ASSESSMENT OF TURBINE GENERATOR
TECHNOLOGY

FOR

DISTRICT HEATING APPLICATIONS

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ABSTRACT

Steam turbines for cogeneration plants may carry a combination of industrial, space heating, cooling and domestic hot water loads. These loads are hourly, weekly, and seasonally irregular and require turbines of special design to meet the load duration curve, while generating electric power.

Design features and performance characteristics of large cogeneration turbine units for combined electric generation and district heat supply are presented.

Different modes of operation of the cogeneration turbine under variable load conditions are discussed in conjunction with a heat load duration curve for urban heat supply.

The performance of the 250 MW district heating turbine as applied to meet the heat load duration curve for Minneapolis-St. Paul area is analyzed, and associated fuel savings are estimated.

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SUMMARY

In a typical thermal power plant, almost two-thirds of the energy released from fuel burning is rejected in the condenser in the form of low temperature water which is not effectively utilized. The possibility of modifying the plant turbine cycle to provide utilization of this readily available energy by consumers requiring lower temperature heat, offers significant fuel savings, and, therefore, merits serious investigation. The effective use of rejected steam or water from a power plant at lower temperatures can be achieved in a district heating system, a concept which was developed years ago and is widely used in Europe. In this system, low pressure steam extracted from the power plant after expansion through turbine stages, is used to heat water supplied for space heating as well as domestic hot water to customers. In this way, cogeneration power plants serve the dual-purpose of providing both centralized heat and electric power. This type of station not only saves fuel, but makes a significant contribution to the reduction of air pollution when compared with individual heating systems of homes and commercial buildings.

Productive use of waste heat by district heating systems in the U.S. has been stressed in President Carter's National Energy Plan, and the Department of Energy is conducting an extensive program leading to the possible construction of a large demonstration district heating system.

The major component of a cogeneration power station is the turbine plant which is capable of supplying the

electrical and district heating loads. The district heating turbines have a number of specific features, which differ from conventional turbines. The intent of this study, therefore, is to assess the unique design features and performance characteristics of large turbine units for district heating applications, with particular emphasis on the large units manufactured in the USSR, where district heating has been extensively developed.

The basic parameters of large district heating turbines manufactured in the USSR are presented in the following table:

<u>Description</u>	<u>TYPE OF TURBINE</u>		
	<u>T-250/300-240</u>	<u>T-175/210-130</u>	<u>T-100/120-130</u>
1. Electric Capacity, MWe			
Nominal	250	175	110
Maximum	300	210	210
2. District Heating Load, MWt			
Nominal	384	314	203
Maximum	384	326	214
3. Nominal Steam Parameters			
Throttle Pressure			
psig	3400	1834	1834
Superheat Temperature			
°F	1040	1031	1031
Reheat Temperature			
°F	1049	-	-
4. District Heating Extraction			
Pressure Range, psia			
Upper Stage	8.5 to 38.5	8.5 to 42.7	8.5 to 35.6
Lower Stage	7.1 to 21.3	7.1 to 28.5	7.1 to 28.5

The above listed district heating turbines are equipped with variable control diaphragms, located in the low pressure element.

The following broad principles have been applied during the development of this type district heating turbines:

1. High capacity district heating turbines have incorporated those design features which have proved reliable and efficient in improving the heat rate on lower capacity machines for many years. Some of these features are as follows:
 - Heating of the district water in series with steam extracted from two space heating extractions.
 - Elimination of throttling of the extraction steam by appropriate widening of the controlled pressure range.
 - Utilization of steam entering the condenser.
 - Design of the turbine blade system taking into account combined operation of the turbine and district heating network.
2. Capability of generating electric power of higher than normal value, not only in partial heat load modes but also in power only modes.
3. Turbines of large capacity in spite of the differences in the initial steam parameters, types and loads, have the same or similar construction features and are standardized wherever possible.

4. The major steam parameters of the turbine are selected, taking into account the existing boiler, electrical, and auxiliary equipment for conventional condensing power plants. In this way the development time for such district heating power plant units is reduced since only the turbine is a new item.

The maximum effect of cogeneration is achieved when the turbine is designed to match the heat consumption which varies substantially during the year. The turbines under discussion have the capability to follow closely a variable heat load. The required amount of heat is provided to the customer by varying the temperature in the supply network line at the power plant as a function of the outdoor ambient temperature. To provide the required network temperature the pressure of the steam extracted for district heating purposes changes correspondingly.

The control system of the turbine makes it possible for the turbine to operate in the following three modes:

1. Power only mode without district heating steam extraction
2. Condensing operation with district heating steam extraction
3. Back pressure operational mode.

The specific features of district heating turbines discussed in the report may be summarized as follows:

- a) Have suitable openings in the turbine cylinders for extraction of large amounts of low pressure steam.

b) Capability to control concurrently and independently the electrical load and the steam extractions over a wide range of combined electrical and thermal load demands.

c) Capability to operate in the condensing (power only) mode.

d) Heating in stages, of the district water, in series extraction heat exchangers.

e) Minimum throttling of extraction steam. This ensures optimal operation of the bleed system over a wide range of parameters.

f) Maximum utilization of the steam entering into the condenser.

To demonstrate the advantages of district heating the possibility of utilizing the T-250/300-240 turbine design for the Minneapolis-St. Paul district heating system has been analyzed and associated fuel savings have been evaluated. To assess the fuel savings, the annual fuel consumption of the proposed cogeneration power plant and peaking boilers have been compared with the fuel consumption of the conventional power plants providing electricity only and the individual boilers providing heat only. The annual fuel savings attained amount to about 4.2 million barrels of oil or 1.6 million tons of western coal.

I. INTRODUCTION

Combined generation of heat and power offers one of the major fuel conservation opportunities and a significant reduction of air pollution, when compared with separate electrical and heat production. In a cogeneration power plant, a substantial part of the throttle steam is expanded in the turbine to generate electricity before the steam is extracted at a pressure and temperature which can supply the required thermal loads. The extracted steam may be directly used as a heating medium and supplied by a piping system to the customers. Another method is to condense the extracted steam in closed-type heat exchangers at the power plant, thereby producing hot water, which can then be pumped to customers. Hot water as a heating medium offers a number of advantages in comparison to steam, and therefore is more widely used in large district heating systems.

Most of the existing power plants in the U.S. are of the single purpose, condensing type, and are not designed for the extraction of the large amounts of steam that are required to supply a thermal load.

Extracting additional heat from an existing turbine involves some reduction in electrical generation. The electricity lost depends directly on the extracted steam flowrate, pressure, and number of extractions.

When considering the conversion of existing turbines for cogeneration operation, the possibility of extracting up to 15 to 20 percent of the throttle steam flow from the crossover point of the turbine could be considered. An additional pressure control system may have to be installed at the crossover pipe to provide reliable operation of the turbine.

It has to be emphasized that the retrofitting of existing turbines to extract the required steam flow from the low pressure element is very difficult and involves a significant redesign of the turbine and its control system, and consequently, a substantial outage time may be necessary for modification.

One of the major differences also, between an existing turbine and a specially designed cogeneration turbine is that in the former, all the low pressure extraction piping is located inside the condenser. For modification of existing turbines, provision must be made for routing this enlarged diameter extraction piping out of the condenser.

To avoid the problems discussed above, different types of cogeneration turbines have been developed and are commercially available. The existing cogeneration turbine designs utilized for district heating can be categorized according to the following features:

1. With one controlled space heating extraction
2. With two controlled space heating extractions
3. With two controlled extractions, industrial and space heating
4. With controlled industrial and two space heating extractions
5. With three controlled extractions - one industrial and two space heating

The cogeneration turbine designs listed above may also be of the back-pressure or condensing type. Back-pressure turbines have a rigid relationship between the heat and electric loads, while the extraction turbines equipped with a condensing tail have the flexibility of changing the ratio of the heat load to electric output.

The major objectives of this study are as follows:

1. Search and analyze available information in order to assess design features and performance characteristics of large turbine units for district heating applications with special emphasis on large units manufactured in the USSR. Parameters to be given prime consideration are:
 - . Performance characteristics of variable diaphragm type turbines specially designed for district heating applications.

- . Extraction pressure ranges available for district heating.
 - . Relationship between turbine extraction steam flows and pressures, and the district heating system loads.
 - . Turbine control systems.
 - . Design features specifically related to district heating application.
2. Analyze performance of a selected large type turbine unit operating under variable load conditions as required to meet the heat load duration curve developed by the Department of Energy with assistance from AB Atomenergi of Sweden for the Minneapolis area.

II. DESIGN FEATURES AND PERFORMANCE CHARACTERISTICS OF DISTRICT HEATING TURBINES

II.1 Control of the District Heating Load

The maximum effect of cogeneration is achieved when the turbine is designed to match the heat consumption which varies substantially during the year. The capability of the turbine design to follow closely a variable heat load is very important in achieving high performance characteristics.

A typical space heating and domestic hot water load duration curve for an urban district is presented in Fig. 1. When the outdoor temperature increases the load decreases and becomes almost a constant value in the summer for only domestic hot water supply.*

Different methods of controlling the heat supply in district heating systems have been developed. The simplest and most widely used is the central temperature control system. In this case, the required amount of heat is provided to the customer by varying the temperature in the supply network line at the power plant as a function of the outdoor ambient temperature. This means that the flowrate of water circulated in the district heating system is almost constant during the heating period.

*Supply of cooling load is not discussed in this paper.

FIGURE 1.

A TYPICAL LOAD DURATION CURVE FOR SPACE HEATING AND DOMESTIC HOT WATER SUPPLY

*Cogeneration coefficient is the ratio of heat load supplied by the cogeneration turbine to the maximum peak demand.

The maximum design temperature of the district heating water is determined by optimization analysis. By increasing the supply temperature in the network, the cost of the piping system and the pumping power required are reduced. However, as a result, the extraction pressure at the turbine is increased and the electrical generation reduced. Therefore a cogeneration turbine is usually designed for base load operation and does not supply the peak load. The optimal cogeneration coefficient for a district heating system is determined by economic trade off studies.

The type of heat load control and the cogeneration coefficient determine the temperature of the hot water supply. In Fig. 2 the dependence of the temperatures in the district heating system on the outdoor temperature is presented.

It is evident that to provide the required network temperature the pressure of the steam extracted for district heating purposes should change correspondingly. Therefore the cogeneration turbine must be designed to closely follow the district heating system loads. In addition to the district heating modes, the cogeneration turbine will operate also in a pure condensing (power only) mode. Therefore the turbine should provide

FIGURE 2.

DEPENDENCE OF THE DISTRICT HEATING SUPPLY (1) AND RETURN (2) WATER TEMPERATURES ON THE OUTDOOR TEMPERATURE FOR THE 100 MW DISTRICT HEATING TURBINE. (3) DISTRICT HEATING SUPPLY TEMPERATURE PROVIDED BY THE TURBINE HEAT EXCHANGERS.

the optimal economical operation during the year, taking into account both cogeneration and power only modes.

The specific features of cogeneration turbines for district heating applications may be summarized as follows:

- a) Have suitable openings in the turbine cylinders for extraction of large amounts of low pressure steam.

- b) Capability to control concurrently and independently the electrical load and the steam extractions over a wide range of combined electrical and thermal load demands.

- c) Capability to operate in the condensing mode.

- d) Heating in stages, of the district water, in series extraction heat exchangers.

- e) Minimum throttling of extraction steam. This ensures optimal operation of the bleed system over a wide range of parameters.

- f) Maximum utilization of the steam entering into the condenser.

The possibilities for unit size increase are more limited for district heating turbines as compared with single-purpose condensing type. This is mainly because the size of a district turbine is determined by the

heat load density of the area to be served. Therefore, the largest district heating units do not exceed 300 MW of electric output at present time. Table 1 lists characteristics of the largest district heating turbines now in operation.

In the following sections design features and performance characteristics of large cogeneration turbines for district heating now in operation are presented. Most of the turbines discussed are variable diaphragm control type machines manufactured in the USSR. Information available at the present time on West European district heating turbines is also presented.

II.2. Variable Diaphragm Control Type District Heating Turbines

The following broad principles have been applied during the development of this type of district heating turbine:

1. High capacity district heating turbines have incorporated those design features which have proved reliable and efficient in improving the heat rate on lower capacity machines for many years. Some of these features are:
 - Heating of the district water in series with steam extracted from two space heating extractions.
 - Elimination of throttling of the extraction steam by appropriate widening of the controlled pressure range.
 - Utilization of steam entering the condenser.
 - Design of the turbine blade system taking into account combined operation of the turbine and district heating network.

CHARACTERISTICS OF THE LARGEST DISTRICT HEATING TURBINES

Table 1

No.	SUPPLIER	INSTALLATION	CITY	COUNTRY	MAXIMUM ELECTRIC OUTPUT, MWe	MAXIMUM HEAT OUTPUT, MWt	STEAM CONDITION Paig/op/op	NUMBER OF HEATING STAGES	SUPPLY AND RETURN TEMPERATURE AT THE TURBINE DISTRICT HEAT EXCHANGERS	YEAR COMMISSIONED
1	UTMZ	9 units in operation: TE2 21, TE2 22	Moscow	USSR	300	304	3400/1040/1049	3	158 to 230 95 to 120	1972
2	BBC-Stal	Vartan	Stockholm	Sweden	270 ¹	384	2566/995/995	2	194/122	1976
3	BBC-Stal	Aros II	Västerås	"	270 ¹	372	2566/995/995	2	194/122	1973
4	KWU	Aros I	Västerås	"	265 ¹	372	2566/995/995	2	194/122	1969
5	KWU	Fyrås	Uppsala	"	197	349	2566/995/995 ²	2	185/131	1973
6	KWU	Bravalla	Morrköping	"	250	337	2566/995/995	2	194/122	1972
7	BBC	Fyn V	Odense	Denmark	285	326	2566/986/986	2	185/113	1974
8	UTMZ			USSR	210	326	1824/1031	3	158 to 262 95 to 120	1978
9	BBC	Nordkraft	Alborg	Denmark	285	267	2566/986/986	1	185/113	1973
10	Stal	Fyn IV	Odense	Denmark	205	262	2566/995/995	2	185/113	1968
11	Stal	Orebro	Orebro	Sweden	115 ¹	244	1986/995 ²	2	194/131	1974
12	KWU	Lichterfelde I	Berlin	Germany	150	233	2624/986/986	1	189/117	1972
13	KWU	Lichterfelde II	Berlin	Germany	150	233	2624/986/986	1	189/117	1973
14	KWU	Lichterfelde III	Berlin	Germany	150	233	2624/986/986	1	189/117	1974
15	UTMZ	Several units		USSR	120	214	1834/1031	3	158 to 257 95 to 120	1967
16	BBC	Esbjerg V	Esbjerg	Denmark	256	192	2276/1004/1004	2	185/126	1969
17	Stal	Heleneholm	Malmo	Sweden	95	192	1493/986 ²	1	194/140	1971
18	UTMZ	Several units	----	USSR	165	163	1834/1049 ³	2	158 to 257 95 to 120	1973

FOOTNOTES:

1. High-pressure feedheater shut down
2. Back-pressure turbine
3. Turbine has one industrial and two space heating extractions

BBC = Brown Boveri, Switzerland
 KWU = Kraftwerk Union AG, Germany
 Stal = Stal-Laval, Sweden (Licenses for BBC steam turbines above 200 MW since 1969)
 UTMZ = Ural Turbo-Machinery Factory, USSR

- Providing additional non-controlled extractions for heat supply which eliminates the throttling of high pressure steam.
- 2. Capability of generating electric power of higher than normal value, not only in partial heat load modes but also in power only modes. This permits a reduction in the power generated by the system less economic conventional power plants during the non-heating period of the year. The increased electrical generation may be utilized also to cover the peaks and provide a reserve in the electric system.
- 3. Turbines of large capacity in spite of the differences in the initial steam parameters, types and loads, have the same or similar construction features and are standardized wherever possible.
- 4. The major steam parameters of the turbine are selected, taking into account the existing boiler, electrical, and auxiliary equipment for conventional condensing power plants. In this way the development time for such district heating power plant units is reduced since only the turbine is a new item. For example, the turbine T-250/300 is designed for operation with the boiler, generator, and feed water heaters used in the 300 MW conventional units.

In the following section the largest district heating turbines manufactured by Ural Turbo-Motor Factory of the USSR are discussed in more detail. The basic parameters of the turbines are presented in Tables 2 and 3.

BASIC PARAMETERS OF LARGE DISTRICT HEATING TURBINES

TABLE 2

MANUFACTURED BY URAL TURBO-MOTOR FACTORY, USSR

Description	TYPE OF TURBINE				
	<u>T-250/300-240</u>	<u>T-175/210-130</u>	<u>T-100/120-130</u>	<u>PT-135/165-130</u>	<u>R-100-130/15</u>
1. Electric Capacity, MWe					
Nominal	250	175	110	135	100
Maximum	300	210	120	165	107
2. Turbine Speed, RPM	3000	3000	3000	3000	3000
3. Nominal Steam Parameters					
Throttle Pressure, psig	3400	1834	1834	1834	1834
Throttle Temperature, °F	1040	1031	1031	1049	1049
Reheat Temperature, °F	1049	-	-	-	-
4. Throttle Steam Flow, lb/hr					
Nominal	2,106,000	1,643,000	1,058,000	1,630,000	1,676,000
Maximum	2,161,000	1,676,000	1,069,000	1,676,000	1,676,000
5. District Heating Extraction Pressure Range, psia					
Upper Stage	8.5 to 38.5	8.5 to 42.7	8.5 to 35.6	12.8 to 35.6	-
Lower Stage	7.1 to 21.3	7.1 to 28.5	7.1 to 28.5	5.7 to 17.1	-
Industrial Stage	-	-	-	171 to 299	-
6. Heat Load, MWt					
District Heating Nominal	384	314	203	128	-
District Heating Maximum	384	326	214	128 to 163	-
Industrial Nominal	-	-	-	371	-
Industrial Maximum	-	-	-	452	-
7. Back Pressure Range, psia	-	-	-	-	171 to 256
8. Nominal Back Pressure Steam Flow, lb/hr	-	-	-	-	1,433,000
9. Feedwater Temperature, °F	505	450	450	450	453
10. Number of Extractions for the Regenerative Feedwater System	9	7	7	7	3

STEAM PARAMETERS OF THE DISTRICT HEATING
TURBINE REGENERATIVE SYSTEMS

TABLE 3

Number of Feedwater Heater	<u>T-250/300-240-2</u>			<u>T-175/210-130</u>			<u>T-100/120-130-3</u>			<u>PT-135/165-130/15</u>			<u>R-100-130/15</u>		
	P psia	T °F	G lbs/hr	P psia	T °F	G lbs/hr	P psia	T °F	G lbs/hr	P psia	T °F	G lbs/hr	P psia	T °F	G lbs/hr
LPH 1	-	-	-	-	-	-	5.4	-	1,323	3.0	-	-	-	-	-
LPH 2	4.0	-	-	14.2	-	28,665	24.2	-	15,435	11.4	-	16,317	-	-	-
LPH 3	13.5	275	38,147	41.2	-	72,765	42.7	271	48,951	35.6	261	58,212	-	-	-
LPH 4	40.5	446	85,995	84.6	365	69,678	82.5	374	38,808	71.1	361	63,945	-	-	-
LPH 5	81.1	644	69,458	-	-	-	-	-	-	-	-	-	-	-	-
Deaerator	145.1	979	33,957	213.4	511	18,963	176.4	511	14,553	213.4	543	34,398	-	-	-
HPH 1	246.1	815	77,837	213.4	531	84,231	176.4	511	37,265	213.4	543	82,026	213.4	543	84,231
HPH 2	590.3	572	207,050	324.3	617	71,442	330.0	639	61,299	318.6	630	61,740	328.6	635	68,355
HPH 3	836.4	653	113,117	482.2	707	74,529	480.8	714	42,998	476.5	721	72,545	485.0	725	74,970

Footnotes:

1. LPH and HPH = low and high pressure heaters respectively
2. P and T = steam pressure and temperature in the extraction chamber respectively
3. G = Extraction steam flow rate

II.2.1. Turbine T-250/300-240

II.2.1.A. Design features and Performance Characteristics

The turbine has a nominal electric capacity of 250 MWe and nominal heat capacity of 384 MWt. The nominal electric capacity is the load which the turbine generates at the nominal heat load, initial steam condition, and extraction, feedwater and condenser parameters. The maximum electric capacity of the turbine with the district heat load disconnected is 300 MWe.

The throttle flow for the turbine is 2,106,000 lbs/hr with steam parameters of 3400 psig throttle pressure and 1040H superheat temperature and 1049F reheat temperature. A longitudinal section of the machine is presented in Fig. 3.

The turbine is tandem compound with four cylinders: high pressure (HP), intermediate pressure No.1 (IP1), and No.2 (IP2), and low pressure (LP).

In developing designs of district heating turbines, one of the major problems to be solved is that of extracting large volumes of steam for district heating and domestic hot water purposes from the turbine. For this turbine for example, the extraction pipes have an overall cross section area of 91.5 sq. ft. Another major problem is to limit the additional axial load exerted on the end thrust bearing, as compared with condensing turbines. This load arises as a result of the change in pressure and cannot, in practice, be compensated for by a dummy piston. The arrangement of the turbine in four cylinders makes it possible

FIGURE 3. A LONGITUDINAL SECTION OF THE 250 MW TURBINE

to provide the district heating extractions of the required size, while retaining acceptable overall dimensions for each rotor, and ensuring almost full compensation for the additional axial loads occurring within the flow section itself.

Main steam from the boiler is supplied to two valve blocks, located one on each side of the HP element. Each valve block is a combination of one stop and three control valves. The HP cylinder has a double-wall counter-flow arrangement and has 12 stages. From the HP cylinder the steam is supplied to the reheat section of the boiler from which the steam is then directed into IP1 through two valve blocks. Each valve block consists of an intercept and control valves. The IP1 cylinder is of the single flow, single wall type and has 10 stages. From IP1 the steam at about 78.2 psia at nominal load is supplied to IP2 which is a double-flow cylinder with 6 stages in each flow, of which 4 stages are located before the upper district heating extraction stage and two stages between the upper and lower stages.

The double-flow design of the IP2 made it possible to decrease the dimensions of the highly loaded blades in the pre-extraction stages, which operate under variable conditions of pressure drops, mass and volumetric steam flows, which vary in more than one way.

The design of the turbine flow section must ensure a high efficiency over a wide range of operating modes, differing in magnitude of heat load and pressure in the

district heating extraction points. A variable operational mode affects the group of stages preceding the extraction stages. Therefore the IP2 stages of the turbine are designed with allowance for simultaneous operation of turbine and heating network. This ensures maximum effectiveness of combined generation of heat and power. The highest efficiency will be obtained when the design temperature of the heating networks and the optimal cogeneration coefficient are achieved under actual operational conditions. For power plants with T-250 turbines, the optimal value of the cogeneration coefficient is 0.6, based on the system optimization studies.

The variable operational conditions of the IP2 stages are taken into account in developing the profiles of the rotating and stationary blades, which must have stable efficiency over a wide range of variation in inlet angles and Mach numbers.

After IP2 a part of the steam may be supplied to the lower extraction point and the rest enters the LP cylinder via two crossover pipes. The pressure after IP2 is about 11.4 psia in the nominal condensing mode. The low pressure section stages are designed on the basis of the operational conditions of the turbine in the condensing mode. The condensing mode, despite the relatively small contribution of the LP section to the total capacity of the turbine, retains its importance and considerably affects the design of the turbine, its efficiency and costs.

In designing the condensing part of the turbine there are three problems to be considered: (1) development of the LP section stages and sizing the exhaust bleeds; (2) determination of optimal cooling water design temperatures, and (3) determination of the optimum condenser and exhaust hood sizes.

The optimization of the above parameters will result in the determination of the incremental power generating capacity in the condensing mode without district heating load.

The maximum electrical capacity of the turbine described is based on the maximum steam flow per unit and is 300 MW (50 MW increment). The increase in capacity in the condensing operational mode over the district heating mode makes it possible to use the unit maximum capacity in summer as well as during the initial operational period when there may be no heating load. At the same time there is the possibility of covering short time peak power demands in the electric system.

The LP cylinder is double-wall, double-flow, each flow having three stages, including the control diaphragm. The steam supply to the LP cylinder is controlled by a grid type diaphragm. The moving blades of the last stage of the LP cylinder have a length of 37 in. with a mean diameter of 94 in, which corresponds to an area for both exhausts into the condenser of 152 square feet. From both ends of the LP section the steam is exhausted into one condenser with a surface area of 150,000 ft² via two field welded connection.

Pressure in the condenser at rated turbine capacity is between 0.7 and 1.3 psia, depending on cooling water temperature.

The anchor point of the turbine is located on the side frames of the LP exhaust section. With a cross-wise

arrangement of the turbine set, a machine hall with a span of about 175 ft. is required.

The flow diagram for the machine is presented in Figure 4. The district heating water is heated in the gland steam condenser and in two district heat exchangers in series, supplied with steam from the lower and upper space heating extraction points from IP2. The extractions have the following range of controlled pressure: in the upper stage between 8.5 and 28.5 psia, and in the lower stage between 7.1 and 21.3 psia. These heat exchangers are of the horizontal type, each with surface areas of 53,800 sq. ft. both having stainless steel tubes. The condensate after the district heat exchangers is pumped forward into the feedwater system.

The feedwater cycle of the turbine includes 5 low pressure heaters, a deaerator of 85 psia and three high pressure heaters for heating the feedwater to 505F. The low pressure heaters No. 2 and 3 are supplied with steam from the district heating extractions.

The boiler feed pump is turbine driven by steam supplied from the extraction point of the turbine after the 14th stage, with an initial pressure of 327 psig and a backpressure of 91 psig under rated operational conditions. The steam after the boiler feedpump turbine, returns to the main turbine between IP1 and IP2 and is partially extracted to the low pressure heater number 5. The steam parameters of the regenerative system of the turbine are provided in Table 3. This data corresponds to the operational mode of the turbine with nominal heat load, with the pressure

A FLOW DIAGRAM OF THE 250 MW TURBINE

(1) Boiler; (2) HP cylinder; (3) IP1 cylinder; (4) IP2 cylinder; (5) LP cylinder; (6) Condenser; (7) Circulating cooling water bundle; (8) District heating bundle; (9) Main ejector cooler; (10) Cooler of the gland condenser ejector; (11) Low pressure heaters; (12) Deaerator; (13) Boiler feedpump; (14) High pressure heaters; (15) Gland steam condenser; (16) Lower stage district heat exchanger; (17) Upper stage district heat exchanger; (18) Water heating peaking boiler; (19) District heating load; (20) Steam from the gland system

in the controlled extraction of 13.5 psia. The drains from the high pressure heaters are cascaded to the deaerator. The drains from low pressure heater 5 are cascaded to heater 4 and from heater 2 are pumped by means of a drain pump into the low pressure heater 3. From heaters 3 and 4 the drains are pumped into the main condensate line.

The turbine allows steam extraction up to 200,000 lb/hr from the exhaust of the turbine drive for fuel drying, oil heating, and make-up deaeration purposes.

The performance characteristics of a typical district heating turbine are normally presented in a graphic form diagram. The diagram represents the dependence between the throttle steam flow, electrical and heat loads of the turbine, and the district heating water temperatures in various operational modes. The diagram permits the determination of the heat rate performance in different operational modes. The performance diagram is supplied usually by the turbine manufacturer and is based on heat balance calculations.

A number of parameters determine the heat performance of a district heating turbine. In addition to the throttle steam flow parameters, electrical and heat loads, the diagram also includes the pressures in the controlled extractions, (or the temperatures of the district water), the flow and the temperature of the circulating water and other parameters. To simplify the utilization of the diagram the relationship between a limited number of major parameters is presented usually on the main chart.

In addition, supplementary charts are provided for determining the influence of other parameters on turbine performance.

In Figure 5 the dependence between the heat and electric loads for the T-250 turbine is presented based on field performance test data. However, the complete performance diagram of the turbine under discussion is not available at the present time. A performance diagram for a similar turbine but of 100 MW size is discussed in Section II.2.3.B.

Use of supercritical steam parameters, reheat, and improved bleed section design, resulted in a high efficiency for the T-250 turbine. When compared with the 100 MW district heating turbine, the power generation in the heat supply mode, increased by 20-25%, and the heat rate in the condensing mode decreased by 10-12%.

The guaranteed heat rate at rated condensing conditions is less than that for the 150-200 MW condensing units, but 2 to 3% higher than that for the 300 MW condensing units with the same steam parameters. However, even with a heating load of 35 MWt or more, the heat rate for the T-250 turbine decreased to 7143 BTU/KWh or less, which makes it possible for the turbine to compete with conventional condensing units. A heat balance for the T-250 turbine while operating to supply a district heating load at an outdoor temperature condition of 23F, is presented in Fig. 6.

2.1.B Description of Operating Modes

The HP cylinder of the turbine is designed for steam

FIGURE 5.

THE DEPENDENCE BETWEEN THE HEAT AND ELECTRIC LOADS FOR THE T-250 TURBINE

Pressure in the upper extraction stage, PSIA

1 - 28.5; 2 - 24.2; 3 - 19.9; 4 - 14.2; 5 - 8.5 to 11.4

flow in the nominal heating mode, and the low pressure cylinder for the condensing mode. When operating with nominal electrical and heat loads, the steam flow to the LP section is restricted. Therefore, it is possible to increase the electrical generation to more than the nominal value by increasing the steam flow to the LP cylinder and at the same time decreasing the controlled extraction load. The maximum electrical generation in the turbine is achieved when the full steam flow is passing through all sections, HP, IP and LP with the heat load disconnected. For a modern district heating turbine the maximum electrical generation is approximately 110 to 125% higher than nominal and depends on the type of the turbine, steam parameters, controlled extraction pressures and the capacity of the electrical generator. For the T-250 turbine discussed the maximum electrical generation is 20% higher than at the nominal load.

As described above, the turbine has two space heating extractions utilized for heating of the district water in series. The turbine has one pressure regulator for space heating extraction consisting of a grid type diaphragm, located in the LP cylinder. The pressure may be controlled in only one of the two extraction stages: in the upper stage with two operating heat exchangers or in the lower stage with only the lower heat exchanger in operation. It should be noted that there are no control valves in the extraction piping.

Distribution of the heating load between the upper and lower extraction stages is determined by the temperatures

of the district heating water before and after the heat exchangers, the flow rate of the district heating water, the electrical load and cannot be changed without a change in operating mode. The steam flow to the lower extraction stage cannot be reduced below that required for stage cooling purposes. Such a flow condition could obtain when only the upper heater is in operation. To prevent such a mode of operation, the district heat load must be distributed equally between the two extraction stages. To ensure this condition, district heating water must pass through the heaters in series at the same flow. Therefore, the operation of the upper heater only, with the lower out of service is not permitted.

The series heating of the district water is similar in principle to feedwater heating. However, in district heat exchangers the temperature difference between entering and leaving water is relatively small, i.e. 72 to 90F, as compared to 270 to 450F in a regenerative feedwater system. Another feature is that 70 to 80 percent of throttle steam is used for district heating, whereas only 20 to 30 percent is utilized in a regenerative feedwater system. This condition imposes a number of difficulties on the turbine design. Therefore, optimization of the series heating system is more important for the performance characteristics of the turbine when compared with a feedwater heating system of a condensing turbine.

The effectiveness of the series heating depends on a number of factors: number of stages, distribution of the load between stages, the load level, flow rate and temperature of the district heating water, changes in these parameters

during the year, type of heat load control, climatic conditions of the district served, cogeneration coefficient, throttle steam conditions, type of turbine, and auxiliary equipment. By changing from a single to a two-stage district heating system, an increase of between 3 and 5% in electrical output is obtained.

The maximum heat load from both extractions takes place when the control diaphragm in the LP element is closed. In these conditions the pressure in the controlled extraction increases, and the flowrate of the cooling steam into the LP section increases in proportion to the pressure in front of the closed diaphragm. When the extraction pressure decreases, the cooling steam flow also decreases.

If the cogeneration coefficient increases, the pressure in the stages increase accordingly. When the extraction pressure changes the heat load is redistributed between the district heat exchangers and affects the steam flow in the upstream stages. The efficiencies of these upstream stages change accordingly. Substantial changes of the extraction pressures and volumetric flows take place for the upstream stages when the seasonal heating period changes, or the turbine operates in the condensing mode. Taking into account the variable conditions of upstream stages the design of the bleed system of the turbine should provide the maximum effectiveness during an annual period.

The use of steam control devices in the turbine increases the possibility of a pressure increase in the IP and LP cylinders as a result of control system

malfunction. Therefore, the extraction stages are supplied with safety devices. The extraction pipe from the upper stage is also supplied with a nonreturn valve to prevent overspeed of the turbine.

In single purpose turbines there is a proportional relationship between the steam flowrate through the turbine, and the amount of drain and pressure drop between the feedwater heaters. This provides in most of the operation modes a reliable cascading of the drains from one feedwater heater to another.

In the cogeneration turbine such a relationship does not exist. When operating with controlled extraction, the flowrate through the LP cylinders is reduced. However, the throttle flow in the turbine increases. When the extraction stages are operated with small steam flowrates, the pressure difference between the interconnected heaters is not enough to provide the cascading of the drains from one heater to another. Therefore, in this type of turbine the cascading mode of operation is limited and drain pumps are often used.

Another specific feature of the cogeneration turbines is when the turbine is running with high heat load, the flowrate of the steam entering into the condenser is low and therefore there is not enough condensate to cool the gland steam condenser and the ejector. Therefore, in this mode, a recirculation line from the feedwater system is provided back to the condenser.

The cogeneration turbine in the district heating mode

of operation provides steam flow for cooling the LP stages and this flow enters the condenser. The minimum cooling flow requirements are determined by the turbine design features such as type of the LP bleeds, control diaphragm leakage and operation factors (such as pressure in the extraction stage and the condenser.)

The heat of the steam entering into the condenser is transferred to the circulating water and is not utilized in the cycle of the power plant. The cooling water also takes away the heat of the steam entering the heaters located on the recirculation line (gland heaters, and the ejector coolers). This additional heat loss may be comparable with the heat of the minimum steam flow entering the condenser, because the recirculation of the condensate takes place in the modes close to nominal, when the leakages of the steam in the gland heater are a maximum. The heat losses in the condenser in the district heating mode reduce the heat performance of the turbine with a controlled extraction when compared with a back pressure turbine, in which such losses do not exist. However, the district heating turbine with a condensing tail has the advantage that it may be operated with independent electrical and district heating loads, whereas a back pressure turbine electrical generation is directly connected to the heat load. The heat loss in the condenser may be reduced to a minimum or completely eliminated in two ways:

a) by reduction of the steam passing through the LP section and other steam flows dumped into condenser and b) by utilization of the steam entering the condenser in the plant cycle, for example, to heat up the district return water or make-up water.

As can be seen, the steam flow through the LP section of a district heating turbine can vary widely from large flows in the condensing mode to limited flows in the heating mode. Therefore, the capability should be provided of changing from the district heating mode to the condensing mode in which the condenser is cooled by the circulating water. If such a capability is not provided, then the turbine will be, for all practical purposes, a back pressure turbine.

The following features are incorporated in the turbine design to utilize the cooling steam flow:

(1) Installation in the condenser of a separate tube bundle, in which the return or make-up district water may enter and pick-up heat.

(2) Construction of the turbine tail-end with a cooling system capable of working with a low steam flow-rate, elevated vacuum and temperature.

When utilizing the cooling steam flow in the condenser, this flow is obviously limited. In this situation the condenser is cooled only by the return district heating water and the pressure in the condenser is increased, which raises the temperature of the turbine exhaust. When operating in the condensing mode, the circulating cooling water is supplied both to the main and the district heating tube bundles of the condenser. District heating water is not provided in this operating mode.

When the cooling steam flow is being utilized, the grid diaphragm in the low pressure element is closed and the turbine is working in a mode similar to that of backpressure operation, that is, all heat supplied to the turbine except for mechanical, generator and radiation losses, is used for electrical and heat generation. However, the ability of controlling the heat and the electrical loads independently is restricted in this mode.

The design of the condenser with a district heating bundle, having separate water boxes in the same steamspace permits changes of the turbine modes without shutting down the turbine and reducing the capacity.

The utilization of the heat entering the condenser increases the heat load of the turbine by about 3 to 4 percent and allows a reduction in the auxiliary power consumption (by not pumping the cooling circulating water). However, the electric capacity of the turbine may be slightly reduced in this mode.

The overall increase in efficiency of the turbine having heat utilization in the condenser in the heating mode results in a decrease in the annual fuel consumption by about one to one and a half percent.

II.2.1.C Description of the Control System

The control system of the turbine contains two control loops: rotor speed and bleeds (Figure 7). The rotor speed loop is hydraulic. The sensor for this channel is a flowless pulsed pump-impeller, the pressure of which is transmitted to a membrane-band speed controller. This loop also has electric interconnection through the electric motor of the speed controller synchroniser and through an electric-to-hydraulic converter (EHC). Through the first connection, the main steam pressure pulse can be transmitted from the protective high limit pressure controller. A signal is transmitted to the fast acting EHC when the generator is being disconnected from the electric grid.

The bleed control loop is electrohydraulic. The electronic bleed controller receives pulses from the sensors of supply and return water temperature and from pressure in the extraction stage and in turn

FIGURE 7.

THE CONTROL SYSTEM DIAGRAM OF THE 250 MW TURBINE

1-speed controller; 2-intermediate pilot valve; 3-pressure controller; 4-system for changing the turbine to backpressure operation; 5-servomotors of the HP; 6-servomotors of the IP; 7-servomotor of the LP; 8-pilot valves of the automatic safety device; 9-membrane divider; 10-electric-to-hydraulic converter. A-input from the automatic safety device from electric protection of the turbine and of the unit; B-pulse line to automatic interlocks of the HP and IP; S-signal from the switch of the generator; EC-electronic controller of thermal load; M-motor set point adjuster. Pulses: P_1 from the protective turbine inlet main steam pressure controller (high side); P_2 -pressure in the extraction; T -network water temperature at the outlet of the upper district heat exchanger; t -network water temperature at the outlet of the lower district heat exchanger; t_1 -return water temperature; M_s -manual set point adjuster.

controls the electric motor of the hydraulic pressure controller synchroniser. The hydraulic controller, having a more rapid response than the electronic controller, carries out protective functions according to the pressure of the extraction point. The actuators of both loops are the hydraulic servomotors of the HP, IP, and LP sections of the turbine. The control system may be connected to a computer to govern the turbine operation.

The control system makes it possible for the turbine to operate in the following three modes:

1. Condensing operation without district heating steam extraction
2. Condensing operation with district heating steam extraction
3. Back pressure operational mode.

In the first mode the rotor speed loop controls turbine steam flow through the main control valves of the HP section. The extraction control loop is inactive, and the control diaphragms of the LP section are fully opened. They are closed in this operational mode only when load is shed. Valves of the IP section in this, as in other operational modes, carry out functions of protection against acceleration of the rotor by steam from the reheat line. They start closing due to the action of the speed controller (also due to the action of the synchroniser) when the turbine steam flow is decreased by 40-50% of the full flow.

In the second mode both control loops are in operation. The speed control loop acts as in the condensing mode described above. The extraction control loop, depending on the conditions of the extraction load can maintain temperature of network supply

water. Supply water temperature is controlled when water from the district heat exchangers flows into the district network and the heat load is, therefore, controlled by the turbine. Temperature difference between supply and return water is controlled in the joint operating mode of the turbine with the peak water heating boiler. In this mode variations in load are taken up by the peaking boiler, and the turbine load determined by the temperature difference between the supply and return water, remains unchanged. Extraction steam pressure pulse transmitted to the electronic controller is provided to improve transient performance of network water temperature control. It can also be used to control pressure in the extraction with higher accuracy.

In the third operational mode the servomotor of the control diaphragms of the LP section is set to the lower stop and the diaphragms are in the minimum position. The extraction control loop controls the turbine by means of the HP section valves. This operational mode is possible only when the turbine generator operates in a large electric system where frequency is maintained by other units operating in parallel. In practice this condition does not impose limitations on the use of the backpressure operational mode. The speed control loop in this operational mode carries out protective functions in shedding load.

Water is used as a working medium for the control system. Water supply for the control system is automatic. Use is made of condensate with addition of an inhibitor (0.3-1.5% of sodium nitrite). The inhibitor, along with proper selection of materials for the components used in control prevents corrosion. The effect of the inhibitor is considerable not only in operation but also in time of outage of the turbine.

II.2.2. Turbine T-175/210-130

This turbine was recently developed to supply space heating, domestic hot water and ventilation requirements to cities. Provisions are incorporated in the turbine for its utilization in district heating systems with supply temperatures higher than 300 F.

The turbine has a nominal electric capacity of 175 MWe. The maximum electric capacity of the turbine with the district heat load disconnected is 210 MWe.

The throttle flow for the turbine is 1,643,000 lbs/hr with steam parameters of 1834 psig and 1031F. The operation of the turbine with throttle steam temperature up to 1049F is permitted.

The nominal heat load of the turbine is 314 MWt. The maximum heat load while utilizing the vent steam in the condenser is 326 MWt. The range of the pressure variation in the heating steam extractions is as follows: in the lower stage between 7.1 and 28.5 psia, the upper between 8.5 and 42.7 psia. The nominal temperature of the feedwater is 450F. The design temperature of the circulating water is 68F.

A longitudinal section of the turbine is presented in Figure 8. The turbine is tandem compound with three cylinders. In the HP cylinder the steam is expanded down to 213 psia. From the HP cylinder the steam enters the IP section and afterwards is supplied to the lower extraction

FIGURE 8. A LOGITUDINAL SECTION OF THE 175 MW TURBINE

stage or to the LP cylinder. The pressure after the IP cylinder in the maximum condensing mode is about 11.4 psia. Steam for the upper district heating stage is extracted from the IP section. The steam distribution assembly of the HP cylinder is of the nozzle type with four control valves. The control of the steam flow in the LP cylinder is performed again by grid type diaphragms. The HP cylinder has 13 stages and it is a standard module interchangeable with the turbine R-100-130 described in Section II.3.2. The IP cylinder has 9 stages including two stages located between the upper and lower space heating extractions. The LP cylinders is of double-flow design with three stages in each flow.

The condensate is heated in sequence in the coolers of the main ejectors, ejectors of the gland sealing system, gland steam condenser, four low pressure feedwater heaters, deaerator of 85 psia and three high pressure feedwater heaters. The steam parameters of the regeneration system are presented in Table 3. The data presented corresponds to the nominal electric and heat modes of the turbine, with a pressure in the controlled extraction of 14.2 psia. It is possible to operate the turbine with the high pressure heaters out of service without reducing the throttle steam flow.

The district heating water is heated in series in two heat exchangers which are supplied with steam from the

lower and upper extraction stages. The control system of the turbine is of electro-hydraulic type similar to that described for the T-250 turbine. The control system may be connected to a computer to govern the turbine operation.

The turbine is scheduled to be in operation in 1978 and no more specific data on the machine is available at the present time.

II.2.3. Turbine T-100/120-130

II.2.3.A Design features and operating modes

This turbine has been widely utilized in the USSR for district heat supply for more than ten years. Therefore, more operational data on this typical district machine is available and discussed below.

The turbine has a nominal electric capacity of 110 MWe and nominal heat capacity of 203 MWt. The maximum electric capacity of the turbine with the district heat load disconnected is 120 MWe.

The throttle flow for the turbine is 1,058,000 lbs/hr with steam parameters of 1834 psig and 1031F. A longitudinal section of the machine is presented in Figure 9.

The turbine is tandem compound with three cylinders. In the HP cylinder the steam is expanded down to the pressure of the highest feedwater extraction of 484 psia. The HP cylinder is designed with counter-flow in regard to the IP cylinder. In the IP cylinder the pressure is expanded down to the pressure of the lower space heating extraction.

The low pressure cylinder is of double-flow design and consists of three parts: one middle and two exhausts. To prevent the misalignment of the turbine while operating in the modes with increased pressure in condenser and temperature of the exhaust sections, the supports of the LP cylinder are installed at the elevation of the cylinder joint line. The turbine has

FIGURE 9. A LOGITUDINAL SECTION OF THE 100 MW TURBINE

two district heating steam extractions: upper and lower which are utilized for heating of the district water in stages. The turbine may operate with either one or two extractions. The turbine is supplied with one pressure regulator of the heating extractions and has two control diaphragms located in the LP section of the machine. The control diaphragms are of a single valve throttling type design (Fig. 10). The radial orifices in the rotating ring are of rectangular shape, each nozzle having a separate orifice. Both diaphragms are controlled by one servomotor. The advantage of the control diaphragm is that it could be placed inside the turbine cylinder without substantial cylinder size increase. The diaphragm is as leak tight as possible having a minimum steam leakage rate.

The extraction pressure can be controlled either in the upper stage only by operating two extractions or in the lower stage by disconnecting the upper stage. Operation of the turbine with the upper stage only is not permitted.

The pressure variation in the upper stage is between 8.5 and 35.6 psia, and in the lower between 7.1 and 28.5 psia. While operating with one lower extraction stage it is possible at the lower than nominal heat mode to decrease the pressure in this stage down to 4.3 psia. While operating with two space heating extractions, the pressure in the lower stage is dependent on the heat

FIGURE 10.

THE CONTROL DIAPHRAGM OF THE 100 MW TURBINE

and electric modes and may be lower than 4.3 psia. The distribution of the heat load between the upper and lower extraction stages is determined by the district water temperature before and after the district heat exchangers, by the district water flow and electric load and can not be arbitrarily changed without changing the operation mode.

When the extraction stage pressures are being controlled, operation of the turbine is limited to upper extraction pressure values of 28.4 psia and 35 psia for the lower and upper extraction stages respectively. The pressure in the higher extraction cannot be lower than 8.5 psia when both extraction stages are in operation.

The two district heat exchangers have an overall surface area of 24,750 sq. ft. and are designed for district water flow of 19,800 gpm. The turbine unit is designed for utilization of the steam entering the condenser while operating in the heating mode. In this case, the control diaphragm in the LP cylinder is closed and the turbine is working in the mode identical to the back pressure operation. In other words, all heat supplied to the turbine except for mechanical, generator, and radiation losses is utilized. Utilization of the vent steam in the condenser increases the maximum heat load of the turbine. For steam utilization the condenser is provided with a separate tube bundle and the return

district water or make-up water may be used as a cooling medium. The district heating bundle constitutes about 15% of the overall condenser surface area. The steam flow in the condenser is limited in this operation mode and utilization of the return district water as a cooling medium results in an absolute pressure increase in the condenser. The control system ensures the operation of the turbine at this condition.

The nominal heating load of the turbine is equal to the sum of both district heating extractions, while the maximum heating load includes also, the heat utilized in the condenser. The nominal and maximum heat loads are determined at the nominal operation modes of the turbine, that is at the nominal electric capacity, throttle steam parameters, controlled extraction pressure, minimum steam flow in the condenser, and with complete feedwater system in operation.

The condensate after the condenser is heated in the main ejector coolers, the gland sealing system ejectors, gland steam condenser, four low pressure feedwater heaters, deaerator and three high pressure feedwater heaters.

The optimal value of the cogeneration coefficient for a district heating system with a T-100 turbine is about 0.5. In Figure 2 the dependence of the supply and return district water temperatures from the outdoor

temperature for the T-100 turbine is presented. The variations of the pressure in the extraction stages imposed by the district water temperature changes are presented in Figure 11. From this Figure one can see that the pressure in the upper extraction stage varies during the heating period between 7.1 to 18.0 psia and in the lower extraction in the range of 3.5 F 9.0 psia. If the cogeneration coefficient increases, the pressure in the stages increases accordingly. It should be noticed from Figure 11 that during the operation of the district networks in the summer and basically during the heating period, the pressure in the controlled district heating extraction may be kept less than 17 psia and the heating of the water in the district heaters takes place by relatively constant temperature differences of 104 and 122°F for the two heaters. This permits effective series heating of the district water. Reducing the pressure of the extracted steam down to a minimum, heating the water in stages, reducing the pressure drop in the extraction piping and the terminal difference in the district heaters, results in a reduction of the temperature level in the heat extracted from the cycle and in this way improved the economics of the district heating turbine.

The Performance Diagram

In Figure 12 the performance diagram for the T-100 turbine, while operating with two extractions is presented.

FIGURE 11. DEPENDENCE OF THE STEAM PRESSURE IN THE UPPER (a.) AND LOWER (b.) EXTRACTION STAGES ON THE OUTDOOR TEMPERATURE FOR THE 100 MW TURBINE.

Design temperatures of the supply and the return lines are 300°F and 150°F respectively. The co-generation coefficient is 0.5.

- (1) Heating Period
- (2) Summer Period with Domestic Hot Water Load
- (3) Condensing Mode

FIGURE 12. THE PERFORMANCE DIAGRAM OF THE 100 MW TURBINE

T = SUPPLY DISTRICT WATER TEMPERATURE, °F

EXAMPLE:

GIVEN: HEAT LOAD 150 MWt (POINT A), SUPPLY DISTRICT HEATING TEMP 212°F (POINT B),
ELECTRIC LOAD 100 MWe (POINT F).

RESULT: THE THROTTLE STEAM FLOW WILL BE 933,000 lb/hr (A-B-C-D-E)

F

The diagram takes into account that in this mode the pressure is controlled only in the upper extraction stage and the two extractions are interconnected through the district heat exchangers. In this case, the pressure in the lower non-controlled extraction is determined by the following parameters: heat load, flow characteristics of the stages located between the extraction points, by the characteristics of the district heat exchangers, and the pressure drop in the piping system between the turbine and the district heat exchangers.

During the heating period the district water flow practically stays constant and, therefore, is not included as a parameter in the diagram. The turbine has a wide range of controlled pressure in the upper extraction stage (between 8.5 and 35.6 psia) which provides a supply temperature in the range of between 158 and 257°F. When that temperature and the corresponding extraction pressure changes take place, the capacity of the turbine changes by about 15 percent. The operation mode changes are affecting the three sections of the turbine adjacent to the district heating extractions and, therefore, the temperature of the supply water is included in the diagram as a parameter. Variation of the return water temperature results in a relatively small change in turbine capacity (about one to one and a half percent on every ten degrees of changing of the return temperature) and, therefore, this parameter is not presented in the diagram.

The diagram includes two basic operation modes:

1) to meet heat load requirements with a fixed relationship between the heat and electric loads and 2) to meet electric load requirements with independently varying heat and electric loads. While operating in the first mode, the control diaphragms of the low pressure section are closed, and the load control is provided by the control valves of the HP cylinder. The steam flow of the turbine is determined in this mode by the heat consumption, by the steam leakage through the closed diaphragms of the LP section, and the steam extractions for the feedwater system. When the turbine operates in the electrical mode, the control diaphragms in the LP section have some degree of opening, which permits for a given heat load additional steam flow into the condenser. This steam flow will generate additional electrical capacity.

II.3. Turbines for Combined District Heating and Industrial Applications

In the USSR cogeneration power plants very often serve both industrial customers who need high pressure process steam, and urban customers, who require hot water heating on a district heating network. The economic feasibility of such an integrated system depends on many factors. One of the major factors is the colocation or proximity of both plants. The cogeneration plant usually is situated close to the industrial complex in order to economically supply industrial steam at high pressure. The location of the power plant at a substantial distance from the urban customers does not pose a problem since hot water can be pumped for extensive distances.

Special type cogeneration turbines for combined industrial and district heat supply have been developed and are widely used. Several different types of turbines are installed at the cogeneration power plant for such a combined heat supply. They are as follows: pure district heating turbines, discussed above; combined industrial and district heating turbines and back pressure turbines, which will be discussed in the following section.

II.3.1. Turbine PT-135/165-130/15

This turbine is designed to supply steam to industrial customers and hot water for the urban users. The turbine has a nominal capacity of 135 MWe, a nominal industrial steam extraction of 706,000 lbs/hr, and nominal district heating extraction as a sum of two extractions of 128 MWt. The throttle flow for the turbine is 1,630,000 lbs/hr with steam parameters of 1834 psig and 1049F. The operation of the turbine with throttle steam temperature of 1031F is also permitted.

A longitudinal section of the turbine is presented in Figure 13. The turbine is a tandem compound machine with two cylinders. The HP cylinder has 13 stages, the main steam assembly is of the nozzle type with four control valves. From the HP cylinder the steam through four pipes is supplied into the industrial extraction stage and through four pipes to the control valves of the IP/LP cylinder. These control valves are located in steam chambers which are welded to the steam inlet section of the IP/LP cylinder. In the IP/LP cylinder are located separately six and two stages of the IP section, and three stages of the LP section. The steam extractions to the upper and lower district heating stages are from the chambers of the IP section. The pressure in the heating extractions is controlled by means of two grid diaphragms.

The IP/LP cylinder consists of three parts: a cast steam inlet part with welded steam and nozzle chambers, a cast intermediate section, and the all-welded exhaust section. The exhaust section is supported by the foundation frames from the bearing ends and on both sides. The fixed point of the turbine is located on the side supports. The expansion of the turbine takes place in the direction of the front bearing.

FIGURE 13. A LOGITUDINAL SECTION OF THE 135 MW TURBINE

The turbine has three controlled extractions: one industrial and two district heating extractions, upper and lower. The ability to control simultaneously the pressure in the two district heating extraction points increases the operational capability of the turbine. For example, when the turbine is installed at an industrial or industrial-heating cogeneration plant, there is a possibility of utilizing steam from the district heating extractions for two different users. The variation of the pressure in the heat extractions by independent pressure control in each of them is as follows: in the upper stage between 12.8 to 35.6 psia; in the lower between 5.7 and 17.1 psia. When operating with two controlled district heating extractions, pressure in the upper extraction should be higher than the pressure in the lower stage by at least 7.1 psia. The turbine may also be operated in the modes typical for district heating turbines with two extractions. In this case, the pressure is controlled then only in the upper stage and the district water is heated in two series heat exchangers. The control diaphragm of the upper stage at this time is completely open. The range of pressure variation by operating both extractions and controlling the pressure in the upper stage is between 8.5 and 35.6 psia. The pressure in the lower extraction may be increased up to 38.4 psia in this mode.

The turbine may operate also with one district heat extraction, either upper or lower, with the second stage disconnected. The pressure range variation will then be between 8.5 and 35.6 psia in the upper heat extraction (by disconnecting the lower stage), and between 5.7 and 17.1 psia in the lower stage (by disconnecting the upper stage). The controlled pressure in the industrial extraction may vary between 171 and 299 psia.

At the nominal electric mode the industrial flow is 860,000 lbs/hr at a pressure of 213 psia with district heat extractions disconnected. By decreasing the electrical capacity down to 100 MW, it is possible to increase the industrial extraction flow up to 1,058,000 lbs/hr by disconnecting the heating load. The maximum heating load of both extractions is 128 MWt. The district heating load may be increased up to 163 MWt by utilizing other power plant heat exchangers for heating the makeup water. The industrial extraction is then disconnected and the turbine capacity is about 100 MWe. The nominal capacity of 135 MWe will be achieved with the industrial extraction of about 507,000 lbs/hr. The turbine may take electrical load up to 165 MWe with industrial steam extraction flow between 485,000 and 557,000 lbs/hr and district heating load disconnected. The maximum capacity of the turbine in the condensing mode is 120 MWe.

It is possible to operate this turbine at 92F circulating water to the condenser (design temperature 68F) at a water flow rate of 54,600 GPM without reducing the throttle steam parameters.

The expanded steam enters the condenser having a surface area of 64,560 sq. ft. The condenser has three separate tube bundles, two major and one district heating, located in one shell, each having separate water boxes. The air removal system of the condenser is by means of ejectors designed to utilize saturated or slightly superheated steam of 71 psia in front of the main ejectors and 64 psia in front of the startup ejectors.

The automatic control system of the turbine provides the turbine speed control and steam control in the three control extractions.

Steam is provided for feedwater heating in series in the coolers of the main ejectors, ejectors of gland sealing system, gland condenser, four low pressure feedwater heaters, deaerator of 85 psia, and three high pressure feedwater heaters. The low pressure heaters 1 and 2 are supplied with steam from the extractions after the 22nd and 20th stages. Low pressure heaters 3 and 4 are supplied with steam from noncontrolled extractions after 18th and 16th stages. Deaerator and the high pressure heater 5 are provided with steam from the industrial extraction after the 13th stage. The high pressure heaters 6 and 7 are supplied with steam from the noncontrolled steam extraction after 9th and 11th stages. The steam parameters of the regeneration system are presented in Table 3. The data presented corresponds to the nominal electric mode of the turbine with nominal heat load, pressure in the industrial extraction of 213 psia and in the upper space heat extraction of 11.4 psia.

The drains from the high pressure heaters are cascaded to the deaerator. The drains from low pressure heater 4 are cascaded into low pressure heater 3 and afterwards by means of a drain pump are pumped into the main condensate line. The drains from the low pressure heaters 1 and 2 are cascaded to the hot wells of the district heat exchangers 1 and 2 and afterwards are pumped together with the drains of these heaters into the main condensate line.

Provisions are made for heating and deaeration of the return industrial condensate and makeup water by utilizing both district heat extractions, the noncontrolled extraction after the 16th stage, and also steam entering the condenser.

II.3.2. Turbine R-100-130/15

The back pressure turbine of nominal capacity of 100 MWe is designed for industrial steam supply. The throttle flow of the turbine is 1,676,000 lbs/hr with steam parameters of 1834 psig, 1049F, and back pressure of 213 psia. The operation of the turbine with a lower throttle steam temperature of 1031F is also permitted. The operation of the turbine is also possible using the following variation of the steam parameters: throttle steam pressure between 1763 to 1906 psig, temperature between 1031 and 1058F, and back pressure between 171 and 299 psia. The maximum electric capacity of the turbine with a back pressure of 171 psia is 107 MW. The maximum steam flow in the back pressure line is 1,433,000 lbs/hr.

A longitudinal section of the turbine is provided in Figure 14. The turbine is manufactured with one cylinder having 13 stages. The steam to the turbine is supplied from two separately installed stop valves. Steam distribution is of the nozzle type with four valves, which are installed on the turbine. The cylinder in the zone of steam inlet and left side steam flow is of the double wall design. The steam enters the internal cylinder in the middle of the casing. The external cylinder and inlet section of the internal cylinder have a barrel shaped form which limits the stresses in the high temperature section.

The turbine may supply noncontrolled extraction steam of 198,000 lbs/hr after the second stage with pressure of about 500 or 550 psia for external use. In this mode the capacity of the turbine and the back pressure steam flow are reduced accordingly.

A control system keeps constant one of two parameters: either the turbine speed or the steam back pressure when the turbine operates on electrical or heat loads, respectively.

FIGURE 14. A LOGITUDINAL SECTION OF THE 100 MW BACK PRESSURE TURBINE

The turbine may be operated on the back pressure line in parallel with other machines having an automatic pressure control.

Steam is provided for feedwater heating in series in the deaerator at 85 psia and three high pressure feedwater heaters, supplied with steam from the 9th and 11th stages and from the back pressure line. The steam parameters of the regeneration system are presented in Table 3.

II.4 District Heating Turbines Manufactured in Western Europe

The development of district heating turbines in Western Europe has been similar to that in the USSR; however, there are some differences in design, which are briefly discussed in this section.

II.4.1. Butterfly Valve Control Type of Turbines

The large European district heating turbines are equipped with a condensing tail, which makes it possible to vary the ratio of the heat to electric output. One of the major features of these types of turbines is utilization of butterfly valves in the crossover pipes and in the extraction pipes between the IP and LP elements in order to control the heat and electrical generation.

A simplified control diagram of the turbines based on Brown Boveri Company (BBC) reference¹⁰ is presented in Figure 15. One system presented in Figure 15a uses butterfly valves R_1 in the crossover pipes in order to throttle the steam entering in the LP section of the turbine. On the other hand, butterfly valves R_2 in the extraction pipes control the steam flow into district heat exchangers

At the design condition the butterfly valves are fully open; however, when the loads differ from nominal, steam throttling takes place.

Another control system presented in Figure 15b utilizes bypass lines around district heat exchangers on the water instead of steam extraction butterfly valves in order to control the temperature of the supply district heating water. At the design condition the bypass valves are closed; however, by load variation, mixing of the district water takes place which results in appropriate thermal losses.

FIGURE 15.

A CONTROL DIAGRAM OF A BUTTERFLY VALVE CONTROL TYPE TURBINES

- a: Control by butterfly valve of the heating steam
 b: Control by bypass of the heating water

S = Steam generator
 T = Turbine
 H = District heat exchanger
 C = Condenser
 V = Live steam valves
 R = Governor

R₁ = Crossover butterfly valve
 R₂ = Extraction butterfly valve
 R₃ = Control bypass
 t₀ = Outdoor temperature
 t' = Supply temperature
 t'' = Return temperature

In a pure condensing mode the crossover butterfly valve is completely open while the extraction butterfly valves are closed. At the maximum heat load the turbine may operate in a mode similar to back pressure with crossover valves fully closed. A small steam flow should be always provided into the LP section to cool the blading system.

Single and two-stage systems to heat the district water are utilized, their schematic diagrams based on BBC reference¹⁰ are presented in Figure 16. The arrangements presented in Figures 16b and 16c are mostly used. A longitudinal section of turbines arranged according to the systems 16b and 16c are presented in Figures 17 and 18, respectively.

The largest European district heating turbines are manufactured according to the system 16d. In Figure 19 a longitudinal section of such an installation with double-flow unsymmetrical IP section is presented. In Figure 20 an overall view of this machine is presented.

FIGURE 16.

SINGLE AND TWO STAGE DISTRICT HEATING
TURBINE SYSTEMS.

- a. Single-stage heating
- b. Two-stage heating with extraction from the IP section
- c. Two-stage heating with extraction from the IP and LP section
- d. Two-stage heating with double-flow intermediate-pressure section

H = District heat exchanger

FIGURE 17.

A LOGITUDINAL SECTION OF THE DISTRICT HEATING TURBINE WITH REHEAT IN HAMBURG PORT AREA

Electrical output 80 MWe, heat output 146 MWt. Two-stage heating. The extraction point H_3 was not used.

FIGURE 18.

A LOGITUDINAL SECTION OF THE DISTRICT HEATING TURBINE IN
ÅLBORG, DENMARK

Electrical output 285 MWe, heat output 267 MWt
Two-stage heating.

H = Heating condenser

FIGURE 19.

A LOGITUDINAL SECTION OF THE DISTRICT HEATING TURBINE
AROS II IN VASTERÅS, SWEDEN

Electrical output 270 MWe, heat output 372 MWt. Reheat
turbine with double-flow intermediate-pressure section
for two-stage heating.

FIGURE 20.

AN OVERALL VIEW OF THE DISTRICT HEATING TURBINE AROS II IN VÄSTERÅS,
SWEDEN

II.4.2 Back Pressure Turbines

In the back pressure turbines the steam flow through the machine follows the district heating requirements which are directly related to power generation. The main valves of the turbine control the supply temperature of the district heating water according to outdoor temperature (Figure 21). The back pressure of the machine varies accordingly. Small back pressure installations usually use one heating stage, while medium and large ones utilize two stages.

Turbines with capacity above 100 MWe usually have double-flow unsymmetrical design of the blading system which prevents the unbalance in the admission side of the district heat exchangers. As an example, in Figure 22, a 200 MWe back pressure turbine for two-stage heating is presented.

FIGURE 21.

A CONTROL DIAGRAM OF A BACK PRESSURE
TURBINE

S = Steam generator
 T = Turbine
 H = District heat exchanger
 V = Live-steam valve
 R = Governor
 t_o = Outdoor temperature
 t' = Supply temperature
 t'' = Return temperature

FIGURE 22.

A LOGITUDINAL SECTION OF A BACK PRESSURE TURBINE WITH
REHEAT FOR TWO STAGE HEATING

Electrical output 197 MWe, heat output 349 MWt

II.5 District Heating Turbines for Nuclear Power Plants

The experience with district heating systems supplied with heat from nuclear power plants is very limited. The well known Swedish Agesta reactor has supplied for ten years 12 MWe of electricity and 55 MWt of heat through 1974 when it was decommissioned. Two nuclear power plants in the U.S.S.R. are now in operation supplying district heating and domestic hot water load to customers.

One U.S.S.R. district heating system has been supplied with thermal energy for ten years from Bielyarsk nuclear power plant. This power plant is equipped with graphite-moderated pressure tube reactors which generate nuclear superheated steam of 1450 psia and 896°F. One reactor has a dual-cycle system, and the second is of direct cycle type. The steam is supplied to three 100 MWe condensing type turbines. Low pressure non-controlled extraction steam is utilized to provide a district heating load of 24 MWt to residential customers and greenhouses located 3 km away from the power plant. The district heating system has two loop (double barrier) closed type heat exchangers.

The other U.S.S.R. cogeneration power plant is located at Bililino and has four graphite-moderated pressure tube reactors with natural circulation. The overall plant capacity is 48 MWe and 116 MWt. Each of the T-12-60 turbines is supplied from the reactor with 212,000 lb/hr of saturated steam of 855 psia. The turbine has two steam extractions for district heating. The district heating system has two loop closed type heat exchangers.

The operational experience of the two plants has proved the technical and economic feasibility of district heating supply from nuclear power plants.

In the last few years, considerable effort is being made in the U.S.S.R. to develop large scale district heating systems supplied from nuclear power plants. A typical U.S.S.R. fossil fuel cogeneration power plant size (consisting of several units) is about 600 to 1000 MWe and 1500 to 2000 MWt. The

environmental problems associated with such large power plants in city areas require remote location of the cogeneration power plants. This results in a substantial increase of heat transport cost. Under these conditions, the cogeneration nuclear power plants are becoming competitive with the fossil fueled cogeneration plants. To reduce the heat transport cost, a single-pipe transmission system has been developed. The nuclear district heating supply methods considered for the future in the U.S.S.R. are as follows:

1. Installation of new nuclear power plants which will utilize the typical PWR and graphite-moderated reactors, already introduced in the condensing type power plants. However, for these types of stations, special type district heating turbines utilizing saturated steam have to be developed. Such a 500 MW turbine is now under design in the U.S.S.R.

2. Further development of graphite-moderated reactors with nuclear superheating of large size. This will permit the use of existing types of district heating turbines, including the T-250-240.

3. Installation of new single purpose nuclear heating plants (heat only).

4. Installation of additional district heating units on existing nuclear power plant sites located relatively close to the heat load centers.

5. Heat supply from low pressure noncontrolled extractions of the conventional type nuclear power plants.

No specific details of the large nuclear district heating turbines referred to above are yet available in current U.S.S.R. technical literature.

III PERFORMANCE OF THE T-250/300-240 TURBINE UNITS AS APPLIED TO THE MINNEAPOLIS-ST. PAUL DISTRICT HEATING SYSTEM

In this section the possibility of utilizing the T-250/300-240 turbine design for the Minneapolis-St. Paul district heating system is analyzed and associated fuel savings are evaluated. The total connected heat load for Minneapolis-St. Paul area is 2642 MWt (1784 MWt for Minneapolis and 858 MWt for St. Paul). Four T-250 turbines could be installed to supply the base heat load of $384 \times 4 = 1536$ MWt. The peaking load ($2642 - 1536 = 1106$ MWt) is assumed to be supplied by means of peaking water heating boilers. The cogeneration coefficient of the system in this case will be $\frac{1536 \text{ MWt}}{2642 \text{ MWt}} = 0.58$. To assess the fuel savings, the annual fuel consumption of the proposed cogeneration power plant and peaking boilers should be compared with the fuel consumption of the conventional power plants providing electricity only and the individual boilers providing heat only. In other words, to obtain the annual fuel savings, two alternates should be compared, as follows:

1. Cogeneration power plant supplying electricity and base heat load, and water heating boilers supplying peak heat load.
2. Conventional power plants providing electricity and individual heating plants supplying heat to the customers.

In both cases the same amount of electricity and heat is assumed to be supplied to the customers.

Comparing the above alternates, it is very important to properly select the performance characteristics of the conventional power plant. As a basis for comparison, in alternate 2, it is assumed that this conventional power plant is of the highest performance available. The performance of an existing conventional power plant for alternate 2 would be considered in the situation when this plant is to be replaced by a new cogeneration power plant.

The annual fuel consumption of a cogeneration power plant will depend on the variety of operational modes, and will take into consideration weighted average extraction steam flows and pressures during the year.

Another important aspect of the comparison analysis should also be mentioned. The heat rate of a conventional power plant is lower than the heat rate of a cogeneration power plant with the same steam parameters, but operating in a condensing mode. This is explained as follows:

1. In a nominal operating mode the blading system of conventional power plants operates at the design conditions with maximum efficiency. Investment in the condenser cooling system to provide maximum vacuum is justified. In a cogeneration turbine the steam flows through the turbine sections and in the condenser vary during the year depending upon the extraction loads, and therefore, the annual turbine efficiency is lower when compared to a conventional condensing unit.

2. The unit size and throttle steam parameters are usually higher for conventional turbine plants. This further contributes to the heat rate reduction for conventional power plants.

Therefore, an associated penalty should be applied against cogeneration power plants when estimating fuel savings.

To estimate the fuel savings the result of an extensive study ¹⁴ taking into account the above-discussed factors for the T-250/300 turbine are used. In Figure 23 the annual fuel savings as a function of the cogeneration coefficient and conventional power plant heat rate are presented. Other influencing factors taken into account in Fig. 23 are as follows:

. Duration of the heating season	5500 hrs/yr
. Electrical capacity factor	0.68
. Difference in power transmission losses and auxiliary power consumption between cogeneration and conventional power plants	0
. Power plant boiler efficiency	0.88
. Individual heating plant efficiency	0.88
. Water heating peaking plant efficiency	0.88

If the actual values of the above parameters are different from the values adopted in constructing Fig. 23, correction factors presented in Figures 24 to 27 should be applied.

To estimate the fuel savings for Minneapolis St. Paul district heating system the following parameters are considered:

. Total connected heat load	2642 MWh

FIGURE 23.

THE DEPENDENCE OF FUEL SAVINGS ON THE COGENERATION COEFFICIENT FOR THE 250 MW TURBINE

FIGURE 24.

THE DEPENDENCE OF FUEL SAVINGS ON THE
HEATING PLANT EFFICIENCY

FIGURE 25.

THE DEPENDENCE OF THE FUEL SAVINGS OR LOSSES ON
THE COGENERATION POWER PLANT BOILER EFFICIENCY (R)

FIGURE 26.

THE DEPENDENCE OF THE FUEL SAVINGS OR LOSSES ON THE DIFFERENCE IN POWER TRANSMISSION LOSSES AND AUXILIARY POWER CONSUMPTION BETWEEN COGENERATION AND CONVENTIONAL POWER PLANTS.

FIGURE 27.

THE DEPENDENCE OF FUEL SAVINGS OR LOSSES ON THE HEATING SEASON DURATION (a) AND ELECTRICAL CAPACITY FACTOR (b) OF THE 250 MW TURBINE

. District heating load	2299 MWt
. Domestic hot water load	343 MWt
	(13% of the total heat load)
. Duration of the heating season	6200 hrs/yr
. Number of T-250/300 turbines installed	4
. Total electric capacity of the cogeneration power plant	1200 MWe
. Total district heating capacity of the cogeneration power plant	1536 MWt
. Conventional power plant heat rate	9400 BTU/KWhr
. Electrical capacity factor	.74
. Individual Heating plant efficiency	.75
. Water heating peaking plant efficiency	.88
. Cogeneration power plant boiler efficiency	0.87
. Difference in power transmission losses and auxiliary power consumption between cogeneration and conventional power plants	+3%

Taking into account the above parameters and applying Figures 23 to 27, the fuel savings for Minneapolis-St. Paul district system can be estimated as follows:

The amount of heat saved annually per MW of thermal load --

$$q = \begin{array}{cccccc} \text{Fig.23} & \text{Fig.24} & \text{Fig.25} & \text{Fig.26} & \text{Fig.27(a)} & \text{Fig.27(b)} \\ (8.12 + 2.1 - 0.4 - 0.92 + 0.66 + 0.26) \times 10^9 = \\ 9.82 \times 10^9 \frac{\text{BTU/yr saved}}{\text{MWt of heat load}} \end{array}$$

The total amount of heat saved annually is --

$$Q = 9.82 \times 10^9 \frac{\text{BTU/yr saved}}{\text{MWt of heat load}} \times 2642 \text{ MWt} = 2.6 \times 10^{13} \text{ BTU/yr}$$

Or --

$$\Delta W = \frac{2.6 \times 10^{13} \text{ BTU/yr}}{6.2 \times 10^6 \text{ BTU/barrel of oil}} = 4.2 \times 10^6 \text{ Barrels of oil/yr saved}$$

Or

$$\Delta W = \frac{2.6 \times 10^{13} \text{ BTU/yr}}{16.6 \times 10^6 \text{ BTU/ton of coal}} = 1.6 \times 10^6 \text{ Tons of Western Coal/yr saved}$$

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