduce reliability, quality and economy of operation of the heat supply systems. Their elimination would provide considerable savings of thermal and fuel resources. The basic shortcomings are as follows:

Exterior corrosion of pipelines of thermal networks. Statistics show that 85 to 90 percent of emergencies and damage in thermal networks occurs for this reason. Anticorrosive coatings of pipes used in construction of thermal networks are not long-lasting (often are missing completely because of their scarcity). The quest for a new reliable anticorrosive coating should be considered a major problem. In coming years maximum pressure must be exerted on construction of enterprises for enameling pipes which is now the most reliable and verified method of protecting pipes against corrosion.

The unsatisfactory quality of thermal insulation of thermal networks. The absolute majority of thermal insulation constructions employed are easily wetted (and because of this thermal losses are higher than calculated) and not durable. A wider and more purposeful quest for quality thermal insulation constructions is required, first of all for channel-less seals. Constructions based on polymer concretes developed at VNIPIenergoprom and undergoing testing in an experimental section are promising.

The lack of comprehensive alignment of the heat supply system which results in exaggerated consumption of network water and accordingly, electric energy to pump it, deregulation of the system, overconsumption of heat and discomfort of consumers and incomplete utilization of the most economical equipment of heat sources. As result there is a considerable overconsumption of fuel. According to the assessment of specialists, quality and timely alignment of systems provides a reduction in the annual consumption of fuel for heat supply by seven to fifteen percent.

Underconfiguration with devices and regulators of thermal points of consumers which nearly eliminates the opportunity for personnel to control optimum conditions and predetermines overconsumptions of fuel of three to four percent per annum.

Unsatisfactory construction and status of heat exchangers of hot water supply at thermal consumer points (in sealed heat supply systems).

Untimely cleaning of preheaters.

The problem of scientists and engineers in coming years is to find a solution to the problems noted above and thereby reduce fuel consumption by two to three percent in centralized heat supply systems.

One more fundamental shortcoming of the country's thermal management should be discussed. In nearly all systems of urban heat supply, separate links (heat sources, heat conduits, thermal networks, consumer thermal points) are administered by different organizations: TETs and main conduits are under the jurisdiction of USSR Ministry of Energy; distributive thermal networks under the jurisdiction of executive policy committees of urban councils; thermal points and internal heat-consuming systems of clients: under the jurisdiction of RZhU and other organizations according to agency affiliation.

The construction of heat supply systems is being done by hundreds of various organizations; the degree of industriality of construction is low; constructions, pipes, thermal insulating materials often do not come up to standard.

Straightening out of the country's thermal management, its concentration in the hands of a limited number of agencies may yield a great economic effect for the state.


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